Study and design of a coconut stripper

Edmond Claude VODOUNNOU[1]*, Guy Clarence SEMASSOU[1], Roger H. AHOUANSOU[1]

[1]Laboratory of Energetics and Applied Mechanics (LEMA), University of Abomey-Calavi, 01 BP 2009 Cotonou, Benin

Abstract

The predominance of the primary sector is noticeable in the economy of most African countries, particularly in Benin. Thus, to make this sector a vector of wealth and development, it is imperative to think about the development of technologies to improve the efficiency of processing and product quality. The mechanization of this sector is therefore necessary to make it more competitive. In Benin we have a lot of products which are made with the coconut. This sector has been relegated to a non-priority plan, whereas the coconut palm is a plant with a very considerable value like the oil palm, of which all the derived products are usable. It seems necessary to pass to the revival of this sector to help to stabilize the incomes of the small producers and traders in particular and to improve the commercial balance of the country in general. The maximization of the output of this sector directs towards the formula of Lavoisier: "nothing is lost, nothing is created, everything is transformed". The application of this formula allows us to transform the by-products of the said product. Currently, the de-shearing of the coconut is done manually with a cutter or with a pointed object. This way of making is too painful for producers who have large exploitations of coconut trees. In order to make this sector more profitable, our end-of-training work focuses on the study and design of a coconut de-sheathing machine intended to accompany coconut producers by reducing their pains and increasing their productivity in order to enhance the transformation of the said product.

Keywords: Coconut, cut-cut or with a sharp object, make this sector profitable, coconut breaking

*Corresponding author: Edmond Claude VODOUNNOU
e-mail: edmonvod@yahoo.fr

1-Introduction

The mechanization of agriculture has been meteoric since the end of the Second World War. Modernization, on the one hand, and the race for innovation and gigantism, on the other, have favored the appearance of a multitude of machines whose main and essential objective was to increase the productivity, profitability and economic efficiency of farms [1]. Thus, the agricultural sector is a fundamental pillar of rural development. Moreover, agriculture is a key sector for the successful entry of some developing countries into globalization. Indeed, it is very difficult to imagine a developed agricultural sector without equipment. Crop yields depend on many factors, but agricultural mechanization is the most important element. The modernization of agriculture necessarily involves the rational mechanization of the various food processing processes, with priority given to arduous work [2]. In Benin, we have quite a lot of products which are made from the coconut among which we can quote: the bag, rope, soap, snacks, oil and herbal tea. This sector has been relegated to second place after corn, cotton, oil palm, cowpea, etc. Whereas the coconut palm is a plant with a very considerable value, as is the oil palm, and all its by-products are usable. It seems necessary to revive this sector to help stabilize the income of small producers and traders in particular and improve the country's trade balance in general. The maximization of the output of this sector directs towards the formula of Lavoisier: "nothing is lost, nothing is created, everything is transformed". The application of this formula allows us to transform the by-products of the said product. Currently, the de-shearing of the coconut is done manually with a cutter or with a sharp object. This way of making is too painful for kinds which have big exploitations of coconut. Attempts to mechanize the debudding operation have been made and are the subject of various patents: the pusher system has most often been exploited, such as patent registration reference No. 7607792, a system by defibration has been exploited patent No. PD

6789/78. A machine has also been designed by Jean-Xavier and Gilles Durand, the portable delimbing pile by cirad as well as the delimber developed by Jean Colas.
In order to make this sector more profitable, we decided to study and design a coconut de-sheathing machine for our end-of-course work.

2- Materials and methods

2.1 Presentation of the equipment

Figure 1 is a 3D representation of the coconut de-skinner**.**

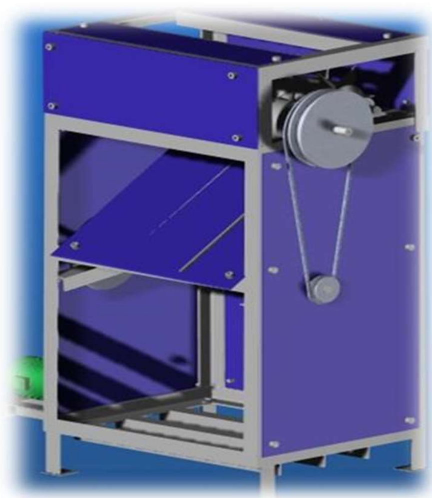

Figure 1: The proposed coconut stripper

The main components of the coconut unscrambler are the motor, unscrambler unit, frame, drive shafts, pulleys and bearing blocks.

2.1.1 The unstuffing unit

The de-sheathing unit consists of two toothed drums, which are considered the main mechanism of the machine. These components, which rotate in opposite directions to each other, are the ones that grip and tear the coconut husks in the de-shearing process.

These drums are made of steel. Each drum is carried by two transmission shafts supported by a frame and guided by two bearings. On the drums, steel teeth of conical shape are fixed. The power transmission is ensured by gearing.

2.1.2 Transmission shafts for gear drums

The transmission shafts support the gear drums and allow them to rotate. On the machine there are three transmission shafts, two of which support the drums of the untwisting unit and the other is replaced by a speed reducer.

2.1.3 The drive motor of the unloading unit

It drives the unloading unit by means of the pulley - belt and gear transmission.

2.1.4 The machine frame

The frame has been designed to support all the components of the machine. It has been manufactured with angle irons L with equal wings: 40x5

2.2 Operating principle

Figure 2 is the kinematic representation of the coconut de-skinner. It shows the operating principle of the equipment. When the motor is started, it transmits its rotational movement to the primary shaft through two pulley-belt connections. The primary shaft in turn transmits a rotational movement to the secondary shaft by means of a spur gear, which makes the two toothed drums turn in opposite directions.

Once the motor is started, the nut is placed on the drums; the opposite rotation of the two toothed drums allows the teeth to penetrate easily into the coconut husk and to tear off little by little this husk until the end. To remove the product at the end, the operator uses the coir fibers left after de-stuffing and the coir flocks come out from the bottom. This is how the coconut is de-stuffed.

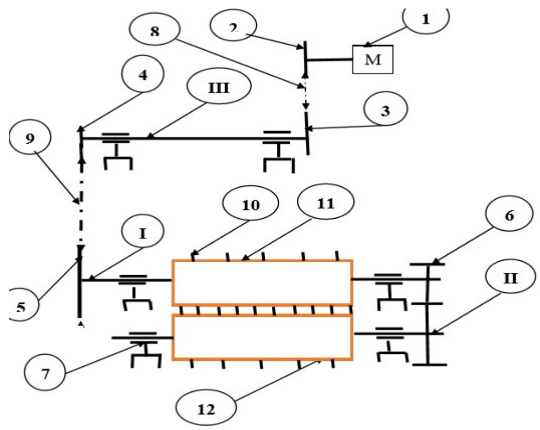

Figure 2: Kinematic chain of the coconut stripper

Legend

1: motor; 2: driving pulley; 3: large intermediate pulley; 4: small intermediate pulley; 5: driven pulley; 6: gear; 7: bearing; 8: belt (a); 9: belt (b); 10: tooth; 11: primary drum; 12: secondary drum; I: primary shaft; II: secondary shaft; III: intermediate shaft

2.3 Modeling

This part is devoted to the identification of mathematical models related to the various components of the dosing machine. These models reflect the physical phenomena that govern the functioning of the components.

2.3.1 Determination of the force required to de-burr a coconut

In order to obtain the breaking force, we conducted an experiment in the workshop. This experiment was done with a dynamometer and a ripe coconut. The experiment consisted in hanging the dynamometer and uncorking the coconut by the pointed end of the dynamometer, which allowed us to determine the mass *m* of the coconut. It should also be noted that this mass may vary depending on the water content of the fluff, the lower the water content is the easier the debudding is.The force required to debud a coconut is then F_1 expressed by :

$$F_1 = \frac{m \times g}{n} \quad (1)$$

With *m* the mass necessary to break the nut, g the force of gravity and *n* the number of teeth of the two drums.

2.3.2 Determination of the force necessary to make the product rotate at the moment of breaking.

The product will rotate provided that the pressure to make it rotate is strictly greater than the pressure to break it.

So $$F' > F_1 \quad (2)$$

2.3.3 Determination of the rotation speed of the toothed drums

In order to determine this speed, a manual test on three (03) coconuts was carried out. The breaking time was timed for each coconut and an average value N1 was chosen. Then we assume that the equipment will work five times faster than the manual unstuffing.

So we have:

$$N = N_1 \, x \, 5 \qquad (3)$$

2.3.4 Determination of the linear velocity of the product

$$V = d_{ext} \pi N \qquad (4)$$

With d_{ext} the outside diameter of the drums and N the speed of the drums.

2.3.5 Determination of the driving power of the product

Let P be the driving power of the product

$$P = F' x V \qquad (5)$$

Where F' is the force required for the product to rotate at the time of unloading and V is the linear speed of the product drive.

2.4 The drive system

The transmission chain is composed of several types of transmissions that transform the power of the motor and determine the speed necessary for the proper functioning of the equipment. It is characterized by two parameters: the total efficiency and the transmission ratio. The type of transmission used here is the pulley-belt transmission and the gear transmission.

2.4.1 Overall efficiency

Its expression is [3]:

$$\eta_{global} = \eta_p^3 \times \eta_c \times \eta_e \qquad (6)$$

With η_p the efficiency per bearing pair, η_e the efficiency of the gear,and

η_c the V-belt efficiency.

2.4.2 The transmission ratio

It is determined by:

$$r = r_1 \times r_2 \qquad (7)$$

$r_1 \; and \; r_2$ being respectively the transmission ratios by pulleys-belts and by gears; with

$$r_1 = \frac{d_3}{d_1} = \frac{N_M}{N} \qquad (8)$$

With d_3 and d_1 respectively the diameters of the driven pulley and the small pulley (driving pulley) in mm ; N_M and N respectively the rotational speeds of the driving pulley (motor) and the driven pulley in tr/min.

And $r_2 = 1$

2.5 Drive shafts

They are essential for the operation of the drive system. The parameters of the drive train are the power, speed and torque exerted on each shaft. Thus:

$$C = \frac{30 \times P}{\pi \times N} \quad (9)$$

2.6 Sizing of the drive belts

The transmission of motion from the motor to the other components will be via V-belts. A belt is characterized by its cross section, pitch length, center distance, winding angle and maximum power rating.

2.6.1 Selection of the belt cross-section

The selection of the belt section depends on the operating power P_s and the speed of the small driving pulley. The expression for the operating power is**:**

$$P_s = K_s \times P_M \quad (10)$$

With K_s representing the service factor, which depends on the type of components, motor and receiver and the daily operating time, and P_M is the motor power. Knowing the values of P_M and N_M, we refer to the graph of transmissible power ranges by belt type to select the type of belt [4].

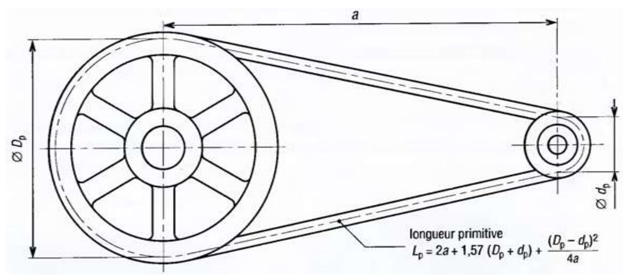

Figure 3: Primitive line of a V-belt [3]

2.6.2 Calculation of the belt center distance

The center distance of the belts is the distance between the axes of two pulleys. In order to determine the theoretical center distance a, its minimum value (a_{min}) and maximum value (a_{max}) must first be determined.

When the ratio $d_b/d_a$is between 1 and 3 (as in the present case), a_{min} and a_{max} are determined by the relations (11) and (13).

With d_a et $d_b$meaning the pitch diameters of the driving pulley and driven pulley, respectively [5]:

$$a_{min} = \frac{d_a + d_b}{2} + d_a \quad (11)$$

$$a_{max} = 3(d_a + d_b) \quad (12)$$

After determining a_{min} and a_{max}, an approximate value for the theoretical center distance is chosen such that $a_{min} \leq a < a_{max}$.

Since the axes of the pulleys are parallel and the belt is not crossed, the theoretical length L_{th} of the belt is written [3] :

$$L_{th} = 2a_{th} + 1{,}57(d_a + d_b) + \frac{(d_b - d_a)}{a_{th}} \times \frac{(d_b - d_a)}{4} \quad (13)$$

After the calculation of L_{th}, the table with indicative pitch lengths of V-belts is used to select a standard length which is an approximation of the calculated theoretical length.

From the standard length, the actual center distance is then calculated as follows:

$$a = a_{th} + \frac{L-L_{th}}{2} \quad (14)$$

2.6.3 Calculation of the basic power P_o

The basic power P_o is a function of the linear speed V and the pitch diameter d_a of the small pulley. V is expressed by:

$$V = \pi d_a N_M \quad (15)$$

Based on the table showing the basic power P_o (in kW) of conventional V-belts, P_o is selected taking into account the calculated V and d_a [3].

2.6.4 Calculation of the winding angle θ

The wrap angle is the angular deviation between the direction of the web and the horizontal. It is determined by the following formula:

$$\theta = 180° - 2\ arcsin \frac{d_b - d_a}{2} \quad (16)$$

2.9.5 Calculation of the admissible power P_a

The admissible power P_a is determined as follows [3]:

$$P_a = P_o \times K_\theta \times K_L \quad (17)$$

With K_θ the correction coefficient as a function of the winding angle θ and K_L the correction coefficient as a function of the length L.

2.6.6 Calculation of the number of belts *n*

The number is determined by the quotient of the operating power and the permissible power.

$$n = \frac{P_s}{P_a} \quad (18)$$

2.7 Pulleys

The system to be designed consists of three pulleys of different diameters. The parameters considered for the pulleys are the weight and the forces that the pulleys exert on the belts.

2.7.1 Weight of the pulleys

The pulleys are made of aluminum alloys. Let be,$\rho = 2{,}7\ kg/dm^3$ the density of the pulleys and P_p the weight of the pulley. Assume that the pulleys are cylindrical without cells of diameter d_p and height equal to the thickness e_p of the pulley. [6] The weight can be determined by the formula:

$$P_p = \frac{\pi \times d_p \times d_p \times e_p}{4} \times \rho \times g \quad (19)$$

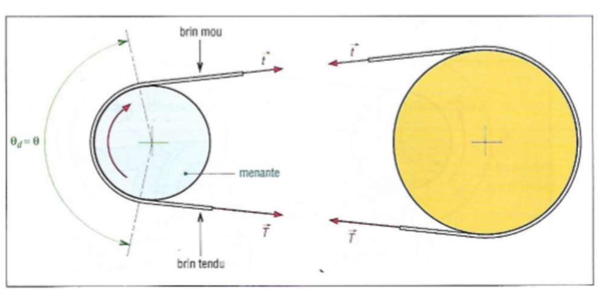

Figure 4: Tension forces [3]

2.7.2 Operating tension forces of the belts

The belt that connects the two pulleys admits a taut strand and a slack strand. Let T be the tension in the taut strand and t the tension in the slack strand. T and t are subject to the following relationships:

$$T - t = \frac{2\times C_M}{D} \tag{20}$$

$$\frac{T}{t} = e^{f\theta} \tag{21}$$

From these two voltages, we obtain the installation voltage

$$T_o = \frac{1}{2}(T + t) \tag{22}$$

The forces exerting the torque F_{py} along the horizontal and F_{pz} along the vertical are then determined such that :

$$F_{py} = 2nT_o \sin\beta \tag{23}$$

$$F_{pz} = 2nT_o \cos\beta \tag{24}$$

With

$$\beta = \arcsin\frac{r_b - r_a}{a} \tag{25}$$

r_b *and* r_a being respectively the radii of the driven pulley and the driving pulley and a, the actual center distance.

2.8 Determination of gear characteristics

2.3.8.1 Determination of the gear diameter d

In order to simplify the data the transmission ratio $r_2 = 1$ is chosen.

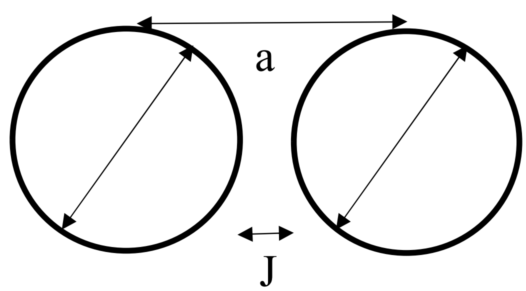

Figure 3: Centre distance between primary and secondary shafts

We have: d = a- J (26)

with

d: the diameter of the drum

J: the clearance between the two drums

a: distance between the drums

2.3.8.2 Determination of the power that the operator must provide to unstuff a coconut.

Let P_O be this power:

$$P_O = \frac{P}{n_e X n_e} \quad (27)$$

With n_e: the efficiency of a spur gear and *P* the driving power of the product.

2.3.8.3 Calculation of the module m of the gear [7]

$$m \geq \sqrt{\frac{F_t}{K \times R_{pe}}} \times 2{,}34 \quad (28)$$

with

F_t , R_{pe} and K

2.3.8.4 Calculation of the number of teeth Z [7]

$$a = \frac{m(Z_1 \times Z_2)}{2} \quad (29)$$

With a the center distance of the drums, *m* the module of the gear, Z_1 and Z_2 respectively the number of teeth of the first and second gear.

2.3.8.5 Calculation of the width b of the teeth [7]

$$b = k \text{ x } m \quad (30)$$

2.3.8.6 Calculation of the tooth thickness e

$$e = \frac{\pi \times m}{2} \quad (31)$$

2.3.8.7 Determination of the weight Pe of the gears [8]

The gears are made of steel of density ρ= 7,85 kg/dm^3

$$P_e = m_e \times g \quad (32)$$

with m_e the mass of a gear and g the force of gravity.

$$m_e = \rho \text{ x } V_e \quad (33)$$

Where

$$V_e = \frac{\pi \times d \times d \times b}{4} \quad (34)$$

With V_e the volume of a gear wheel and *d* the diameter.

2.9 Shaft diameter

During normal operation, the stump grinder shafts are subjected to bending, twisting and shearing. They must therefore simultaneously meet the resistance conditions for the three (3) physical phenomena mentioned below. [9] Therefore:

Resistance to bending

$$\begin{cases} \tau_t \leq R_{pg} \\ d \geq \sqrt[3]{\frac{32M}{\pi \sigma_{adm}}} \end{cases} \quad (35)$$

Shear strength

$$\begin{cases} \sigma_f \leq R_p \\ d \geq \sqrt{\frac{4T_{max}}{\pi R_{pg}}} \end{cases} \quad (36)$$

Resistance to torsion

$$\begin{cases} \tau_c \leq R_{pg} \\ d \geq \sqrt[3]{\frac{16C}{\pi R_{pg}}} \end{cases} \quad (37)$$

With M the maximum bending moment, T_{max} the maximum shear force, C the maximum torque on the shaft and R_{pg} the practical slip resistance of the steel used for the shaft.

After calculating the diameter in all three cases, the standard diameter immediately above the value of the largest of the three is selected.

2.10 Estimation of the cost of the machine

The global cost (C) of the machine will be evaluated according to: the cost (Cm) of the raw materials used; the cost (Cu) of machining the parts; the cost (Cp) of the standard parts and accessories; the cost (Cb) of the design office.

3. Results and analysis

3.1 Results

The results of the coconut analysis as well as the cost of the equipment are recorded in Tables 1,2,3,4,5,6,7,8,9,10,11 and 12.

Table 1: Physical characteristics of the coconut

Variables	Measurements
	Ovoid
Shape	163,4
Polar Diameter (PD) (mm)	140,2
Equatorial Diameter (ED) (mm)	1,17
Ratio DP/DE	963240
Volume ($mm3$)	1160,41
Whole nut mass (g)	600
Coconut fluff mass (g)	126,41
Coconut shell mass (g)	272,25
Mass almond (g)	174,08
Coconut water mass (g)	40,3
Coconut shell thickness (mm)	11,60
Almond thickness (mm)	37

Table 2 : Values of drum characteristics

Characteristics	**Values**
Outside diameter	100 mm
Inside diameter	90 mm
Length of the drums	420 mm
Length of the teeth	30 mm
Number of teeth on a drum	32
Interval between teeth	50 mm

Table 3: Values of the gear characteristics

Characteristics	**Values**
Module *m*	0,5
Number of teeth *Z*	15
Tooth width *b*	3 mm
Tooth thickness *e*	1 mm
Diameter *d*	110 mm
Center distance *a*	110 mm
Play *J*	10 mm
Weight P_e	2,2 N
Rotational speed *N*	15,6 tr/min
Linear speed *V*	0,082 m/s

Table 4: Values of the characteristic parameters of the drive chain

<table>
<tr><th>Components</th><th colspan="4">Parameters</th></tr>
<tr><td rowspan="5">Drive chain</td><td colspan="3">Efficiency</td><td>Transmission ratio</td></tr>
<tr><td>Bearings (η_p)</td><td>Gear (η_e)</td><td>Belts (η_c)</td><td rowspan="4">0,021</td></tr>
<tr><td>0,95</td><td>0,95</td><td>0,96</td></tr>
<tr><td colspan="3">Overall efficiency</td></tr>
<tr><td colspan="3">0,75</td></tr>
</table>

Table 5: Values of the motor characteristic parameters

Components	Parameters	
Motor	Power P_M (W)	Rotation speed N_M (rpm)
	5	15,6

Table 6: Values of the characteristic parameters of the trees

Components	Parameters		
	Power (W)	Rotation speed (rpm)	Torque $(N.m)$
Shaft I	4,51	15,6	2,76
Shaft II	4,3	15,6	2,62

Table 7: Values of the characteristic parameters of the pulleys

Components	Parameters			
Driving pulley and intermediate pulley	Driving pulley	Intermediate pulley	Forces exerting the torque	
	$P_p = 1,94\ N$	$P_p = 25,26\ N$	$F_{py} = 5,35\ N$	$F_{pz} = 39,62\ N$
Intermediate Pulley and driven pulley	Intermediate pulley	Driven pulley	Forces exerting the torque	
	$P_p = 25,26\ N$	$P_p = 48.4\ N$	$F_{py} = 0,97\ N$	$F_{pz} = 15,54\ N$

Table 8: Values of the characteristic parameters of the drive belt

Components	Parameters			
Belt Driving pulley - intermediate pulley	Operating power P_s (in kW)	Linear speed v $(in\ m/s)$	Center distance (in mm)	
	0,108	2,47	$a_{min} = 189,5$	$a_{maxi} = 759$
			Theoretical center distance (mm) $a_{th} = 474$	
			Real center distance (mm) $a = 474,65$	
Belt Intermediate Pulley - driven pulley	Operating power P_s (in kW)	Linear speed v $(in\ m/s)$	Center distance (in mm)	
	0,108	7,46	$a_{min} = 442,5$	$a_{maxi} = 1515$
			Theoretical center distance (mm)	

			$a_{th} = 979$
			Real center distance (mm) $a = 985{,}75$

Table 9: Values of the characteristic power transmission parameters of the belt

Components	Parameters				
Belt Driving pulley - intermediate pulley	Standard length L (mm)	Belt wrap angle θ	Basic power P_o (in kW)	Admissible power P_a (in kW)	Number of belts
	1355	178,43°	1,23	1,19	1
Belt Intermediate Pulley - driven pulley	Standard length L (mm)	Belt wrap angle θ	Basic power P_o (in kW)	Admissible power P_a (in kW)	Number of belts
	2768	178,43°	1,41	1,36	1

Table 10: Values of the characteristic parameters of the drive shafts

Components	Parameters			
	Maximal torque $C\ (in\ N.m)$	Maximal shear force $T_{max}\ (in\ N)$	Maximal bending moment $M\ (in\ N.m)$	Practical slip resistance $R_{pg} (in\ MPa)$
Shaft I	2,76	62,5	6,5	82,25
Shaft II	2,62	40,048	10,69	82,25

Table 11: Values of the characteristic parameters of the drive shafts

Components	Parameters			
	Diameter d (in mm)			Lengh L (in mm)
	Bending strength	*Shear strength*	Resistance to torsion	
Shaft I	$\geq 8{,}24$	≥ 1	$\geq 13{,}07$	640
		Selected diameter (in mm) 15		
Shaft II	$\geq 9{,}53$	$\geq 0{,}8$	$\geq 5{,}12$	640
		Selected diameter (in mm) 15		

Table 12 : Cost of stripper

Cost of materials (in $)	Machining cost (in $)	Cost of the design office (in $)	Total cost (in $)
540.07	18.82	236.64	795.53

3.2 Analysis of the results

3.2.1 Cost Cb of the design office

The cost of the design office is a function of the number of design hours and the hourly cost of the design.

The study of the rolling mill deliberator is developed in eleven (11) days at a rate of six (06) working hours per day, i.e. a total of sixty-six (66) working hours.

The hourly cost of the study is 2000fcfa.

3.2.2 Choice of drum dimensions

For both drums a thick pipe with an outer diameter d_{ext} = 100 mm and an inner diameter d_{int} = 90 mm is chosen. Taking into account the length of the coconut, which is about 300 mm, the length of the drums was set to 420 mm.

3.2.3 Choice of tooth dimensions

The dimensions of the teeth are chosen taking into account the clearance between the two drums and also taking into account the thickness of the coconut husk. For the teeth a conical shape of length 30mm is chosen, this value will allow the teeth to remove the wads without breaking the shell, they are placed on the drums at intervals of 50 mm what makes a total of 64 teeth.

3.2.4 The motor

The desired characteristics for the motor of the dosing machine are the following : P_M = 5 W and regime N_M = 15.6 rpm. For safety reasons (risk of overloading) and because the values found are not within the standard range of motor characteristics, the motor with the next highest power rating is selected. After consulting the catalog of electric motors, the LP 90L electric motor of power P_M = $\mathbf{0}$.09 k$\mathbf{W}$ and regimeN_M = 750 rpm has been selected.

4. Conclusion

The realization of this document, within the framework of our end-of-study work, will make it possible to improve the income of rice production in Benin in a sustainable way by limiting weeding time, to protect workers from the dangers related to manual weeding, to limit labor and to fight against the accelerated use of chemical products for weeding. The manufacture of the proposed equipment should have considerable socio-economic benefits for producers in particular and will open up many business opportunities in general. The State must also play its part by accompanying the realization of the weeder to allow its popularization and the promotion of the local production in our country.

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