Design And Build A Double Roll Type Pecan Breaking Machine With A Power Of 0.5 Hp

Arion Marslino Simanjuntak[1*], Sahat [2] , Winfrontstein Naibaho [3]

[1] Teknik Mesin , Fakultas Teknik Dan Pengelolaan Sumberdaya Perairan ,
Universitas HKBP Nommensen Pematangsiantar **,** Sumatera Utara Indonesia.
[2,3] Manajemen Pengelolaan Sumberdaya Perairan, Fakultas Teknik Dan Pengelolaan Sumberdaya Perairan
Universitas Sumatera Utara, Medan, North Sumatra 20155, Indonesia.
[*]Corresponding Author:
Email: winfrontsteinnaibaho@uhnp.ac.id

Abstract.
Pecan is one of the spices that become an ingredient in the process of the food and cosmetic industry. The pecan separator machine is made to facilitate and speed up the process of breaking the pecan fruit in order to increase work efficiency in the hope that the machine can achieve high efficiency in the form of good fruit yields. The purpose of this pecan fruit separator machine is to help facilitate the community in processing pecan fruit to the maximum of its production, and the community can also increase the processing productivity of pecan fruit, which can improve the community's economy. The research method applied in this study is an experimental method, namely the initial stage of designing a double roll type pecan breaking machine using Auto CAD. Then build the machine in the workshop based on the design that has been done before. The stages passed are making design drawings in Auto CAD and analyzing the components used in the pecan breaking machine. Such components are the frame, inlet funnel, shaft, breaker roller, bearing and outlet funnel. From the results of the analysis, the motor power is 0.5 HP with a rotation of 1400 rpm. The belt length used from the calculation of the pulley size on the 3" machine and the pulley on the 10 breaking roll is 42". For breaking rollers the outer diameter of the roller = 120 mm, the inner diameter of the roller = 100 mm, and the length of the roller = 190 mm. The shaft on the drive motor is 24 mm in diameter. The shaft isestimated to be of S30C carbon steel with tensile strength = 48 kg/mm².

Keywords: *Pecan; Breaker; amd Capacity.*

I. INTRODUCTION

Non-timber forest products such as pecans (Aleurutes moluccana wild.) are widely spread in Indonesia, have considerable potential in their use. This plant is useful as a cooking spice, traditional medicine, and pharmaceutical ingredient and part of the trunk of the pecan tree is used as a pulp and matchmaking agent, can before the process of breaking the pecan bark is done manually [1]. Manual solving has many disadvantages, namely low solving capacity and many pecans whose quality is low after being solved. This manual breakdown can only solve 10 to 15 kg of pecans in a day and the percentage of pecan cores that are still round after breaking the shell with this method is very small, namely 40% to 60%, thus reducing the selling value of pecans, therefore a pecan breaking machine is made to facilitate and speed up the process in order to improve work efficiency in the hope that the machine can achieve high efficiency in the form of perfect fruit yields and is well separated from the shell.

The method of breaking pecans starts from a manual method which is then applied into a machine with almost the same motion constantly and continuously. There are various methods used to solve pecans with a large capacity but with good results.This pecan fruit breaking machine has several supporting components. The components are in the form of an engine frame, an entry funnel, a shaft, a pecan breaking roll, and an outlet funnel. These components have their own functions. If one of the functions of the component is not fulfilled, it will result in the work of the pecan fruit breaking machine itself. Pecan breaking tools are one of the best alternative solutions to make it easier for farmers to do their work, so that time and energy are not confiscated, with a fairly profitable selling value. So the researcher took the research title "**Design and Build a Double Roll Type Pecan Breaking Machine With a Power of 0.5 Hp"**

1.2 Problem Formulation

From the background of the problem above, the formulation of the problem is taken, namely how

1. How to design a Double Roll Type Pecan Breaking Machine With 0.5 Power.
2. How to build a Double Roll Type Pecan Breaking Machine With a Power of 0.5 Hp.

1.3 Limitations of the issue

The limitation of the problem in this study is the design of a pecan seed breaking machine with a double roll type with a power of 0.5 hp.

1.4 Hypothesis

Based on the description above, to direct the course of this study, a hypothesis was proposed, namely the researcher suspected that the design of a pecan seed breaking machine with a double roll type with a power of 0.5 Hp would affect the efficiency of the engine shaft rotation speed and production results.

1.5 Research Objectives

The purpose of the study is

1. Knowing the design of a double roll type pecan breaking machine with a power of 0.5 hp.
2. Knowing the build of a Double Roll Type Pecan Breaking Machine With a Power of 0.5 Hp.

1.6 Research Benefits

The benefits of this research are:

1. Make it easier for people to break pecans with a double roll type
2. The process is faster and easier than manually.
3. Get more satisfactory results than manually.
4. Helping to facilitate the community in processing pecan fruit with the maximum of its production
5. Assist the community in increasing the productivity of processing pecan fruit, which can improve the community's economy.

II. METHODS

This research was conducted at the Mechanical Engineering Laboratory of the University of HKBP Nommensen Pematangsiantar on Jalan Sangnawaluh No.4 District. East Siantar. The length of the research planned is 6 months.

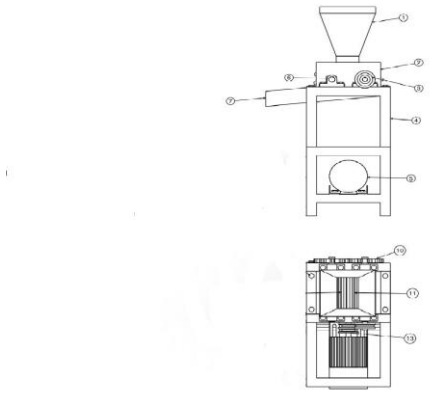
2.1 Tools and Materials

3.2.1 Tools

The tools used in this research:

1. Plat
2. Bearing
3. Pulley
4. Welding Elektrode
5. AC Motor

After assembling the machine :paddle wheel engine components with solar cells, it will look like Figure 3.1. The schematic of this paddle wheel machine with solar cell clearly illustrates the process in the growth of microalgae.



Description:

1. Hopper
2. Casing
8. Gear
9. Main breaking shaft

3. breaker pulley
4. SKeleton
5. Electric motor
6. Bearing seated
7. Funnel out
10. Shaft breaking support
11. Electric motor pulley
12. Bolts and nuts

III. RESULT AND DISCUSSION

4.1 Capacity

With the data of the size of the candlenut, the diameter of the roller, and L = the length of the roller to be made =190 mm

Machine Theoretical Capacity:

1) Roll Area $= \pi . d . l = \pi . 10{,}2 . 19$
$= 609\ cm^2$

2) D = Candlenut diameter $= \frac{\pi . (3\ cm)^2}{4}$
$A = 7{,}06\ cm^2$ / biji

3) Pecan Mass In One Round
$609\ cm^2$ / (7,06 cm^2/biji) $\times$ 10gr/biji $= 863$ gram

4) Machine Capacity In 1 Round If The Pecan Directly Peeled = 863 gram /round

Real Capacity:

$\frac{1\ Kg}{28\ S} = \frac{1\ Kg}{28\ S} \cdot \frac{60\ s}{menit} = 2{,}14\ kg/manit$

2,14 kg/menit $= \frac{2{,}14\ kg}{menit} \cdot \frac{60\ menit}{jam} = 128{,}4\ kg/jam$

4.2 Test Results

This test is carried out by inserting 1 (kg) candlenut into the machine funnel with a roll distance of 1.5 cm. The results of the test were weighed by weight of candlenut which was broken whole, broken and not broken. Then the results are compiled data in tabular form.

Table of the results of testing the candlenut crusher machine using a 1.5 cm roller distance:

Jarak roll	Waktu	Pecah Utuh	Belah dua	Inti Hancur	Biji lolos	cangkang
1,5 cm	28 Detik	10 gr	120 gr	240 gr	70 gr	560

4.3 Drive Motor Power

The motor is the center of the movement in the whole system, therefore it must be considered and calculated carefully and correctly so that the system that we design can run as we expect.

It is known that the electro motor power = 375 Watt

It is known that the rotation of the electro motor = 1400 rpm

Fig 4.1. Motor Listrik

4.4 Transmission And Pulley System

The transmission system on the candlenut breaker is by pulley, with a motor rotation of 1400 rpm. The data on the machine designed:

1. Drive motor pulley 3" (76.6 mm)
2. Pulley on screw press 10" (254.4 mm)

Each pulley is as follows:$N = n_1 \times \frac{d1}{d2}$

Where:

d1 = diameter of the drive pulley

n1 = rotation of the drive pulley

d2 = diameter of driven pulley

Pulley rotation on the screw press is:

$N = n_1 \times \frac{d1}{d2}$

$= 1400 \times \frac{76,6}{254,4}$

$= 421$ rpm

The design axis distance is taken 2 times the diameter of the large pulley, then

On Breaking Rollers

$C_{rencana} = 2 \times 243,4 = 486,8$ mm

The design belt length (L) can be calculated as follows:

On Breaking Rollers

$$L = 2 \times 486,8 + \frac{\pi}{2}(243,4 + 76,6) + \frac{1}{4\ X\ 486,8}(243,4 - 76,6)^2 = 1490,53 \text{ mm}$$

From the table in Appendix 1, the standard belt length can be selected as 26 inches and 59 inches, so the axis distance of the shaft can be calculated as follows:

On Breaking Rollers$C = \frac{4020,4 + \sqrt{4020,4^2 + 8(243,4 - 76,6)^2}}{8} = 1008,54$ mm

Where to:

Breaking Roller $b = 2 \times 1490,53 - 3,14(243,4 + 76,6) = 4020,4$ mm

Menurut Sularso Elemen Mesin

$$L - \frac{Dp - dp}{2} \geq C\ ,\ 1490,53 - \frac{243,4 - 76,6}{2} \geq 1008,54\ ,\ \text{baik } C\ \frac{Dk + dk}{2}\ ,$$

4.5 Breaking Roller Dimensions

The crushing roller is designed based on the existing capacity so that the number of teeth used to break is sufficient for 10 pieces.

Breaking roller design:

D = roller diameter = 102 mm

d = roller inner diameter 100 mm

l = roller length 190 mm

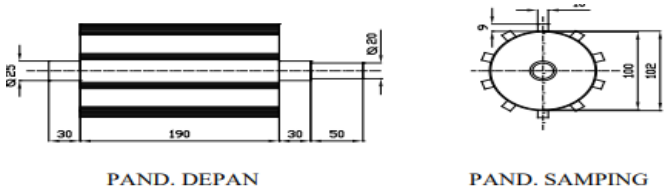


Fig 4.3. Dimensi rol pemecah

So that the mass of the crushing roller is:

$m = 7850.[0,25\pi .(0,102^2 - 0,1^2)0,19]$

$m = 0,47$ kg

For shatter teeth are designed:

Length = 190 mm

Width = 1cm

Thickness = 1cm

So the mass of 10 teeth is:

m = 10.7850.(0,01.0,01.0,19)

m = 1,49 kg

Thus, the total mass of the crushing roller is:

= 0,47 kg + 1,49 kg

= 1,96 kg

4.6 Shaft

4.6.1 Shaft Strength Analysis on Motor

The pros on the drive motor are 24 mm in diameter. then the shaft is estimated from S30C steel With tensile strength (σ B) = 48 kg/mm2, then a **is:**

$\sigma_a = \frac{\sigma_B}{Sf1 \; x \; Sf2} = \frac{48}{6 \; x \; 2}$ = 4 kg/mm^2, τ_a = 0,5 x 4 = 2 kg/mm^2

Torsi (kg.mm) adalah :

T = 9,74 x $10^5 \frac{0,375}{1400}$ = 260,89 kg.mm

Tegangan geser yang timbul :

$\tau = \frac{5,1.T}{d_s^3} = \frac{5,1 \; X \; 260,89}{24^3}$ = 0,096 kg/mm^2

Jadi dapat dikatakan bahwa konstruksi aman karena $\tau_a > \tau$

Ta = $\frac{Tb}{f1 \times f2}$ $[N/\text{mm}^2]$ …. (sularso dan Kiyokatsu suga, elemen mesin hal 8)

where :

Ta = Permissible shear stress $[N/mm^2]$

Tb = Allowable tensile stress 315 G 450S = 62(kg/mm^2)

∫f1 = Mechanical safety factor k=6 (diambil dari S-C)

∫f2 = factor of safety due to the influence of stress concentration and stress strength and surface strength

Ta = $\frac{62 \; kg/mm^2}{6 \; X \; 2}$

= 5,17 N/mm^2

4.7 Bearing Dimension

In this machine the bearings used are rolling bearings. Rolling bearings have the advantage of very low rolling friction when compared to sliding bearings.

If it is known that the radial force from the shaft is 260.89 (N). The bearing shear moment can be determined as follows:

M_t = F.f.(D/2)

Dimana : Mt = Bearing shear moment (N.mm)

F = Radial force (N)

F = Bearing shear coefficient = 0.0015 for single ball bearing

D = Shaft diameter (mm)Maka :

M_t = 260,89 . 0,0015. (24/2)

M_t = 4,69 (N.mm)

Due to the frictional force that arises it will cause some of the power to be lost. Then the amount of power lost is:

P_{loss} = M_t .N.(2π / 60)

Where : P_{loss} = Daya hilang (Watt)

M_t = Momen geser bantalan (N.mm)

N = Putaran poros (rpm)

so : P_{loss} = 4,69 . 1400. (2π / 60)

P_{loss} = 687,58 (Watt)

4.8 Cost Analysis

The following are the prices of engineering materials needed in the design process:

Table 5. Details of engineering material costs

No	Nama Bahan	Ukuran [mm]	Harga Satuan	Unit	Biaya Pengantaran	Harga Total
1	Profil L	40x40x3	125.000	2	30.000	160.000
2	Plat	2400x1200x2	500.000	1		500.000
5	Motor Listrik	0,5 Hp	500.000	1	20.000	500.000
						20.000
	Besi kotak	20x20	70.000	2	10.000	140.000
						10.000
7	Bantalan	NoP205	40.000	2		80.000
8	Poros	380x90	290.000	1		290.000
		325x90	280.000	1	20.000	290.000
						290.000
9	Pulley	10 inch	100.000	1		100.000
		3 inch	50.000	1		50.000
						20.000
11	Elektroda Las	2,6	23.000	2		46.00
12	Mata Gerinda Potong		10.000	9		30.000
13	Mata Gerinda Kasar		8.000	3	10.000	24.000
						10.000
14	Sabuk	A42	42.000	1	20.000	42.000
15	Baut		3.000	30		90.000
						20.000
16	Cat minyak		47.000	1		47.000
16	Drat	Drat M19	78.000	2		156.000
18	Upah Pekerja	60.000 x 3 hari				180.000
Total						2.655.000

IV. CONCLUSION

Based on the calculations and planning carried out, the conclusions are as follows:

1. Planning specifications
   a. Material used: Pecan fruit
   b. Machine capacity: 128.6 kg/hour
   c. Transmission system: Belt and pulley
2. Tool construction
   a. Motor power: 0.5 Hp
   b. Drive motor rotation: 1400 Rpm
   c. Crew press shaft rotation: 421 Rpm
3. Transmission system
   a. Transmission system: belt and pulley
   b. Pulley size: 10” on the motor shaft: 3” on motor shaft
   c. Belt length: 42” split roller shaft
4. Shaft and bearing
   a. Shaft diameter of crushing roller 25 mm
   b. Shaft bearing Rolling bearing NoP205
5. The results of the candlenut crushing machine test using a roller distance of 1.5 cm

V. ACKNOWLEDGMENTS

The authors would like to thank the research members, the Dean of the Faculty of Engineering and Water Resources Management and all fellow lecturers of the mechanical engineering study program who have assisted in the research published in this journal.

REFERENCES

[1] R. Andriani, "DOUBLE ROLL PADA VARIASI PERLAKUAN LAMA THE PERFOMANCE TEST OF DOUBLE ROLL TYPE CANDLENUT SHELLING MACHINES AT VARIOUS OF," 2020.

[2] F. Sutrisno and A. I. Sinaga, "Analisa uji produktivitas kerja mesin pemecah buah kemiri dengan tumbukan tenaga pegas kapasitas 60 kg / jam," ***J. Ilm. "MEKANIK"*** *Tek. Mesin ITM*, vol. 4, no. 2, pp. 70–76, 2018, [Online]. Available: https://jurnal.mesin.itm.ac.id/index.php/jm/article/view/89

[3] M. Sari, D. Ibrahim, E. Wardiana, B. Penelitian, T. Rempah, and T. Industri, "PENAMPILAN DAN MANFAAT LIMA SPESIES KEMIRI," pp. 67–72.

[4] K. H., K. M.H., and K. M., "Aleurites moluccana (L.) Willd.: ekologi, silvikultur dan produktivitas," *Aleurites moluccana Willd. Ekol. silvikultur dan Produkt.*, 2011, doi: 10.17528/cifor/003480.

[5] K. Akhir and A. Laedan, "Mesin Pemecah Kemiri Dengan Kapasitas 20 Kg Per Jam Industri Program Diploma – Iv Fakultas Teknik Universitas Sumatera Utara," 2010.

[6] P. Studi, T. Pertanian, and J. T. Pertanian, "Rancang Bangun Alat Pemecah Kulit Kemiri Dengan Modifikasi Gigi Pada Posisi Horizontal Sistem Rotary 450 Rpm," 2019.

[7] R. Sinaga, "BIJI KEMIRI (Aleurites moluccana Willd .) SISTEM RIPLE MILL," 2017.

[8] R. P. Adolf, "Analisis Alat Pemecah Kulit Kemiri Dengan Modifikasi Gigi Pada Posisihorizontalsistem Rotary450 Rpm," 2019, [Online]. Available: http://repository.ummat.ac.id/id/eprint/455

[9] F. Abdarrasyid, "Mesin Pemecah Cangkang Kemiri Tipe Double Roll the Effects of Rotation Speed and Roll Distance To the Seed Candlenut Crack Percentage on Candlenut Shell Cracker Machine Type Double," 2019.

[10] J. Jasman, F. Febriwanto, and Z. A. Putra, "Rancang bangun mesin pemecah buah kemiri dengan kapasitas 20 kg /jam (perancangan)," ***J. Pendidik. Tek. Mesin,*** vol. 1, no. 1, pp. 1–13, 2018, [Online]. Available: https://www.researchgate.net/publication/328475663_RANCANG_BANGUN_MESIN_PEMECAH_BUAH_KEMIRI_DENGAN_KAPASITAS_20_KGJAM_PERANCANGAN_DESIGNING_UPPER_CRUSHER_FRUIT_MACHINE_WITH_20_KG_HOUR_CAPACITY_DESIGNING

[11] Sularso and K. Suga, *Dasar Perencanaan Dan Pemilihan Elemen Mesin*, Kesembilan. jakarta: PT Pradnya Pramita, 2018.

[12] J. T. Mesin, F. T. Industri, and U. Trisakti, "Kagoshimaken kōritsu shō chūgakkō kyōshokuin chōki jinji idō no hyōjun.," pp. 1–11, 1974.

[13] H. Hadiyanto, M. M. Azimatun Nur, and G. D. Hartanto, "Cultivation of chlorella sp. As biofuel sources in palm oil mill effluent (POME)," ***Int. J. Renew. Energy Dev.***, vol. 1, no. 2, pp. 45–49, 2012, doi: 10.14710/ijred.1.2.45-49.

[14] Y. Li, Q. Zhang, Z. Wang, X. Wu, and W. Cong, "Evaluation of power consumption of paddle wheel in an open raceway pond," *Bioprocess Biosyst. Eng.*, vol. 37, no. 7, pp. 1325–1336, 2014, doi: 10.1007/s00449-013-1103-3.

[15] S. S. Sawant, S. N. Gosavi, H. P. Khadamkar, C. S. Mathpati, R. Pandit, and A. M. Lali, "Energy efficient design of high depth raceway pond using computational fluid dynamics," *Renew. Energy*, vol. 133, pp. 528–537, 2019, doi: 10.1016/j.renene.2018.10.016.

[16] R. Slade and A. Bauen, "Micro-algae cultivation for biofuels: Cost, energy balance, environmental impacts and future prospects," *Biomass and Bioenergy*, vol. 53, no. 0, pp. 29–38, 2013, doi: 10.1016/j.biombioe.2012.12.019.

[17] S. Sivakaminathan *et al.*, "Light guide systems enhance microalgae production efficiency in outdoor high rate ponds," *Algal Res.*, vol. 47, no. September 2019, p. 101846, 2020, doi: 10.1016/j.algal.2020.101846.

[18] S. Bahri, R. Setiawan, W. Hermawan, and M. Yunior, "Perkembangan Desain dan Kinerja Aerator Tipe Kincir," ***J. Keteknikan Pertan.***, vol. 2, no. 1, p. 21685, 2014, doi: 10.19028/jtep.02.1.

[19] Y. Chisti, "Biodiesel from microalgae," *Biotechnol. Adv.*, vol. 25, no. 3, pp. 294–306, 2007, doi: 10.1016/j.biotechadv.2007.02.001.

[20] D. Kumar and S. Sarkar, "A review on the technology, performance, design optimization, reliability, techno-economics and environmental impacts of hydrokinetic energy conversion systems," *Renew. Sustain. Energy Rev.*, vol. 58, pp. 796–813, 2016, doi: 10.1016/j.rser.2015.12.247.