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Development of a Pedal Powered Garri Frying Machine

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Abstract

This study was aimed at developing a pedal powered garri frying machine key design criteria were considered using a decision matrix. The bowl type garri frying machine which had the highest grading based on the criteria considered was selected for production. The design parameters considered were heat required for garri frying, load bearing capacity and structural strength of pedal mechanism, effort required pedaling the bicycle, length of the chain, shaft and frame design, design of key and bearing selection. The fabricated garri frying machine was assembled and tested using measured loads of wet garri mash. Five different frying trials were carried out with the machine manned by various individuals of given grades and gender distribution. The result of the performance evaluation of the fabricated machine showed that there was a reduced involvement of human effort in the frying of garri which reduced fatigue and discomfort, and the use of motorized stirrer enhanced more uniformity in the grain dynamics and heat propagation which enhanced garrification during frying.

1. Introduction

Cassava is biologically called "*manihot esculenta*" and it is a crop with many varieties. Cassava, is a perennial woody shrub in the Euphorbiaceous (Spurge family) native to South America, but now grown in tropical and sub-tropical areas worldwide for the edible starchy roots (tubers) which are a major food source in the developing world, in equatorial regions including Africa, South America, and Oceania. Cassava also known as yucca, manioc, and tapioca is a major food crop in Nigeria [1]. It supplies about 70% of the daily calorie of over 50 million people [2] and about 500 million people in the world. It is a basic staple food to more than 70% of Nigerian population and consumed at least once every day [3]. History has it that the emancipated slaves introduced the cassava crop into southern Nigeria while returning from South America through Sao Tome and Fernando Po Islands. According to [4], cassava as a crop, did not become important in Nigeria until the end of the nineteenth century when processing techniques were introduced. Cassava is largely consumed in many processed forms in Nigeria. Its use in the industry and livestock feed, is well known, but is gradually increasing, especially as import substitution becomes prominent in the industrial sector of the economy.

Garri is a processed fermented product from cassava, which is consumed in Nigeria as well as in most countries of the West African coast and in Brazil. In sub-Saharan Africa, cassava is mainly a subsistence crop grown for food by small Scale farmers, who sell the surplus and it grows well in poor soils [5]. Its particle size ranges from less than 10µm (fine) to more than 2000µm (gross) [2].

Nigeria is currently the largest producer of cassava in the world with an annual production of over 34 metric tons (Mt) of tuberous roots [4] and Edo State is the largest producer of the crop in Nigeria followed by Oyo State. Garri is a widely consumed Nigerian food with an estimated 4.2 million tons produced in 2009.

Garri frying is still mostly being done manually, with the attendant ergonomic risk associated with the repetitive movement of the arms, musculoskeletal disorders and hazards due to unhealthy smoke inhalation. Traditionally, women in shallow earthenware of cast-iron pans (locally called “Agbada” in Benin) fry garri over a wood fire [6]. Women use spatula-like paddles of wood or calabash to press the sieved mash against the hot surface of the frying pan and turn it vigorously to avoid caking. Attempts at the development of improved methods for the processing of cassava have focused mainly on full automation with the use of electric, diesel and solar powered machines. The non-availability of electricity in the rural areas, the ever-increasing cost of diesel, attendant air pollution and lack of needed technical manpower for proper maintenance of installed machines is the propelling factor behind this project. The question which this project aims to answer is how this village technique can be simulated and mechanized using the same approach to achieve the same or better quality of the product. The various designs of electric powered cassava frying machine available in the industry are costly to maintain and have the tendency to introduce pollutants to the processed garri. A major source of such pollutant is the grease used in oiling the gears which sits right above the processing area of the garri. On close observation, some of the grease was seen dropping into the garri while frying. Also, little fragments of the grinding metal wheels also introduced impurities to the garri due to the wear and tear of the metal parts during rotation. An alternative to these mechanized garri frying machines is the development of a pedal driven garri fryer.

2. Methodology

The methodology covered the design concept, material selection, shaft and frame design, design of key and bearing selection. Other design parameters considered include, heat required for garri frying, load bearing capacity and structural strength of pedal mechanism, pedaling effort required by the bicycle and length of the bicycle chain.

2.1 Design Concept

Design criteria considered include: no form of electrification of the machine; the frying machine should be devoid of human swiveling with the hand; direct contact of humans with the fire during frying should be eliminated; the frying bowl should have a mechanical turner actuated by a pedal; humans should be comfortably seated to pedal and actuate the garri turner; the frying bowl should be made of materials with heat capacity high enough to hold the frying temperature (110 -130^{o}C) of garri. The two concepts considered were pedaled fryer with halved cylindrical frying trough and pedaled garri frying machine with bowl type trough. Grade points were allocated to the two concepts based on the criteria (such as cost, ease of product, bulkiness/space requirement, and simplicity. Decision matrix was used to select the pedaled garri frying machine with bowl type trough which was a more viable concept based on the design criteria considered.

2.2 Material Selection

The materials used for this study were selected based on availability, low cost, and ability to carry out specific functions with regards to the service condition of the developed garri frying machine.

Cast aluminium was selected for the frying bowl of the machine because of its good corrosion resistance ability and thermal conductivity. Mild steel was selected for the shafts and frame of the machine because it is cheap, resistant to shear and bending forces.

2.3 Methods

To carry out the machine production with proper dimensioning and materials selection, key design and operational inputs were considered. These inputs include the following: the anticipated garri load requirement was put at 2 bags of 50kg per bag per day that will meet daily consumption of a sampled population of 100 people; due to losses and additional mouths to feed arising from visitors and unforeseen situations, an additional bag was added, hence total bags of garri to be fried will be 3 bags per day; consumption rate of garri by an individual form the sampled population is 1.5kg/day; average working hours for garri production is from 5hrs; therefore, kg/hr production is 30kg/hr; and considering the bulk density of garri, the volume equivalent of 30kg of garri is mass of garri (kg)/bulk density of garri.

According to Ukpabi and Ndimele [7], well-fried and good garri should have a bulk density of 0.56g/cm^3 to 0.908g/cm^3. Therefore, volume of fried garri (anticipated) will be 3303cm^3 (i.e.$^{3000}/_{0.9.8}$). The frying bowl should be such that 2πrh = 3303cm^3. For batch frying, the garri can be fried intermittently within the hour,

$$\therefore \text{frying bowl volume (FBV)} = 2\pi rh + \text{factor of safety} \tag{1}$$

where, r = radius of bowl,

h = height of bowl.

The height of the entire machine is such that an operator of average height of 5.6ft can access it conveniently. Hence, height of machine is required to be << 5.6ft. The major parts of the machine are made from steel material, hence, some design specification according to the American Society of Mechanical Engineers (ASME) code have to be adopted as follows:

i. Density of steel (Ds) = 7.86410kg/m^3
ii. Allowable working stress of steel (in tension or compression) = 112MPa for shaft without keyways and 84MPa for shaft with keyways (ASME)
iii. Combined shock and fatigue factor applied to bending moment k_b=1.5
iv. Torsional moment factor, C_t = 1.0
v. Shear modulus of rigidity of steel =7.93*10^3 N/m^2

Other relevant variables and design inputs include:

vi. Specific heat capacity of water is = 4.2 kg/kJ/°C
vii. Mass of moisture to be removed from the garri per round = 16.29 kg (as measured from moisture content of wet unfired garri)
viii. Average load bearing capacity of bike = 100kg (experimental determined to accommodate weighty people
ix. The height of the bicycle for ease of mounting and pedaling
x. The structural stability of the bicycle mechanism
xi. The torque generated by the pedaling of the bike.

2.3.1 Heat Required for Garri Frying

From heat equation [8],

$$Q = mc(T_f - T_a) \tag{2}$$

where, Q = Heat required

m = moisture content of un-fired garri
c = Specific heat capacity of water (moisture), 4.2kg/kJ/°C
T_f = Drying temperature of Garri, 90°C
T_a = Ambient température, 34.3°C
$\therefore Q = 3810\ kJ/s.$

2.3.2 Design of the Frame

The frame was designed to give balance to the frying bowl as well as create adequate clearance from the ground for proper airflow from under the machine. For proper ergonomic design, the human operator should be able to view the bowl conveniently without subjecting the body to stress.

$$\text{Height of the frame (HF)} = \frac{average\ height\ of\ human\ operator}{2} \tag{3}$$

$\therefore HF = {}^{1.68}/_{2} = 0.84m.$

2.3.3 Load Bearing Capacity and Structural Strength of Pedal Mechanism

From experimental weights taken from various individuals, the following body measurements were taken:

(i) Maximum operational load that can be carried by the bicycle = 100kg
(ii) Average height of man = 1.77m
(iii) Approximate Length of upper body = 0.8m
(iv) Approximate length of lower body = 0.91,
∴ vertical height of bicycle = 0.91m
(v) Distance between bicycle sit and handle ≤ 0.8m say 0.5m (assumed)
(vi) Pedal distance from sit = 0.91/2 = 0.455m.

Considering the estimated bicycle dimensions, the load diagram is shown in Figure 1.

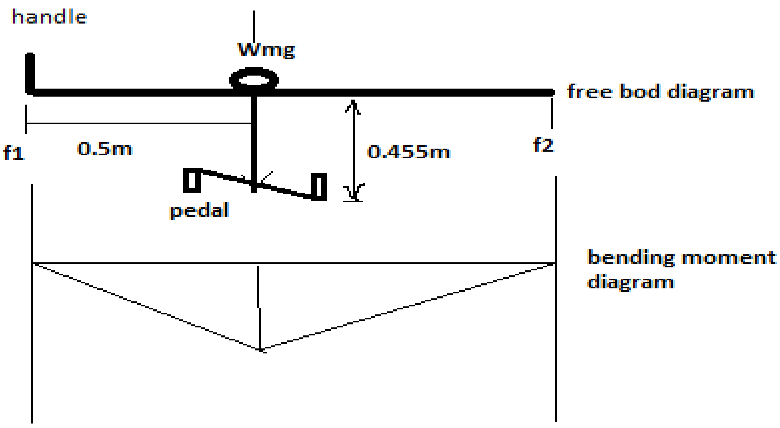

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Figure 1: Load Diagram of Power Generator Bicycle

Considering the free body diagram of Figure 1, the total resultant moment on the bicycle should be zero. That is,

$$(F1 \times 0.5) + (F2 \times 0.455) - W_{mg} = 0 \tag{4}$$

$\therefore W_{mg}$ = 1000N

The bending moment diagram shows the point of likely failure of the load-carrying span, hence, should be made with a hollow structural pipe of appreciable strength and reinforced by tensioned springing or a vertical bracing along the point of failure. The point $F1$ and $F2$ should be vertically supported with mild steel of strength each capable of carrying a weight of above 1000N hence design and load carrying capacity of the bicycle will be efficient.

3.3.4 Effort Required Pedaling the Bicycle

Power required to power bicycle is equal to the torque on the chain. Where;
Torque on pedal chain = (weight of pedal chain + length of chain)
The resulting torque from the chains must be exceeded by the human effort or energy to rotate the wheels of the bicycle through the pedals.

2.3.5 Length of the Chain

The length of the chain, L = K.p (5)

But, $K = \frac{(T_1 + r_{2)}}{2} + 2x/p + [\frac{(T_2 + T_{1)}}{2\pi}]^2 (p/x)$ (6)

where,
T_1 = number of teeth on the smaller driven (shaft sprocket),
T_2 = number of teeth on larger driver (pedal sprocket),
p = pitch of the chain, and
x = center distance.

According to Khurmi [9], the value of K obtained from the expression in Eqn. (6) must be approximated to the nearest even number. Increasing T_2 and reducing x will decrease the value of K and consequentially the value of L. This will make the torque on the driving pulley increase and effectively pull the driven pulley provided friction at the wheels and pulley are drastically minimized or eliminated. The increase of T_2 relatively larger than T_1 will increase the number of turns of the driven sprocket.

2.3.6 Shaft Design

Two shafts are utilized in the machine and are evaluated as follows:

2.3.6.1 Vertically Oriented Shaft

The weight of shaft connecting the gear train to the paddles is given as:
Weight of shaft + weight of stirring paddles
Weight of shaft = mg = ρgv (7)
where, ρ = density (σ) of mild steel = $7.85kg/m^3$
The volume of the steel material, $V = \pi r^2 h = 0.11m^3$
Therefore; weight of the shaft = 8.5kg
Weight of stirrer = mg
where, Mass = Volume × Density.
but, volume, V= area × thickness of plate
For the two stirrers =Area = $2 \times (20 \times 4 \times 10^{-3}) m^2$
Therefore; Volume = $2\times (20 \times 4 \times 10^{-3} \times 0.002)\ m^3$
Mass = density x volume = 0.0025kg
Weight of paddles = mg = mass of the paddles x g

where, g = acceleration due to gravity (g) = 9.811m/s^2)
∴Weight of paddles = 0.024kg
Total weight of shaft and stirrers = 8.5 + 0.02 8.52kg
Since the vertical shaft is subjected to twisting moment, hence the torque acting on the shaft is

$$T = \frac{J\tau}{r} \tag{8}$$

where,

$J = \frac{\pi d^4}{32}$ = polar moment of inertia of the shaft about the axis of rotation,
T = torsional shear stress,
r = $d/2$ = distance from neutral axis to the outer most fibre,
d = diameter of the shaft

For the solid round shaft, the torque acting on it may be written therefore as
$T = \frac{\pi\tau d^4}{16}$ = 0.264N-m
Power transmitted by the shaft [10],

$$P = \frac{2\pi NT}{60} \tag{9}$$

where, N = speed of the shaft in r.p.m (speed of the gear train),
∴ Power transmitted by the shaft, P was determined to be 0.1kW.

Considering factor of safety of a gear train, a motor of 0.25kW may be used to drive the shaft. The average power produced by man is approximately 75W (0.075kW) for a healthy non-athlete. A simple rule is that most people who engage in delivering power for an hour or more will be most efficient when pedaling in the range of 50 to 60 revolutions per minute (rpm). To reduce the effort required to pedal the drive the machine and yet maximize the work output can be achieved by reducing using a bigger sprocket at the pedal end and a smaller sprocket at the garri frying end. In that case for a complete revolution of the bigger sprocket with n number of tooth, the smaller sprocket turns with m number of tooth turns by ($\frac{n}{m}$) revolutions.

2.3.6.2 Horizontally Oriented Shaft

The horizontal shaft is subject to both a twisting and bending moment under the influence of the sprocket and chain. For a uniformly distributed load on the shaft, with a given diameter of the shaft, d = 0.2m, r =0.1m, the bending moment (M) on the shaft is given as:

$$M = \frac{\pi\sigma_b d^3}{32} \tag{10}$$

where, d = the diameter of shaft,
σ_b = allowable bending stress = 84N/mm^2.
∴ M = 0.065Nm.

For the ductile steel material, the maximum shear stress theory or Guests theory applies, hence, the equivalent twisting and bending moments are 0.264N-m and 0.065N-m respectively. From the above calculation the power required to drive the shaft is 0.234kW and it is therefore suitable for the design to use 0.50kW to 1kW electric motor considering factors of safety and availability.

2.3.7 Design of Key

The length (L) of key is given as,

$$L = \frac{\pi D}{2} \tag{11}$$

where, L = length of key in mm

d = shaft diameter = 20×10^{-3}m

∴ L = 31.4mm.

2.3.8 Bearing Selection

The number of bearings utilized in the machine is two, which were used for the horizontal shaft rotation and balancing. Some governing conditions guided the selection of the bearings used for supporting the rotating shaft, chain and gear. These include: a) The selection of rolling contact bearings over sliding contact bearings due to the former's advantages that were closely desired for the nature of the machine crucial amongst which included; Its low starting and running friction within the desired low speed, its ability to withstand momentary shock loads, accuracy of shaft alignment and low cost of maintenance. b) The desired speed to be transmitted from the shaft as supplied from the motor is desired to be low and far less than 2000rpm. c) The bearings required needed to have ability to bear load at this speed. d) The minimum static and dynamic load rating of the bearing has to exceed the bearing load of the shaft.

2.3.9 Dynamic Equivalent Load for Rolling Contact Bearings (DEL)

This is the constant stationary radial load (in case of radial ball or roller bearings) or axial load (in case of thrust ball or roller bearings) which, if applied to a bearing with rotating inner ring and stationary outer ring, would give the same life as that which the bearing will attain under the actual condition of load and rotation [10]. The DEL is denoted by W, for the radial and angular contact bearings under combined constant radial load, W_R and constant axial or thrust load W_A is given by the following expression;

$$W = X.V.W_R + Y.W_A \tag{12}$$

where, V = A rotation factor = 1 for all types of bearings when the inner race is rotating and the values of radial load factor X and axial or thrust factor Y for the dynamically loaded bearings may be taken from references or appendix of this literature.

2.3.10 Dynamic Load Rating for Rolling Contact Bearings under Variable Loads (DLR)

The DLR denoted by C is the constant stationary load (in case of radial ball or roller bearings) or constant axial load (in case of thrust ball or roller bearings) which a group of apparently identical bearings with stationary outer ring can endure for a rating life of one million revolutions. This is equivalent to 500 hours of operation at 33.3 rpm) with only 10 parent failure ([10]. It is given as,

$$C = W\ (L\ /\ 106)\ 1/k \tag{13}$$

where, W = equivalent dynamic load,

L = service life rating of the ball or roller bearing.

The relationship between the life in revolution L and the life in working hours L_H is given by

L = 60N.L_H revolutions where N is the speed in rpm

k = 3, for ball bearings and 10/3 for roller bearings

For this study, ball bearings were found suitable for the design. In selecting the most suitable ball bearing, the basic dynamic radial load was multiplied by a service factor (K_s) to get the design basic dynamic radial load capacity.

The basic design calculations that were done were used to develop the working drawing of the pedal operated garri frying machine as shown in Figures 2, 3, 4 and 5 respectively. The pictorial view of the developed pedal operated garri frying machine is shown in Plate 1.

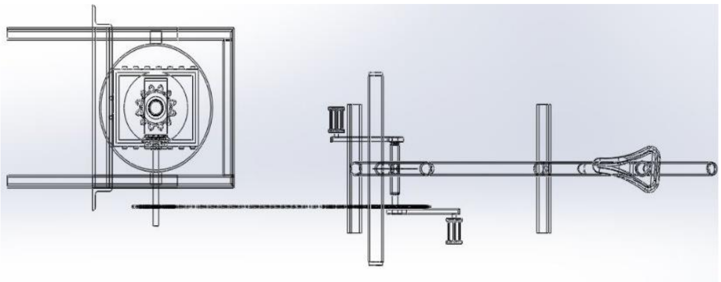

Figure 2: Plan View of Pedal Operated Garri Frying Mach

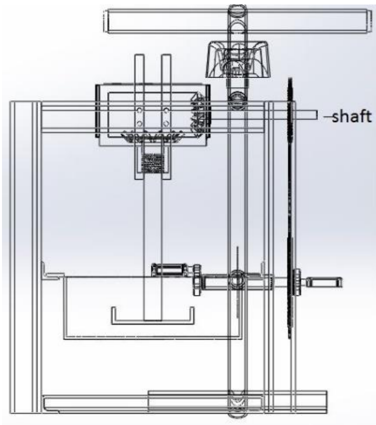

Figure 3: Front View of the Pedal Operated Garri Frying Machine

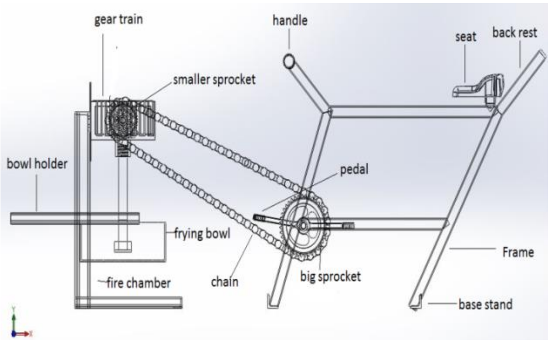

Figure 4: Side View of the Pedal Operated Garri Frying Machine

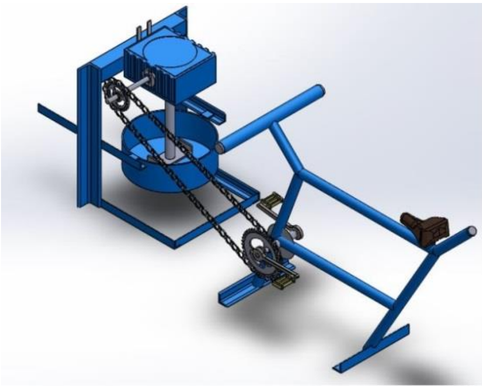

Figure 5: Rendered Isometric View of the Pedal Operated Garri Frying Machine

Plate 1: Pictorial View of the Developed Pedal operated Garri Frying Machine

2.4 Performance Evaluation

The fabricated garri frying machine was assembled as shown in Plate 1. Measured loads of wet garri mash were introduced into the frying bowl to be fried. The combustion chamber was fired prior to the introduction of the wet garri in order for the bowl to heat up and remove any wetness or moisture in it. After a given time of frying while stirring the garri inside the bowl, it was discharged into a collecting ware. Five different frying trials were carried out with the machine manned by various individuals of given grades and gender distribution. The use of different age grades and genders was necessitated to study the ease of operation of the machine and the loading capacity for efficient frying operation.

3. Results and Discussion

The batch loads, time of frying, weight of the garri before and after frying were recorded and tabulated as shown in Tables 1.

Table 1: Results of Fried Garri using the Produced Pedal Fryer

Test no	Weight of garri (kg) before frying	Weight of garri (kg) after frying	Moisture lost (%)	Frying temperature(°C)	Time of frying (min)
1	15	12	20	105	20
2	15	11.8	21	102	25
3	15	11.6	22	105	21
4	15	12	20	105	25
5	15	11.9	20	103	24
Average	**15**	**11.86**	**20.6**	**104**	**23**

Figures 6 and 7 show graphs illustrating the time it took the human fryer (male and female) to pedal the machine before fatigue set in.

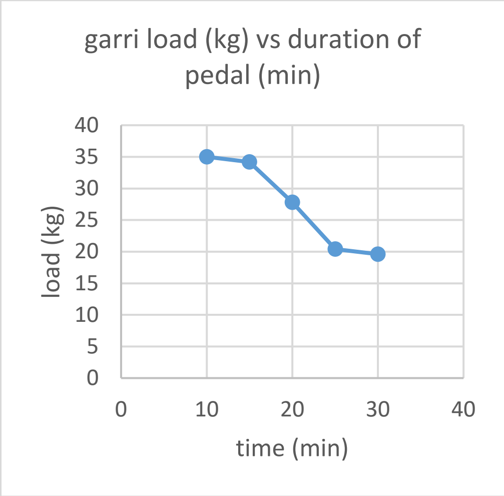

Figure 6: Graph of Garri Load against Time to Pedal by Male Fryer

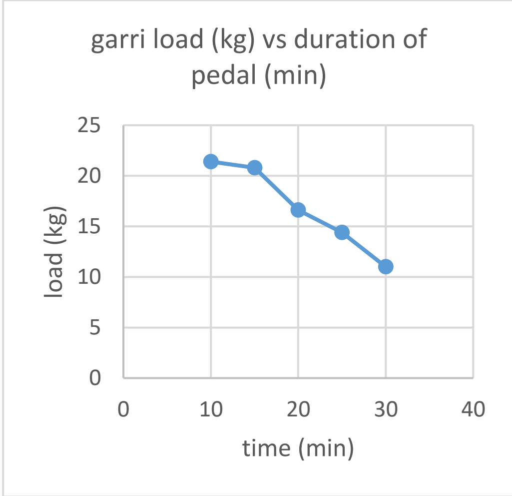

Figure 7: Graph of Garri Load against Time to Pedal by Female Fryers

From Figure 6, it took the male operators a time range of 19.6minutes to 35 minutes to pedal the machine during garri loading of 10kg and 30kg respectively, while Figure 7 showed that it took the female operators an average time range of 11 minutes and 21.4 minutes to operate the machine for garri loads of 10 kg and 30 kg respectively. From the test carried out on the fatigue level of the various individual operators comprising male and females while operating the machine for different garri loads, it was observed that most of the operators took longer time to pedal garri loads between 10 to 20 kg after which there was a significant reduction in duration of operation of the machine amongst the males and females across the age grades considered. On the average, it was concluded that majority of the operators showed high level of alertness and energy in operating the machine and hence 15kg was selected as a marked batch load for the machine during the frying operation and was further used for the garri frying test. Following the test experiment, weighted amounts of 15kg of test samples of the garri mash were fried as shown in Table 1.
The simple average mean was used to determine average mass of garri before and after frying as 15kg and 11.86kg respectively, with an average percentage moisture loss of 20.6%. The average frying temperature was 104°C and the average time taken to fry each batch of garri was 23mins. The ambient temperature of the cassava mash was 32.5°C. The efficiency of the developed garri frying machine was calculated as 79% using the following relationship:

$$\text{Machine efficiency, } \eta = \frac{Average\ load\ output\ of\ garri}{Average\ load\ input\ of\ garri} \times 100 \quad (14)$$

4. Conclusion

Garri frying is a laborious and tasking job considering the conventional method of outdoor frying with local frying pan ‘agbada’ and fire wood. However; with the use of the mechanized garri frying machine to carry out the garri frying operation, it becomes an easy task in which the human fryer is assured of his safety and product quality. Though the best quality garri is obtained by the conventional frying technique as practiced by local producers who manually fry garri using wood charcoal, the development of the present garri frying machine was conceived based on closely related operations and configuration of the conventional method. In that case, the objective was to achieve high grade garri using the pedal operated garri frying machine with additional gains on reduced frying time and safety. It should be of note that in a bid to develop this pedaled power garri frying machine, key features were considered as crucial to the efficient operation and performance of the machine. Such features include: controlled temperature mechanism; a good stirring mechanism; and ergonomically safe seater for the human fryer.

The developed pedal operated frying machine can be operated by both male and female genders with a wide age distribution considering the high level of safety and ease of operation, unlike the conventional method which is considered unsafe for a good number of categories of people. The use of the frying machine has a greater product output than the human operator who is directly involved with the loading, stirring and unloading of each and every batch loads of the garri fried.

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