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Study and Design of a Dry Palm Kernel Separator

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Abstract

Palm oil and palm kernel oil production in Benin relies primarily on the artisanal sector. Surveys were conducted among rural women in southern Benin to detail each phase of the artisanal production process, from raw material procurement to product marketing. Some practices have not encouraged mechanization, and processing processes that have long remained manual are now poorly mechanized. The present study was carried out with a view to studying and designing a "dry palm kernel separating machine". It is intended to facilitate the separation of hulls and palm kernels in order to optimize the yield of separation, and reducing the pain of the nut processors. The realization of the machine has required the collection of a number of materials and parts among which we can cite: the motor, steel shafts, plates and bearings. It was built according to three systems: a drum, vibrating perforated grids and a conveyor belt to ensure good separation. First of all, there is a first separation at the level of the drum which allows in a first place to recover the not crushed shells. After this first separation, a second one is made on two perforated grids of different diameters. This separation is to collect hulls and broken almonds that could pass and finally, the third separation based on a conveyor belt, allowing to collect round almonds and without hulls. However, it is equipped with a Robbin engine of 1 HP that rotates at a speed of 1500 rpm. The tests and studies on the equipment have allowed the separation of hulls and kernels in the transformation process. These tests have resulted in a separation of 200 Kg of crushed nuts per hour. The machine has been partially designed and tested manually to attest its functionality, which is quite satisfactory.

Keywords: Shell, almond separator, palm kernel oil, crushed nuts, almonds and shells.

Introduction

Being at the heart of development, agriculture is a sector with great potential that presents many challenges[1]. Located in West Africa, Benin is a developing country. With 4.8 million hectares of arable land, of which barely 1 million hectares are exploited by approximately 400,000 farms, Benin's agriculture presents many challenges[2]. Thus, in order to make this sector a vector of wealth and development, it is imperative to consider the development of technologies to improve the efficiency of processing and product quality. However, we had to note that only the fruit of the palm tree is the subject of a commercial exploitation. Indeed, after the harvest of the fruit, and the first transformation into palm oil or red oil, the oilcake or at least the palm nuts are devalued and therefore thrown away[3]. To date, palm oil producers have asked us to find a dry solution for the separation of kernels and shells.

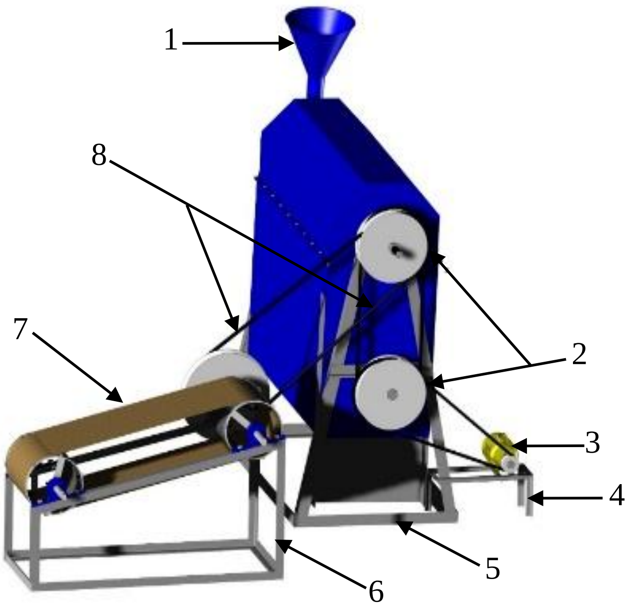
During the last few years, a lot of design work has been done on the design of a palm kernel and shell separator and some of it has even led to their realization. Despite all these studies, palm oil producers are still asking us to find an efficient dry solution for the separation of kernels and shells. The current practice involves either clay baths or manual sorting of the kernels, but these solutions are not very popular, so the vast majority of palm nuts are discarded and not processed[4]. Faced with this situation and to make our contribution to agricultural and artisanal production, we have chosen to study the design and development of an unfinished prototype of a separating machine and dry palm kernel shells that meets the standards of producers and that will lead to a realization in the coming years.

Methodology

Description of the proposed equipment: The study of the design of the separating machine has been conceived and realized partially (prototype) from mechanically welded parts, which will offer ease of manufacture to the various welding shops formed for this purpose. Figure-1 is the proposed equipment in 3D while figure 2 in addition to being a section of the equipment without the belt is in 2D.

The hopper: The hopper is the first part of the machine that allows to convey the product that is the crushed nuts in the drum for the beginning of the dry separation. It is essential and indispensable for the separator.

The drum: Being at the very beginning of the chain, it is mounted on the IV axis. Its role is to ensure a first separation of the product as soon as it is introduced into the machine. Carried by meshes of different spacing, it ensures a first separation, because the not crushed nuts are collected after this first separation of the equipment.



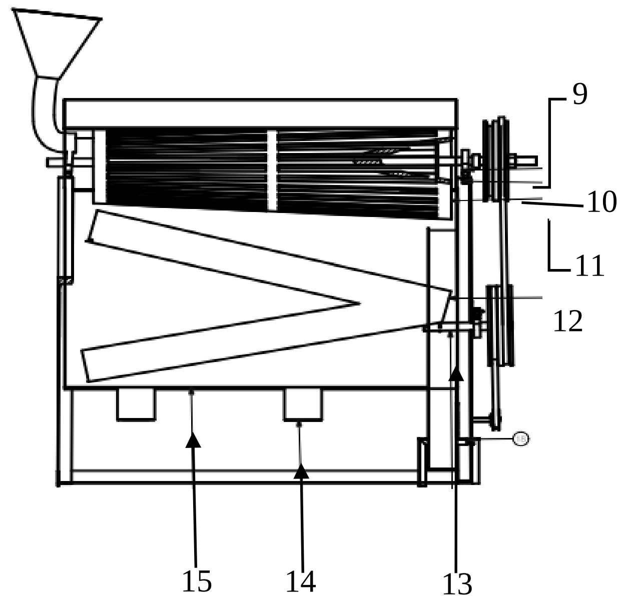


Figure- 1: Proposed separating machine **Figure 2:** Cross-sectional representation of the separating machine.
1: Hopper; 2: Double groove pulleys; 3: Motor; 4: Motor frame; 5: Drum frame; 6: Conveyor belt frame; 7: Burlap bag; 8: Transmission belts; 9: Bearing; 10: Bearing; 11: Rolling drum; 12: Perforated grids; 13: Eccentric; 14: Chute; 15: Steel plate.

Perforated with different diameters and not in the same way, the grid with larger diameter holes (being 12 *mm*) is placed at the top while the one with smaller diameter (being 10 *mm*) is placed below.

The eccentrics: As mentioned in 2.1.3., their role will be to vibrate the two perforated grids. There are four of them, two per grid.

The treadmill: The conveyor belt is mounted on the axes V and VI, as it has two rollers on which the belt is placed ensuring a rotational movement. Its role is the separation of the product in order to have almonds of desired qualities.

The gasoline or thermal engine: It drives the drum by means of a speed reducer without forgetting the belt and pulley transmission system designed for this purpose.

Operating principle of the separator: For the operation of the separating machine, the motor (M) drives the gearbox (2). The reducer in turn, through the belt (3), drives the shaft IV (7) of the drum in a rotational movement. Once the crushed nuts are poured into the drum (8), the almonds and the broken shells fall on the first perforated grid (9) in vibration thanks to the eccentrics (11). In this vibration, the almonds without forgetting the broken shells are conveyed towards the second perforated grid (10). Always in this phenomenon of vibration, the almonds are collected on the conveyor belt with the broken hulls that have passed. On the belt, the almonds roll and are recovered in the collecting tray (19) when the shells retained by the belt are collected by the exit of the shells (17).

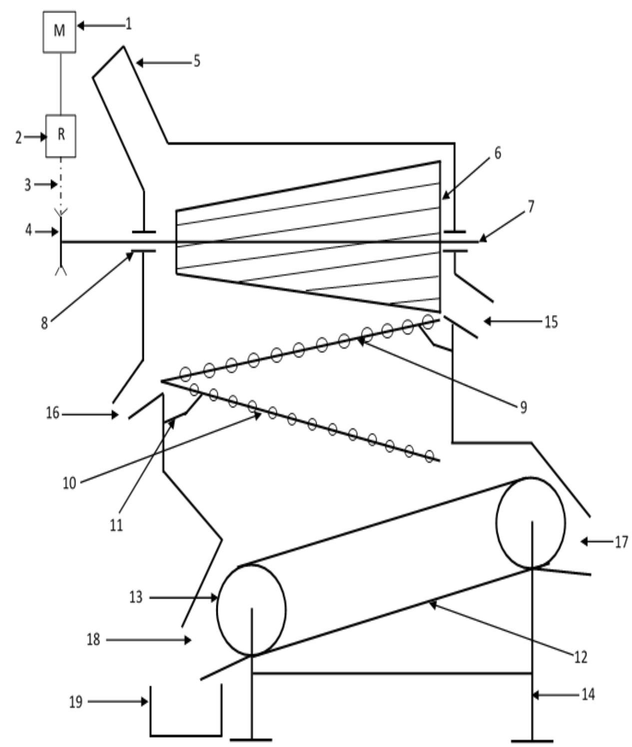


Figure-3: Schematic of the operating principle.
1: heat engine; 2: speed reducer; 3: belt; 4: driven pulley; 5: hopper; 6: drum; 7: IV shaft; 8: bearing; 9: 12 *mm* perforated grid; 10: 10 *mm* perforated grid; 11: eccentric; 12: belt; 13: belt roller; 14: frame; 15: outlet of uncrushed shells and nuts; 16 and 17: outlet of shells; 18: chute; 19: recovery bin.

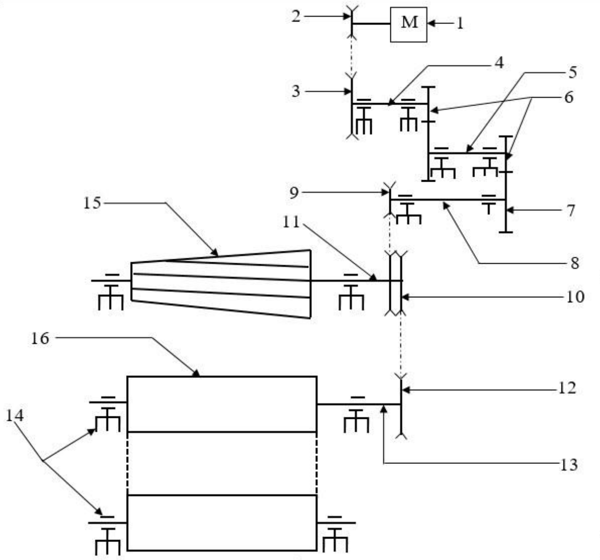


Figure-4: Kinematic 4 diagram of the separating machine. 1: Heat engine; 2: Engine driving pulley; 3: Driven pulley; 4: Gearbox shaft I; 5: Gearbox shaft II; 6: Driving gearwheels; 7: Driven gearwheel; 8: Gearbox shaft III; 9: Driving pulley; 10: Double groove drum pulley; 11: Drum shaft IV; 12: Pulley driven by pulley 10; 13: Belt shaft V; 14: Bearings; 15: Drum; 16: Belt roller.

Modeling

This part is devoted to the identification of mathematical models relating to the various components of the shell and palm kernel separating machine. These models reflect the physical phenomena that govern the operation of the components. The essential parameters of the assembly of the separating machine have a total of six shafts in addition to that of the motor. Having opted for a reducer, the latter alone has three shafts that are shafts I, II and III. The rolling drum is carried by a separate shaft, shaft IV. The two carpets are connected to each other and carry the shaft V and finally the eccentric for the shaft VI.

Drumsizing: Power output P_u: The useful or absorbed power is the power necessary to bring to the drum so that it can turn in order to ensure its separating function. The determination of the useful power will thus be done by taking into account the useful force and the speed of rotation of the drum.

According to the British Medical Research Council System[5] we have:

$$f_u = 3.5\% \text{ x } F_{phy} \quad (1)$$

Thus, the useful power of the drum is:

$$P_u = f_u \text{x} V_{tam} \quad (2)$$

With f_u the useful force to turn the drum in N; V_{tam} the rotation speed of the drum in m/s; P_u the useful power in W.

Sizing of the conveyor belt: Power output P'_u: The useful or absorbed power P_u' is the power necessary to bring to the belt so that it can turn in order to ensure its separating function. The determination of the useful power will be done by taking into account the transmission ratio and the power P_u of the drum. Then:

$$P_u' = r_7 \text{x } P_u \quad (3)$$

With r_7 the transmission ratio on the belt shaft; P_u the useful power of the drum in W; P_u' the useful power of the belt in W.

Eccentric sizing: Useful power: The eccentrics being also an element of the machine must absorb a certain power. This power P_{N1}will be determined according to the transmission ratio on the shaft I and the power on the shaft. We have:

$$P_{N1} = r_1 \text{x } P_{m1} \text{ or } P_{m1} = \frac{P_u}{r_{tam}} \text{ hence: } P_{N1} = \frac{r_1.P_u}{r_{tam}} \quad (4)$$

With: r_1 the transmission ratio on the I-shaft; P_u the useful power of the drum in W; r_{tam} the transmission ratio of all the rotating elements.

Motorsizing: Motor power Pm: The dimensioning of the motor becomes easier to obtain since the necessary power of all the rotating elements of the machine has been evaluated. The power of the motor will therefore be a function of the power output of the drum, the conveyor belt and the eccentric. Thus:

$$Pm = P_u + P_u' + P_{N1} \quad (5)$$

With: P_u the useful power of the drum in W; P_u' the useful power of the belt in W; P_{N1} the useful power of the eccentrics in W; Pm the motor power in W.

Regarding the speed of rotation, the motor has been chosen according to the corresponding power and according to the choice, it carries a speed of rotation Nm.

Belts, sprockets and shafts: Model of the characteristic parameters of the belt: As mentioned above, the dry palm kernel separator alone has six (06) different distinct shafts. The determination of the number of belts, center distances, pitch length and many other things between the pulleys follows the same approach which will be presented below.

V-belts are used to transmit the movement of the engine to other components. A belt is characterized by: its original length, its center distance, its winding angle, its admissible power, the tension of the slack side and the tension of the taut side.

Model of the theoretical center distance: The center distance is the distance between the axes of the two pulleys. To determine the center distance$a$you must first determine its minimum value (a_{mini}) en $mm$and its maximum value (a_{maxi}) also in mm.

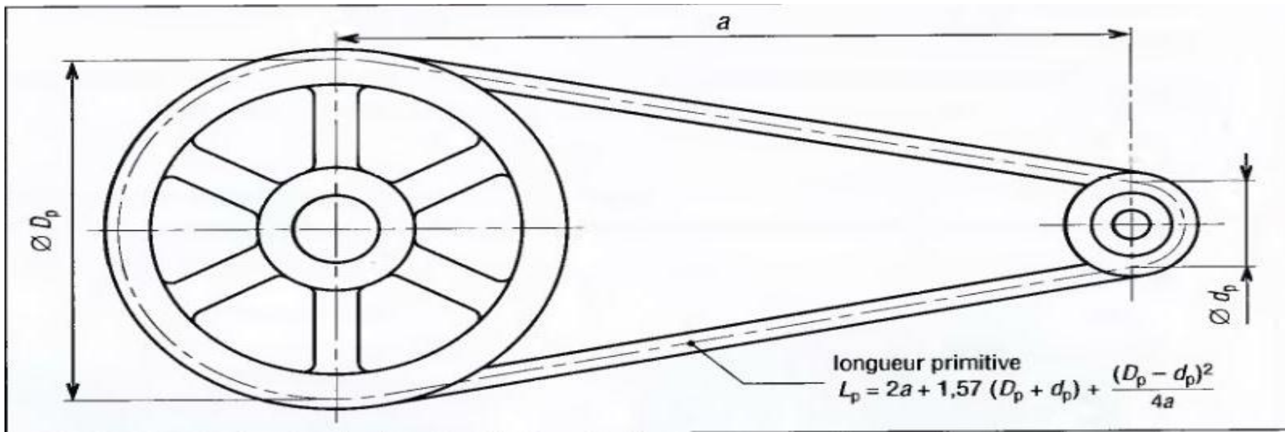


Figure-5: Diagram of a belt drive[6].

To determine the belt center distance, first determine the ratio $\frac{D_p}{d_p}$.[6]

With: D_p the diameter of the large pulley and d_p the diameter of the small pulley.

When the ratio $\frac{D_p}{d_p}$ is between 1 and 3, a_{mini} and a_{maxi} are determined by the relations (9) and (10).

$$a_{mini} \geq \frac{D_p+d_p}{2} + d_p \quad (6)$$

$$a_{maxi} < 3(D_p + d_p) \quad (7)$$

After calculation, an approximate value of the theoretical center distance is chosen such that:

$$a_{mini} \leq a < a_{maxi} \quad (8)$$

Primitive length model: The primitive length calculated here is the reference length given by the following expression[7]:

$$L_p = 2a + 1{,}57(D_p + d_p) + \frac{(D_p-d_p)^2}{4a} \quad (9)$$

After calculating L_p en mm $a$fter calculating the distance between centres of the belt, the table with the extracts of the indicative basic lengths of V-belts is used to select a standardised length in relation to the calculated theoretical length. From the normalized pitch length, the actual center distance (in mm) of the belt is calculated by the formula[6] (13):

$$a_r = \frac{L-\frac{\pi}{2}(D_p+d_p)}{2} - \frac{(D_p-d_p)^2}{4\left[L-\frac{\pi}{2}(D_p+d_p)\right]} \quad (10)$$

Winding angle model: The winding angle is the angular deviation between the direction of the belt and the horizontal. It will be determined by the following formula[6]:

$$\theta = 180 - 2\sin^{-1}\left[\frac{D_p-d_p}{2a}\right] \quad (11)$$

With: Dp the diameter of the large pulley in *mm*; dp the diameter of the small pulley in *mm*; θ the winding angle in °C.

Model of the admissible power P_a: The allowable power is determined with the formula below:

$$P_a = P_b.K_L.K_\theta \quad (12)$$

With: $P_b$the basic power KW; K_L correction coefficient for the basic length; $K_\theta$correction coefficient for the winding angle[2]; Pa the permissible power in KW.

Parameter model of the gear system: Module: The modulus m is the value that is used in the use of gears to define the characteristics of the gears[6],

$$m = 2{,}34\, X \sqrt{\frac{F_t}{KXR_{pe}}} \quad (13)$$

With: F_t the tangential force on the tooth in N; K the tooth width coefficient; R_{pe} the practical tensile strength in MPa; m the modulus.

Not primitive: The primitive pitch is determined according to the mathematical model below[7]:

$$\boldsymbol{p} = \pi.\boldsymbol{m} \quad (14)$$

With: π a constant value; m the modulus; $\boldsymbol{p}$ the primitive pitch.

Centre distance: The center distance is determined according to the formula[7]:

$$a = \frac{D_2 + D_3}{2} \quad (15)$$

With: D_2 and D_3: the diameters of the gears in *mm*; a: the center distance in *mm*.

Tree parameter model: For the determination of the different diameters that the machine comprises, it will be necessary to take into account the phenomena of bending, torsion and shearing since they are subjected to them. Thus, each shaft in resistance to these different phenomena, the following formula[8] is used to determine the shafts:

$$\begin{cases} d \geq \sqrt[3]{\frac{16*C_{max}}{\pi*R_{pg}}} \\ d \geq \sqrt[3]{\frac{32*M_{fmax}}{\pi*R_p}} \\ d \geq \sqrt{\frac{4*T_{max}}{\pi*R_{pg}}} \end{cases} \quad (16)$$

Once the various determinations are made, using the table of standard values for diameters in the designer's guide, the appropriate value is chosen. The same principle is to be followed for all the trees in the separator.

With: C_{max} the driving torque in N.m; R_{pg} the shear strength or stress in MPa; M_{fmax} the bending moment in N.m; R_p the torsion strength or stress in MPa; T_{max} the tension in N; d the diameter in *mm*.[8]

Estimate of the cost of the machine: The estimation of the cost of the machine corresponds to the determination of the cost at which the machine can be sold. This cost will be determined according to[9]: i. The cost C_m of the raw materials used; ii. The cost C_u of machining the parts; iii. The cost C_P of standard parts and accessories; iv. The cost C_b of the design office; v. The cost C_o of labor.

Let: $C = C_m + C_u + C_p + C_b + C_o$(17)

Results and discussion

Results: All the results from the mathematical models (modeling) of each part of the palm kernel separator are listed in Tables 1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, and 12 below.

Table-1: Value of the characteristic parameters of the drum.

Parameters	Rotation frequency N (*rpm*)	Useful force f_u (*N*)	Drum diameters $_{t1}$ Det D_{t2} (*mm*)	Power consumption P_u (*W*)	Kinetic energy E_c (*J*)	Moment of inertia at start J_Δ (*kg.m*^{2})
Values	6	18,97	260 370	4,401	1,35	0,0568

Table-2: Characteristic parameter values of the belt drum.

Carpet Drum				
Parameters	Rotation frequency N (*rpm*)	Useful force $f_u^{'}$ (*N*)	Drum diameter (*mm*)	Power input $P_u^{'}$ (*W*)
Values	6	17,129	210	3,974

Table-3: Value of the characteristic parameters of eccentrics.

Parameters	Rotation frequency N (*rpm*)	Power consumption P_{N1} (*W*)
Values	300	5,154

Table-4: Characteristic parameter values of the motor (Engine).

Parameters	Motor power P_m (*W*)	Engine speed N* (*rpm*)
Values	15,685	1500

Table-5: Characteristic parameter values of the belt drive system between the motor and the gearbox.

Transmission system between the motor and the gearbox.							
Parameters	Operating power P_S (*kW*)	Belt speed V_C (*m/s*)	Original length L_P (*mm*)	Centre distances a_r (*mm*)	Basic power P_b (*kW*)	Allowable power P_a (*kW*)	Winding angle θ (°)
Values	0,1768	4,398	1205	415,98	0,395	0,267	144,74

Table-6: Characteristic parameter values of the belt drive system between gearbox and eccentric.

Transmission system between the Gearbox and the Eccentric.							
Parameters	Operating power P_S (*kW*)	Belt speed V_{C1} (*m/s*)	Original length L_P (*mm*)	Centre distances a_r (*mm*)	Basic power P_b (*kW*)	Allowable power P_a (*kW*)	Winding angle θ (°)
Values	0,16442	4,39823	2100	610,4	0,395	0,395	180

Table-7: Characteristic parameter values of the belt drive system between the gearbox and the drum.

Transmission system between the gearbox and the drum.							
Parameters	Operating power P_S (*kW*)	Belt speed V_{C6} (*m/s*)	Original length L_P (*mm*)	Centre distances a_r (*mm*)	Basic power P_b (*kW*)	Allowable power P_a (*kW*)	Winding angle θ (°)
Values	0,148	0,113	1455	356,45	0,027	0,02	149,59

Table-8: Characteristic parameter values of the belt drive system between the drum and the belt drum.

Transmission system between the drum and the Belt drum.							
Parameters	Operating power P_S (*kW*)	Belt speed V_{C7} (*m/s*)	Original length L_P (*mm*)	Centre distances a_r (*mm*)	Basic power P_b (*kW*)	Allowable power P_a (*kW*)	Winding angle θ (°)
Values	0,1336	0,1131	2700	784,8	0,08	0,11	180

Table-9: Characteristic parameter values of the gear system.

Transmission system between the drum and the belt drum.							
Parameters	Module m	Number of teeth Z_1, Z_2, Z_3 and Z_4	Not primitive *p*	Center distance a (*mm*)	Tooth width *b* (*mm*)	Head diameter d_{a2} and d_{a3} (*mm*)	Foot diameters d_{f2} and d_{f3} (*mm*)
Values	2	Z_1= 30, Z_2=100 Z_3=100, Z_4= 30	6,28	130	19	d_{a2}= 64 d_{a3}= 204	d_{f2}= 55 d_{f3}= 195

Table-10: Parameter values of tree characteristics.

Parameters	Tree I		Tree II		Tree III	
	Diameter d (*mm*)	Safety coefficient *s*	Diameter d (*mm*)	Safety coefficient *s*	Diameter d (*mm*)	Safety coefficient *s*
Values	15	2,5	15	2,5	15	2,5

Parameters	Tree IV		Tree V	
	Diameter d (*mm*)	Safety coefficient *s*	Diameter d (*mm*)	Safety coefficient *s*
Values	25	2,5	25	2,5

Table-11: The cost of the separator.

Cost of raw materials C_m (*$ US*)	761
Cost of machining parts C_u (*$ US*)	78
Cost of design office C_b (*$ US*)	125,85
Cost of labor C_o (*$ US*)	300
Overall cost C (*$ US*)	1264,85

Analysis: Engine: The power of the motor required to run the separator is 15,685 W or 0.021 horsepower (*HP*). In order to preserve the machine from malfunctions during use and the scarcity of 0.5 horsepower motors, we have opted for a 1 horsepower motor with a rotation speed of 1500 rpm according to the standardisation of motors.

Drive shafts : Given the complexity of the machine due to the large number of shafts and pulleys involved, it is noted that the calculation on shafts I, II, III, IV and V are respectively according to the standardized values [8]d_1=15 mm, d_2 = 15 *mm*, d_3= 15 *mm*, d_4= 25 *mm* and d_5 = 25 *mm*. Nevertheless, with the help of the draughtsman's guide the standard values of these shafts have been checked with the safety coefficient of the shafts, which attests to the correct dimensioning of the diameters listed above.

The cost of the design office: The cost of the design office represents the 15% of the global cost of the machine or the cost of raw materials, standard parts, accessories and machined parts[9].

Conclusion

The objective of this work is to provide palm oil producers with a low-cost machine capable of helping them in the separation of crushed nuts. To this end, an inventory of the equipment for the separation of palm kernels and shells was carried out with an analysis of their insufficiencies. This phase has allowed us to propose a model of separator adapted to our working environment, whose prototype is under construction, including an analysis of performance and economic profitability.

The nominal production capacity of the dimensioned machine is 41.5 tons per year with a power of 1 *HP*.

The industrial realization of this equipment will be of great use to palm oil producers. It will allow to eliminate the major constraints that disturb the palm oil producers in Benin.

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