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Design and Development of an Integrated Rice Milling Plant

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Abstract

Most Nigerians prefer imported rice to the locally produced ones which contained a lot of pieces of stones. The presence of pieces of stone in the locally processed rice is due to poor handling. The parboiled rice is sun-dried by spreading on top of mats on the ground, and pieces of stone find their way into the rice. In attempt to improve the quality of rice through better processing technique, this paper focuses on design and construction of an integrated rice milling plan. This system has a drying unit which works on fluidized bed drying technology. The dried rice is passed into the miller for dehulling in a sequential manner.

The parameters of fluidized bed were calculated to establish minimum fluidizing velocity, bed height and pressure drop. The various forces at play in the rice milling during the process of milling have been analysed and calculated. The parboiled rice had moisture content of 32% and after fluidized bed drying, the moisture content drop to 18%. The rice was milled in the miller and during the process parameters like milling rate, total mill recovery and coefficient of husking were determined and presented in the text.

Keywords

Parboiled rice, fluidized bed, dehulling, fluidizing velocity, pressure drop, milling rate, moisture content, mill recovery, coefficient of husking.

Introduction

There are three key types of milling machine, viz; the huller, cone-mill machine and rubber roll mill. The huller removes the husk and the bran in a single operation. It consists of a hopper mounted on a half hollow cylinder, which contains a solid fluted cylinder that rotates within the assembly. The lower half of the cylinder is perforated. As for the cone-mill machine, it consists of a twin horizontal disk with abrasive coating. The upper disc is stationary while the lower one rotates. The machine hulls and polish rice in separate operations and the polishing is done in a separate unit. The rubber roll mill consists of hopper, rubber roll aspirator and polishing unit.

On the belt drive mechanism it was noted by Shingley and Mischke (1989) that the tighter the belt, the more power that can be transmitted, but the more damaging is the tension both to the belt and to the supporting shafts and bearings. It was further observed that for a given maximum tension, a v-belt can transmit more power than the flat belt. For this reason v-belts are preferred over flat belts although v-belts are slightly less efficient than the flat belts i.e. 70 – 96% for v-belts and about 98% for flat belts respectively. To improve the efficiency of v-belts a number of belts can be used on a single sheave in a multiple manner. In belt drive design, Zimmerman (1997), explained that it is essential to have not only the equation of the power transmitted but also a relationship linking belt tensions to the belt size, strength, and speed and to the drive geometry and the frictional co-efficient. Further more Artus (1977), noted that the total tension required by a v-belt is independent of the brand type or number of belts. He explained further that, two belt drives that are identical require the same total tension but the drive with fewer belts require greater tension per belt.

In the design of shaft, Allen (1992), said shaft design consists primarily of the determination of correct strength and rigidity when the shaft is transmitting power under various operating and loading conditions. He explained further that, bending and torsional moments are the main factors influencing shaft design. It was observed by Aaron (1982), that to determine shaft diameter. It will be necessary to calculate the bending moment and the torque distribution along the full length of the shaft, when gear, pulleys, flywheels, friction wheels, cams and ratchets are mounted on shaft in various combinations and locations. On the shaft geometry, Shigley and Mischke (1989), observed that there is no rigid formula to determine it for any given design situation. It was recommended that, geometry of a shaft should generally be a stepped cylinder. A suggestion was also made that the best approach would be to study the existing designs to learn how similar problems have been solved and then combining the best of these solutions to solve the immediate problem. Knowledge about the shaft deflection is very essential as noted by Aaron (1982), because it can be used to establish the minimum permissible clearance between the pulley, gear and housing for shaft assembly. It was noted further that, it helps in determining the minimum bearing clearance for sleeve bearings as well as self-align bearings required.

On the design of drier, Archibong and Manalabi (1995), explained that, the successful development of mechanical driers involves both technical and socio-economic considerations. It was suggested by Hein et al (1995) that for paddy with very high moisture content of 30% m.c. and over, fluidized bed drying should be more suitable than an ordinary columnar drier where moist grains tend to stick to metal surfaces and block flow. Report was given by Simon (1995), that paddy can be dried in rotary drum driers too. Advantage of this process as observed by Teter et al (1986), is that mechanical driers are not affected by the whether conditions. Although, they explained that, their efficiency depends upon the climatic conditions, type of the paddy and its moisture content at the point of drying. The use of artificial driers was initially impeded by resulting air pollution as submitted by Bakker-Arkema et al (1995), and said further that it was after the initiation and success of dust filtration accessories they came to be generally accepted. Some experiments on batch fluidized bed drying of paddy in Australia were reported by Sutherland et al (1992) and noted some negligible loss in head rice yield when paddy was dried from 26 – 18%m.c. using air at 60 – 90°C.

On fluidizing gas Teter et al (1986) said, even natural air can be used as drying medium. They noted that the main problem will be getting the moisture content to below eighteen per cent within twenty hours to prevent costly damage to the paddy. Generally, some advantages of fluidized bed drying over mechanical driers with rotating drums were listed by Hien, et al (1995) as :- homogeneous drying due to thorough mixing of air and grain and short drying time due to high heat and mass transfer rates. This observation was supported by Yonglin and Graver (1995) who also observed that, in fluidized bed driers, the grain is well mixed with the drying air during the fluidizing process. In order to develop the technology locally and support poverty alleviation of the present government and also to inform the wider world the success in the development effort of this technology, this paper is presented.

2.0 Design analysis and results of design calculations

2.1.1 Design analysis

In the design of belt drive, the mean torque has to be calculated first and foremost. The mean torque can be calculated using equation 2.1

$$T_m = \frac{P}{\omega_1} \qquad 2.1$$

Where, P is the rated power of the motor i.e. 5h.p (3.73kW)

ω is the angular velocity of the electric motor i.e. 1450 rev/min (15184 rad/s).

To determine the output pulley speed, the speed of the driven pulley n_2 can be calculated using the equation 2.2

$$n_2 = \frac{n_1 \times d_1}{d_2} \qquad 2.2$$

Where, n_1 is the speed of the electric motor

d_1 is the pitch diameter of the drive pulley

d_2 is the pitch diameter of the driven pulley.

The mass of the belt is given by equation 2.3, while the normal acceleration in relation to angular velocity is given by equation 2.4.

$$m = bt\rho l \quad (kg) \qquad 2.3$$

Where, b is the belt width

t is the belt thickness

ρ is the belt density

l is the length of belt

$$a = \omega_1^2 R_1 \quad (m/s^2) \qquad 2.4$$

The centrifugal force on the belt can be calculated using equation 2.5

$$F_c = m\omega_1^2 R_1 \quad (N) \qquad 2.5$$

Where, m is the mass per a length of the v-belt (0.28kg/m)

R_1 is the radius of the drive pulley

For unequal pulley radii, angle of twist (α) can be expressed by equation 2.6

$$Sin\alpha = \frac{R_2 - R_1}{C} \qquad 2.6$$

Where, R_1 is the radius of the drive pulley

R_2 is the radius of the driven pulley

C is the distance between the pulleys

The angle of contact on the small pulley is given by equation 2.7

$$\theta = \pi - 2\alpha \qquad 2.7$$

Where, α is the angle of twist

The expression for v-belt tension relationship is given by equation 2.8

$$\frac{F_1 - F_c}{F_2 - F_c} = \frac{f\theta_1}{e^{\sin\beta}} \qquad 2.8$$

Where, F_1 is the tension on the tight side

F_2 is the tension on the slack side

f is the coefficient of friction for the belt = 0.25 for a rubber canvass belt (source Zimmerman, 1977)

F_c is the centrifugal force on the belt

θ_1 is the angle of contact on the small pulley

β is the v-belt section angle = 20° (source Shigley and Mischke, 1989).

The transmitted torques are calculated using equations 2.9 and 2.10

$$T_2 = (F_1 - F_2) R_2 \qquad 2.9$$

$$T_1 = (F_1 - F_2) R_1 \qquad 2.10$$

If F_1 is made the subject of formula in equation 2.8 and then we replace the right hand side of the equation with γ then we now have equation 2.11 as follows:-

$$F_1 = F_c + \gamma(F_2 - F_c) \qquad 2.11$$

Where γ is the co-efficient of friction.

If F_2 is made the subject of formula in equation 2.10, then we now have equation 2.12 as follows:-

$$F_2 = F_1 - \frac{T_1}{R_1} \qquad 2.12$$

Replacing F_2 in equation 2.11 with the value of F_2 in equation 2.12, we now have equation 2.13 as follows:-

$$F_1 = F_c + \left[\frac{\gamma}{\gamma - 1}\right]\frac{T_1}{R_1} \qquad 2.13$$

In equations 2.9 and 2.10 we have a relationship $F_1 - F_2$ called the net driving force. The net driving force on the sheave can be expressed by equations 2.14 and 2.15.

$$F_N = F_1 - F_2 \qquad 2.14$$

$$F_N = \frac{T_1 \times D}{2} \qquad 2.15$$

Where, T_1 is the transmitted torque

D is the diameter of the sheave.

For v-belt drives, the ratio of the tight side tension to slack side tension is given in equation 2.16.

$$\frac{F_1}{F_2} = 5 \qquad 2.16$$

and the relationship between the bending force F_b and the net driving force F_N for v-belts is expressed in equation 2.17 as follows:-

$$F_B = 2.0F_N \qquad 2.17$$

2.1.2 Calculation of the minimum Shaft Diameter

The minimum shaft diameter can be calculated by combining the distortion energy theory with the modify Goodman line of failure for metal fatigue. Thus, for a shaft with steady torsion and reversed bending, minimum shaft diameter is given in equation 2.18.

$$d^3 = \frac{32n}{\pi}\left[\frac{k_f M_a}{S_e} + \frac{\sqrt{3T_m}}{2S_{ut}}\right] \qquad 2.18$$

Where, M_a is the mean applied bending moment

T_m is the mean applied torque

S_e is the endurance limit

S_{ut} is the minimum ultimate tensile strength

n is the design factor of safety

k_f is the fatigue stress concentration factor.

For grey cast iron, $S_{ut} = 431$ MN/m^2, $S_e = 169$ MN/m^2, $k_f = 1.50$

Source: Shigley and Miscke (1989).

The schematic diagram of the load distribution on the shaft is given in Figure 2.1.

To calculate the reactions at the support, we first of all find the bending force on the sheaves. Equation 2.12, 2.13, 2.14 and 2.17 are used to calculate F_N, F_1, F_2 and F_B respectively. With reference to Figure 2.1, sheave A is located at 20° below the shaft. The vertical component of F_B can be calculated by equation 2.19, while the horizontal component is given in equation 2.20.

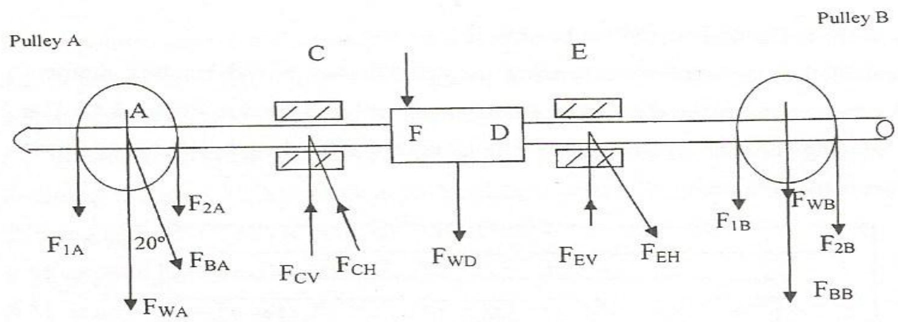

Fig. 2.1: Schematic diagram of load distribution on the shaft.

$F_{BAV} = F_BCos\ 20°$ 2.19

$F_{BAH} = F_BSin\ 20°$ 2.20

Sheave B is located at the same vertical plane with the drive shaft. The bending force therefore is given by equation 2.21.

$F_{BB} = F_1 + F_2$ 2.21

This force acts parallel to the line of centers of the two pulley. When we take the moment of force about the bearing C in the vertical plane, we have equation 2.22.

$$F_{BA}Cos\ 20°(1 - X_1) + F_{WA}(1 - X_1) + F_{EV}(X_1 - X_4)$$
$$= F_{WF}(X_1 - X_2) + F_{WD}(X_1 - X_3) + F_{BB}X_1 + F_{WB}X_1 \quad 2.22$$

Where, F_{WA} is the weight of the pulley A

F_{EV} is the vertical reaction at bearing E

F_{WF} is the weight of rice on the shaft

F_{WD} is the weight of the fluted cylinder

F_{WB} is the weight of the pulley B.

Summing up the forces in the vertical plane, we have equation 2.23.

$F_{BA}Cos\ 20° + F_{WA} - F_{EV} + F_{WF} + F_{WD} + F_{BB} + F_{WB} - F_{CV} = 0$ 2.23

In the same manner, the moment about the bearing C can be taken into account in the horizontal plane and the reactions at the support can be calculated as shown in equation 2.24.

$F_{BA}Cos\ 20°(1 - X_1) = F_{EH}(X_1 - X_4)$ 2.24

Summing up the forces in the horizontal plane, we have equation 2.25.

$F_{BA}Sin\ 20° - F_{CH} + F_{EH} = 0$ 2.25

Where, F_{CH} is the horizontal reaction at bearing E

In calculating the maximum bending moment, the shear and bending moment diagrams for the vertical plane are first of all sketched, and presented in Figure 3.8. The maximum bending moment occurs at a point along the shaft length where the shear diagram crosses the zero axis.

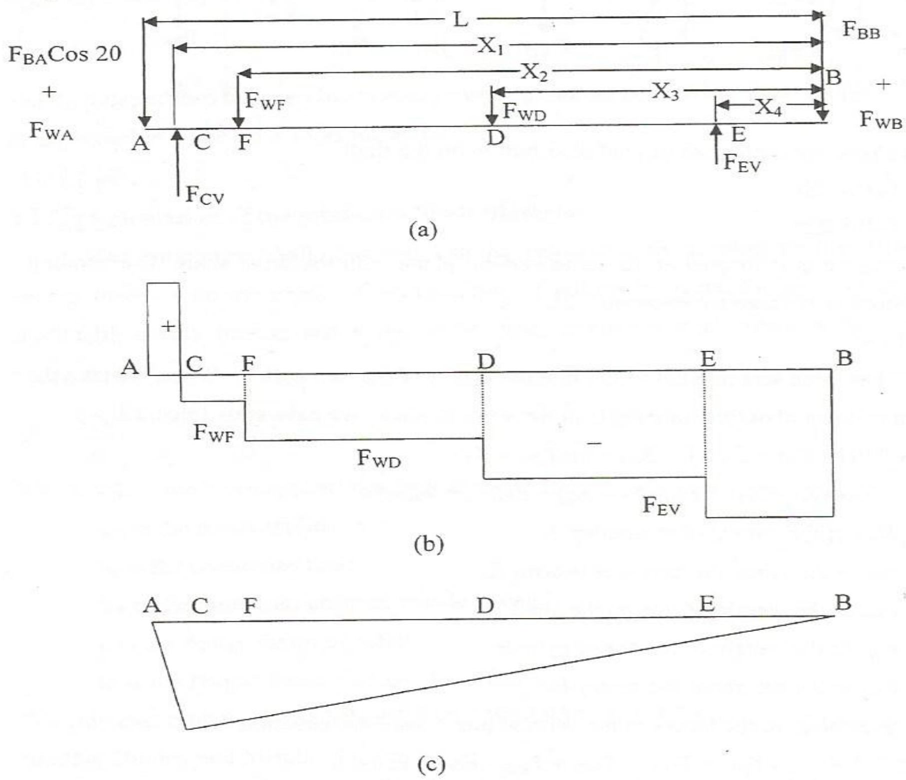

Fig. 2.2: The shear and bending moment diagrams

(a) Vertical plane load diagram

(b) Shear force diagram

(c) Bending moment diagram in the vertical plan

Referring to Figure 2.2(b), we take downward forces as negative. The shear force (S.F.) at point A up to C = $F_{BA}\text{Cos } 20° + F_{WA}$

S.F. at point C up to F = $F_{BA}\text{Cos } 20° + F_{WA} - F_{CV}$

S.F. at point F up to D = $F_{BA}\text{Cos } 20° + F_{WA} - F_{CV} + F_{WF}$

S.F. at point D up to E = $F_{BA}\text{Cos } 20° + F_{WA} - F_{CV} + F_{WF} + F_{WD}$

S.F. at point E up to B = $F_{BA}\text{Cos } 20° + F_{WA} - F_{CV} + F_{WF} + F_{WD} - F_{EV}$

Referring to Figure 2.2(c), which is the bending moment diagram.

Bending moment (B.M.) at point C = $(F_{BA}\text{Cos } 20° + F_{WA})(1 - X_1)$

B.M. at point F = $(F_{BA}\text{Cos } 20° + F_{WA})(1 - X_2) - F_{CV}(X_1 - X_2)$

B.M. at point D = $(F_{BA}\text{Cos } 20° + F_{WA})(1 - X_3) - F_{CV}(X_1 - X_3) + F_{WF}(X_2 - X_3)$

B.M. at point E = $(F_{BA}\text{Cos } 20°+F_{WA})(1 - X_4) - F_{CV}(X_1 - X_4)+F_{WF}(X_2 - X_4)+F_{WD}(X_3 - X_4)$.

The diagrams in Figure 2.3 present moment, shear and bending moment diagrams for the horizontal plane.

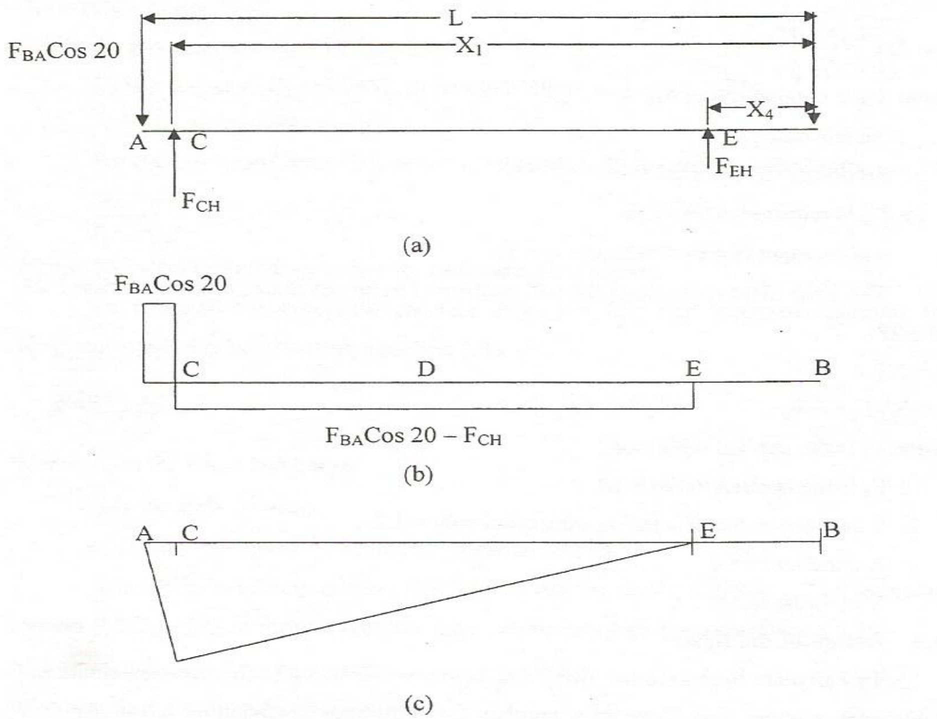

Fig. 2.3: The shear and bending moment diagrams

(a) Horizontal plane load diagram

(b) Shear force diagram

(c) Bending moment diagram

In Figure 2.3b, shear force (S.F.) at point A up to C = F_{BA}Cos 20°

S.F. at point C to E = F_{BA}Cos 20° – F_{CH}

In Figure 2.3c, the bending moment (B.M.) at C = F_{BA}Sin 20° (1 – X_1)

B.M. at E = F_{BA}Sin 20° (1 – X_4) – F_{CH}(X_1 – X_4)

In this way, the vector sum of the maximum bending moments at the points at which it is maximum in the vertical and horizontal planes is then calculated.

For the bearings selection, they are selected based on the shaft diameter and catalog radial rating. The catalog radial rating is calculated using equation 2.26.

$$F_r = F_D\left[\frac{L_D N_D}{L_R N_R}\right]^{\frac{1}{a}} \quad 2.26$$

Where, L_R is catalog radial life

N_R is catalog

F_D is equivalent radial load

L_D is required design life

a is constant (for ball bearings, a = 3)

The equivalent radial load for ball bearings can be calculated using equation 2.27 and 2.28.

$$F_D = VF_r \quad 2.27$$

$$F_D = XVF_r + YF_a \quad 2.28$$

Where, F_r is the applied radial load

F_a is the applied thrust load

V is rotation factor – for rotating outer ring = 1.2

X is radial factor

Y is thrust factor

2.1.3 Design of the Drier

To calculate the minimum fluidizing velocity, we first and foremost calculate the Archimedes number and Reynold's number for minimum fluidization. And for low pressure process, Archimedes number is given by equation 2.29. Source (Rogers and

Mayhew, 1981).

$$A_r = \rho_{air} \frac{g d_p^3 \rho_p - \rho_{air}}{\nu} \qquad 2.29$$

Where, ρ_{air} is the density of air (0.946 kg/m^3)

ρ_p is the bulk density of paddy

g is the acceleration due to gravity

ν is the kinematic viscosity of air

The average length of long grain variety paddy (blue honment) type according to Champ et al (1995) is taken as 9.86 x 10^{-3}m and the bulk density of the same variety at 12 – 18% moisture content as 600.55kg/m^3. The Reynold's number for minimum fluidization is calculated using equation 2.30.

$$Re = (C_1 + C_2A_r)^{1/2} - C_1 \qquad 2.30$$

Where, C_1 is a constant, C_1 = (27.2 – 33.4)

C_2 is a constant, C_2 = 0.0408, Source☹William and Adel, 1989)

A_r is the Archimedes number.

For the minimum fluidizing velocity, equation 2.31 is used for calculation

$$U_{mf} = \frac{Re \times \nu}{d_p \times \rho_{air}} \qquad 2.31$$

Where, Re is the Reynold's number for minimum fluidization

To estimate the minimum pressure drop, we first and foremost calculate the minimum fluidizing height using equation 2.32.

$$L_{mf} = \frac{L_1(1-\varepsilon_1)}{1-\varepsilon_{mf}} \qquad 2.32$$

Where, L_1 is the static bed height

ε_1 is the static voidage

$\varepsilon_{m.f}$ is the voidage at minimum fluidization height

According to Champ et al (1995) ε_1 = 0.496 for paddy with 14 – 22% moisture content (M.C.). The minimum pressure drop can be calculated using equation 2.33.

$$\Delta P_{mf} = (\rho_p - \rho_{air})g(1 - \varepsilon_{mf})L_{mf} + \rho g L_{mf} \qquad 2.33$$

Where, ρ_p is the bulk density of paddy

ε_{mf} is the voidage at minimum at minimum fluidizing height

L_{mf} is the minimum fluidizing height

The design of the distributed plate consists mainly of calculating the diameter of the orifice and the number of the orifice per square meter of the plate. To calculate this, we first and foremost calculate the Reynold's number (Re) for the flow approaching the plate to get orifice coefficient (C_d) using equation 2.34.

$$Re = \frac{d_t \times \rho_{air} \times U_{mf}}{\nu} \qquad 2.34$$

Where, d_t is the diameter of the plate

U_{mf} is the minimum fluidizing velocity

The velocity of gas through the orifice can be calculated using equation 2.35.

$$U_{or} = C_d \sqrt{\frac{2g\Delta P_d}{\rho}} \qquad 2.35$$

Where, C_d is the orifice coefficient

ΔP_d is the minimum pressure drop

The superficial velocity of the gas can also be calculated using equation 2.36

$$U_o = \frac{\pi d_{or} U_{or} N_{or}}{4} \qquad 2.36$$

Where, d_{or} is the diameter of the orifice

N_{or} is the number of orifice per square meter

2.2.0 Result of Design Calculations

Using the analysed empirical formulae to calculate some major parameters, the result of the calculations are presented in Table 2.1.

3.0 Testing, Results and Discussion

3.1 Testing Results

3.1.1 Result of the First Test

Milling rate = 1.32g/s

Total milling recovery = 16.85% at moisture content of 8.77%

Coefficient of husking = 0.80

3.1.2 Results of the Second Test

Milling rate = 1.35g/s

Total milling recovery = 17.33% at moisture content of 8.77%

Coefficient of husking = 0.89

Table 2.1: Result of the Design Calculation

Parameter	Symbol	Unit	Value
Mean Torque	T_m	Nm	24.57
Output pulley speed	n_2	rev/min	387.261
Centrifugal force on the belt	F_c	N	5.81
Angle of twist	α	rad	0.171
Angle of contact	θ_1	rad	2.80
Tension on the tight side of the belt	F_1	N	946.323
Tension on the slack side of the belt	F_2	N	127.323
Net driving force on drive pulley	F_{NA}	N	819
Bending force on the drive pulley	F_{BA}	N	1638
Torque on the driven pulley	T_2	Nm	94.185
Net driving force on pulley B	F_{NB}	N	5.651
Bending force on pulley B	F_{BB}	N	11.302
Reaction at the bearing support C in the vertical plane	F_{CV}	N	2027.205
Reaction at the bearing support E in the vertical plane	F_{EV}	N	688.497
Reaction at the bearing support E in the horizontal plane	F_{EN}	N	128.268
Maximum bending moment in the in the horizontal plane		Nm	214.03
Minimum bending moment in the in the horizontal plane		Nm	76.191
Resultant maximum bending moment		Nm	227.162
Minimum shaft diameter	d	mm	30
Minimum fluidizing velocity	U_{mf}	m/s	1.51
Minimum fluidizing height	L_{mf}	m	0.114
Minimum pressure drop	ΔP_{mf}	Pa	282.694
Diameter of the orifice	d_{or}	mm	0.5
Number of orifice per square centimeter	N_{or}	No	1

3.1.2 Results of the Third Test

Milling rate = 1.28g/s

Total milling recovery = 85.30% at moisture content of 18.75%

Coefficient of husking = 0.85

3.2 Discussion of the Results

The milling rate which is the amount of the grain milled per the milling time ranges between 1.28g/s and 1.35g/s for the three tests and it was more constant than other tested parameters. The total milling recovery for the first and the second tests were very low and stood at 16.85% and 17.33% respectively. This is due to very low moisture content which stood at 8.77% for both of the tests. This also resulted to high amount of broken rice in the first and second tests. When we correlate this with the third test where the moisture content of the paddy before milling is 18.75%, only 14.7% of the rice were broken given a very high total milling recovery of 85.3%.

Coefficient of husking for the three tests is very high above 0.80, however that of the second test is the highest which is 0.89, followed by the third test which stands at 0.85 and lastly the first test which is about 0.80. The optimum moisture content in the three tests is 18.75% before milling because it gives the highest milling recovery of 85.3%.

Conclusion

Based on the results obtained from performance tests of the integrated rice milling machine, the milling rate ranges between 1.28g/s and 1.35g/s. Low moisture content of paddy led to low total milling recovery. Moisture content of paddy of paddy not less than 18.75 results in low rice breakage and high total milling recovery. The coefficient of husking is as high as 0.89.

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