Design and Construction of an Orange Juice Extractor

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Abstract—**It is required to construct a manually operated household orange juice extractor to be portable and used in extraction of juice. It is also required to be cheap and easily manufactured. It can be easily operated and its efficiency is very high. The dimensions are 160mm diameter and 350mm height. The machine combines the actions of chopping and beating, often by macerating. It consists of two main parts – a goblet and a manually operated mechanical unit. The manually operated mechanical unit consists of a pair of bevel gears, two bearings and two shafts. The bevel gear casing was constructed using 2mm thickness of mild steel sheet. The bearings were then put into position. Afterwards, the shafts were fastened to the bevel gear and passed through the bearings. The handle was then welded to the horizontal shaft. Similarly, the goblet was formed using 1mm thickness of mild steel sheet. Small sharpened blades were then made and fixed onto the impeller shaft. A bearing was then fixed underneath and a shaft passed through. A dynamic seal was put between the shaft, bearings and goblet to prevent leakage. The connection between the gear casing and goblet was done by means of an Oldham coupling designed for misalignment. The machine can extract the juice of about 180-220 oranges per hour.**

***Index Terms*—juice extractor, gears, bearings, shaft, orange**

I. Introduction

One of the oldest cultivated fruits, oranges are among the most popular of fruits for eating out of hand and are by far the most important source of fresh, frozen and canned juice.

A. The orange

The orange fruit is a specialised type of berry known to botanists as hesperidia. It has a soft, pithy central axis surrounded by 10-15 segments containing pulp and juice. Enclosing the segments is a leathery, oily rind that has a white spongy inner part and a harder, orange coloured outer part containing many glands that secrete oil. The segment juice contains sugar; several organic acids (chiefly citric acid) many other components; which give it a distinctive flavour; and high amounts of vitamins A, B and C.

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Oranges grow on evergreen trees of the family Rutaceae. The trees grow to a height of about 30 feet (9m) and are symmetrical and upright. The fruit varies in the number of seeds, from none to many.[1,2]

B. Objectives

The aim of this project is to design and construct a manually operated juice extractor that will extract juice from the orange fruit using available local materials.

C. Importance of the project

Prior to 1920, the orange was considered principally as a dessert fruit. With the spread of orange juice drinking, instead of eating the flesh of the fruit and the growing appreciation of the nutritional value of oranges which are rich in vitamins A, B and C, there is the need for the design and construction of machines for the extraction of juice so they could be stored in refrigerator or canned and drunk at any time.

II. Design Calculation

A. Introduction

Design can be described as a decision-making process where plans are formulated for the physical development of a machine or piece of equipment, whereby all the user's requirements are satisfied.

B. Definition of the problem

It is required to design an extractor to obtain juice from oranges with little effort.

The machine should be

i. Easily operated.
ii. Cheaply and easily manufactured with local resources.
iii. Easily maintained
iv. Portable and easily stored.

C. The components, their function and the operation of the machine

The components of the orange juice extractor may be broadly classified into the goblet and mechanically operated unit. The goblet consists of the following components: a bearing, a bearing support, impeller shaft, a dynamic seal, small sharp blades attached to the impeller. However, the mechanically operated unit consists of two bearings, two bearing supports, a pair of level gears, two pinions, a gear shaft and a handle. The bearings allow for free rotation of the shafts. The bearing supports hold the bearings in position and the dynamic seal prevents leakage of the juice.

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The operator turns the handle and the motion is transmitted to the pinion through the pinion shaft and then to the gear which transmits it to the gear shaft and thence to the impeller shaft. Motion is transmitted from the mechanical unit to the goblet in impeller shaft by means of an Oldham coupling. The blades at the top of the impeller shaft rotate in the orange which has been cut and placed on it, thereby extracting juice from it. Sectional views of the assembly and the various components are shown in figures 1 and 2 of the appendix respectively.

D. Material selection

The material for the pinion is to be of medium carbon steel with 0.45% carbon, hardness with high tempering and of BHN194 and allowable stress (σ) = 500N/mm^2; (σ-1)b = 130N/mm^2. For the gear, medium carbon steel is also chosen with up to 0.45% carbon, but normalised and of BHN 173 and allowable stress and of (σ) = 450N/mm^2 and (σ-1)b = 116N/mm^2.

E. Computation of parameters of the bevel gear

Transmission ratio is the ratio of speed of pinion to the speed of the gear, given by equation 1.

$$i = N1/N2 \tag{1}$$

where N1 is speed of the pinion and N2 is the speed of the gear. N1 = 100rpm, N2 = 300rpm, hence, i = 300/100 = 3:1

Power, P, of a manually operated system is 1/7th of horse power, i.e. approximately 107W. The torque of the machine is given by the ratio of power to angular speed, w, as shown in equation 2.

$$T = P/w \tag{2}$$

The angular speed is given by equation 3, where N is the number of revolutions per minute.

$$w = 2\Pi N/60 \tag{3}$$

Substituting equation 3 in equation 2, we obtain torque as given in equation 4

$$T = p.60/(2\Pi N) \tag{4}$$

Substituting the value of power of 107W and rotational speed of 100rpm for the pinion in equation 4, the torque value, T, is 10.2Nm

From this the tooth length factor is chosen to be ψ_{Re} = 0.30 from charts.

From chart and for M2 = T = 10.2 x 10^3 Nmm, ψ_{Re} = 0.3, i = 3:1, (σ_e)cs = 450N/mm and K = 1.6, we see that external pitch diameter, de$_2$ = 100mm.

The number of teeth for pinion Z_1, say 20 (any number for which undercutting will not result Z_1>18) is suitable. Hence, the number of teeth of gear (Z_2) is given by equation 5 as 60, since i = 3.

$$Z_2 = Z_1 i \tag{5}$$

External pitch module, Me, is given by equation 6 below. For external pitch diameter of 100mm and Z_2 of 60, the value of Me is 1.67mm. From tables, the next standard module is chosen as 2.0mm.

$$Me = de_2/Z_2 \tag{6}$$

Refining the basic reducer parameters, namely external pitch diameter, de$_2$ is given by equation 7. Substituting Me and Z_2 in equation 7, we obtain de$_2$ as 120mm.

$$de_2 = MeZ_2 \tag{7}$$

In our case, de$_2$ = 125 and i = 3.15 are standard as seen in the relevant tables 8 and 9. Also, Me = 2 has been standardised.

To verify the speed, n_2 for i = 3.15. For i = 3.15, Actual rpm $n_2 = n_1/i$ = 300/3.15 = 95.239; n_2 is taken to be 98. Deviation from given rpm is (100-98)/100 x 100% = 2% which is allowable.

$$n_2 = n_1/i \tag{8}$$

Pitch cone radius, Re, is given by equation 9 below. For Me, Z_1 and Z_2 values of 2mm, 60 and 20 respectively, Re value is obtained as 63.25mm.

$$Re = Me/2\sqrt{(Z_1^2 + Z_2^2)} \tag{9}$$

The tooth length or width of face, b, is given by equation 10. For values of ψ_{Re} and Re of 0.3 and 63.25mm respectively, b is obtained as 18.975mm. From tables and standards b is selected as 19mm.

$$b = \psi_{Re} \text{ x } Re \tag{10}$$

External pitch diameter of pinion is given by equation 11. For Me and Z_1 values of 2 and 20 respectively, de$_l$ is obtained as 40mm.

$$de_l = MeZ_1 \tag{11}$$

Pitch angle cot δ_1 = 3.15; from which $\delta_1 = 72.39^0$ is obtained from equation 12 below.

$$\delta_2 = (90^0 - \delta_1) \tag{12}$$

Mean pitch diameter, d_1, is given by equation 13 and the value is obtained as 102.46mm.

$$d_1 = 2(Re-0.5b)\,\mathrm{Sin}\,\delta_1 \tag{13}$$

Mean pitch module, M, is given by equation 14 and the value obtained as 5.12mm.

$$M = d_1/ Z_1 \tag{14}$$

Mean pitch line velocity, V, is given by equation 15 below and the value obtained as 1.61m/s. From tables, at this speed for straight bevel forth gearing and for hardness <BHN 350, 9th degree of accuracy can be taken but because of the need to lower the dynamic load, instead, of 9th, the 8th was adopted.

$$V = \Pi d_1 n_1/60 \tag{15}$$

Refinement of load factor is given by equation 16 and is obtained as approximately 0.19 and for 8th degree of accuracy.

$$\psi_d = b/d_1 \tag{16}$$

In the case that the pinion is mounted symmetrically at the end of the shaft, K' conc. = 1.15.

K conc. Is given by equation 17 and the value obtained is 1.0775.

K conc. = (K' conc. + 1)/2 (17)

From tables, for V = 1.61m/s; BHN<350; 8th degree of accuracy and for bevel gearing; Kdyn = 1.3.

Hence, Load factor, K, is given by equation 18 and the value is approximately 1.4.

K = K conc. x Kdyn (18)

The design contract stress is given by equation 19 below. The value of K_1 is 1.0 and the σ_{cs} is computed as 94.6N/mm^2 Since 94.6 N/mm^2 is less than the allowable stress of 450 N/mm^2 of the weaker component (the gear); the design is safe

$\sigma_{cs} = 1050/((1\text{-}0.5)\ \psi_{Re})\sqrt{1}(M_2K.i)/(de^3{}_2\psi K_1)]$ (19)

$\leq(\sigma)$cs

Determination of main parameters or proportion of the pinion and gear. Already we have de_1 = 40mm; de_2 = 100mm and b = 19mm.

Tooth outside parameters, da_1 and da_2, are computed as 41.21mm and 103.81mm and are given by equations 20 and 21 respectively.

$da_1 = de_1 + 2Me \cos \delta_1$ (20)

$da_2 = de_2 + 2Me \cos \delta_2$ (21)

Tooth root diameters, de_1 and de_2, are computed as 38.9mm and 95.23mm and are given by equations 22 and 23 respectively.

$de_1 = de_1 - 2.5Me \cos \delta_1$ (22)

$de_2 = de_2 - 2.5Me \cos \delta_2$ (23)

In checking the strength of teeth in Bending, Virtual number of teeth of pinion, Zv or, is given by equation 24 and has the value of approximately 66. Similarly, for the gear, Zv, is given by equation 25 and has the value of approximately 63.

$Zmod_1 = Z1/\cos \delta_1$ (24)

$Zmod_2 = Zv = Z2/\cos \delta_2$ (25)

From tables, the tooth from factory, for virtual numbers of teeth Ze or Zv = 66 is y ≈ 0.471. Similarly, for Ze = 63, y = 0.470.

Comparative weighting of bending strength of pinion and gear teeth is given by equation 26. For the pinion and gear the values obtained are 61.23N/mm^2 and 54.52N/mm^2 respectively.

$y(\sigma)b = y \times (\sigma\text{-}1)b$ (26)

Design bending stress, σ_b, at the weakest section of the pinion tooth is given by equation 27 below. For K_L = 1, σ_b = 14.1N/mm which is less than $(\sigma_{b\text{-}1})b_1$ = 130N/mm and thus safe.[3,4,5,6,7,8,9,10,11,12]

$\sigma_b = (2.35\ M_1K \cos \psi)/(Z_1\ y_1\ b\ m^2\ K_L)$; (27)

F. Shaft design

A shaft is a rotating member, usually of circular cross section (either solid or hollow), transmitting power. It is supported by bearings and support gears, sprockets, wheels, rotors, etc. and is subjected to torsion and to transverse or axial loads, acting singly or in combination.

G. Determination of shaft size on the basis of strength

The maximum shear stress, τ_{max}, of the shaft is given by equation 28 below.

$$\tau_{max} = 16/(\prod d^3)\sqrt{((K_m M) + (K_t T))} \quad (28)$$

where τ_{max} is the maximum shear stress, d is the shaft diameter, K_m is the numerical combined shock and fatigue factor to be applied, K_t is the corresponding factor to be applied to the computed torques, M is the bending moment, T is torque. For this case bending moment is negligible. K_t is equal to 1.0 for rotating shaft with gradually applied loads. Assuming bending moment is negligible equation 29 is obtained from equation 28 by making shaft diameter the subject of the formula. For τ_{max} = 30N/mm2, T = 10.2Nm, the shaft diameter is obtained as approximately 12mm.

$d = (16/(\prod \tau_{max})T^{3/2})$ (29)

H. Bearing selection

Equations for calculating basic static load rating(Co) in Kg for radial ball bearings is given by equation 30.

$Co = fo\ I\ Z\ D^2 \cos \alpha$ (30)

where i is number of balls in the bearing, Z is the number of balls per row, D is the ball diameter in mm, α is the normal angle between the line of action of the ball load and a plane perpendicular to the bearing axis, fo is 0.34 for self-aligning ball bearing made of hardened steel.

Static equivalent load (Po) is calculated using equation 31 below.

Po = XoFr + YoFa (31)

where Fr is radial load and Fa is the thrust load, Xo is radial factor; Xo = 0.6 for radial contact groove ball bearing, Yo is thrust factor; Yo = 0.5 for radial contact groove ball bearing.

For shaft connected by gears with rotating machines, and no impact, load factor, k, takes any value between 1.1 – 1.5.

Desired Life is given by equation 32.

$L = (C/P)^k$ (32)

Where C is basic dynamic load rating, P is equivalent load K = 3 for ball bearings and 10/3 for roller bearings.

From tables for machines used for short periods or intermittently, i.e. manually operated machines, and whose breakdown would not have serious consequences, generally, Life in working hours, L_h = 4000 – 8000.

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For a ball bearing with life of 4000 hours at 100rpm, from tables C/P = 2.88. For C/P = 2.8, for ball bearings, life in millions of revolution is 23.

Also, for a bearing with life of 4000 hours at 300rpm from tables C/P = 4.01. For C/P = 4.01 for ball bearings life in millions of revolution is 65.

For $d = 12mm$, $D = 28mm$, $H = 11mm$, $d_2min = 10.2mm$, $r \approx 1mm$; static capacity, Co = 15000kg; Dynamic capacity, C = 10.40kg; Maximum permissible speed = 8000rpm.

I. Couplings

A coupling is a device which connects two parts of a mechanical system. An Oldham coupling (double slider coupling) is used for shafts with lateral misalignment amounting to as much as 5% of the shaft diameter with considerable axial play and with relatively small angular misalignment (one degree). It consists of three parts – two slotted hubs and a central piece having two tongues, one on each face, perpendicular to each other.

The design is based on allowable pressure, P on the sliding surface, given as $P = 67kg/cm^2$ from tables.

The force, F due to the total pressure on each side of the tongue is given by equation 33.

$$F = PDh/4 \tag{33}$$

where D is the outer diameter of the centre piece ≈ 3 to 4 x shaft diameter, h is the height of tongue, and torque transmitted by both sides of the tongue is given by equation 34.

$$T = 2F \times D/3 \tag{34}$$

Substituting equation 33 in equation 34, we obtain equation 27 below.

$$T = 1/6\ P\ D^2\ h \tag{35}$$

For this machine, from tables, allowable torque, T, is 10.24Nm.

Allowable pressure of sliding surfaces $P = 67Kg/cm^2$
Shaft diameter, $d_s = 12mm$

Outside diameter of centre piece is given by equation 35. Substituting 12mm for d_s in equation 36 we obtain D as 48mm.

$$D \approx 4 \times d_s \tag{36}$$

Width of tongue W is given by equation 37 and has a value of 5.4mm. Hence, the value is approximated to 6mm.

$$W = 0.45 \times d_s \tag{37}$$

Hence, from equation 33, making height of tongue, h, the subject of the formula we obtain equation 38 and the value of h as 3.96mm. This is approximated to 4mm.

$$h = 6T/(PD^2) \tag{38}$$

J. Seals

Seals are employed to reduce leaks through the gaps between the moving and stationery parts and also to prevent the flow of a substance between two machines having relative motion.

Dynamic shaft seals are relatively more insensitive to dimensional variations and are used for high speed and continuous duty. These seals have positive sealing ability and have low initial cost. The life is dependent on the shaft condition and its finish, speed, alignment, end play, eccentricity, etc., temperature at seal, fluid pressure on seal, and nature of sealed medium.

The life of the seal can be increased by proper and continuous lubrication of the seal, using a shaft with good surface finish, having a minimum disturbance of seal due to shaft end play or eccentricity, keeping dirt away from shaft and seal interface, etc.

The material for the sealing elements could be:
Nitrile with excellent abrasion resistance and temperature range of 0-75^{o}C; Polyactic with fair abrasion resistance and temperature range of 0-100^{o}C; Leather with excellent resistance and temperature range of -10-95^{o}C; and Silicon with good resistance and temperature range of -25-260^{o}C;

III. Assembly, Maintenance and Cost Analysis

A. Assembly

The machine is assembled as shown in the assembly drawing (see drawing attached). The bevel gear casing is made of 2.0mm thickness of mild steel which is joined together by welding. The machine is so constructed that it still remains steady on the ground while in operation. In the assembly of the lower member the bearings are put in position after which the bevel gears mounted on shafts are put in position. For the goblet, the bearing and mechanical seal are put in position. The impeller shaft is then passed through.

B. Maintenance

The design is simple, so that the maintenance of the machine can be done with ease. The goblet should always be washed and dried after use. Any repair work on the impeller can be done with ease of access to the impeller. The goblet can easily be removed from the lower member. The goblet should also be periodically checked for leakage. Bearings should be lubricated periodically or whenever there is no grease in them. Easy access to the bearings is provided. Equally, the bevel gear should also be lubricated whenever dry to ensure longevity.

C. Cost analysis

Cost evaluation depends on the current market price and cost of labour. Market prices are never uniform and tend to rise sharply and unpredictably. Labour costs also vary considerably. This depends on the workshop size, efficiency and reliability of the personnel needed to perform some of the work. Based on the above, a correct analysis is difficult. Cost evaluation for commercial production involves

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consideration of protracted economic principles and characteristics such as direct cost, indirect cost, overhead cost and so on. This was done by estimating production of a single component which gives an insight into commercial production. The table below shows approximate labour and material costs for production of a manually operated household orange juice extractor.

TABLE I
MATERIAL COST

ITEM	QTY	UNIT COST(₦)	TOTAL COST (₦)	($)
Bevel gear(pair)	2	500.00	1000.00	6.45
Bearings	3	200.00	600.00	3.87
Handle (Rubber or plastic)	1	250.00	250.00	1.61
Gear shaft	1	200.00	200.00	1.29
Pinion shaft	1	200.00	200.00	1.29
Impeller shaft	1	200.00	200.00	1.29
1mm sheet of metal for gobet	1	500.00	500.00	3.23
2mm sheet of metal for base (lower base)	1	800.00	800.00	5.16
Dynamic seal	1	200.00	200.00	1.29
Impeller blades	1	200.00	200.00	1.29
Engaging union	1	250.00	250.00	1.61
Sieve	1	100.00	100.00	0.65
Bevel gear (pair)	2	500.00	1000.00	6.45

D. Labour cost

Cutting, bonding, machining, welding ₦500.00
Fitting, painting ₦5,000.00

E. Total

The total cost for the production approximates to ₦5,000.00 as computed from table 1 for material cost and the labour cost. The total cost per item for large-scale production would be less than this amount stated above.

F. Cost of labour

It is the expected that a technician would collect as pay during the period of fabrication excluding the costs of materials, the sum of ₦1,000.00 per day. An estimated five working day period would be required, and this translates to ₦5,000.00.

G. Overhead costs

This includes all other costs that may not be wholly accounted for, e.g. power used in machine operations, tax for production and marketing the juice extractor, transportation, etc. Other costs, which include welding, risk allowance, depreciation, etc., are estimated at ₦500.00 per unit for the juice extractor.

Summary of Total cost = (5,000 + 500) = ₦5,500.00
Approximately ₦5,500.00 ≈ $36.67 is required to fabricate this machine.

IV. Conclusion and Recommendations

A. Conclusion

The orange juice extractor has been designed and constructed using scientific and engineering principles. The machine has a diameter of 160mmm and a height of 350mm. The juice extraction is achieved by means of small sharpened blades on a shaft which rotates with the aid of the bevel drive. The rotation is achieved by turning the handle. The project has made it possible to extract juice from orange fruits easily and faster. The machine is of a unique type and is readily available to those that are interested at an affordable price.

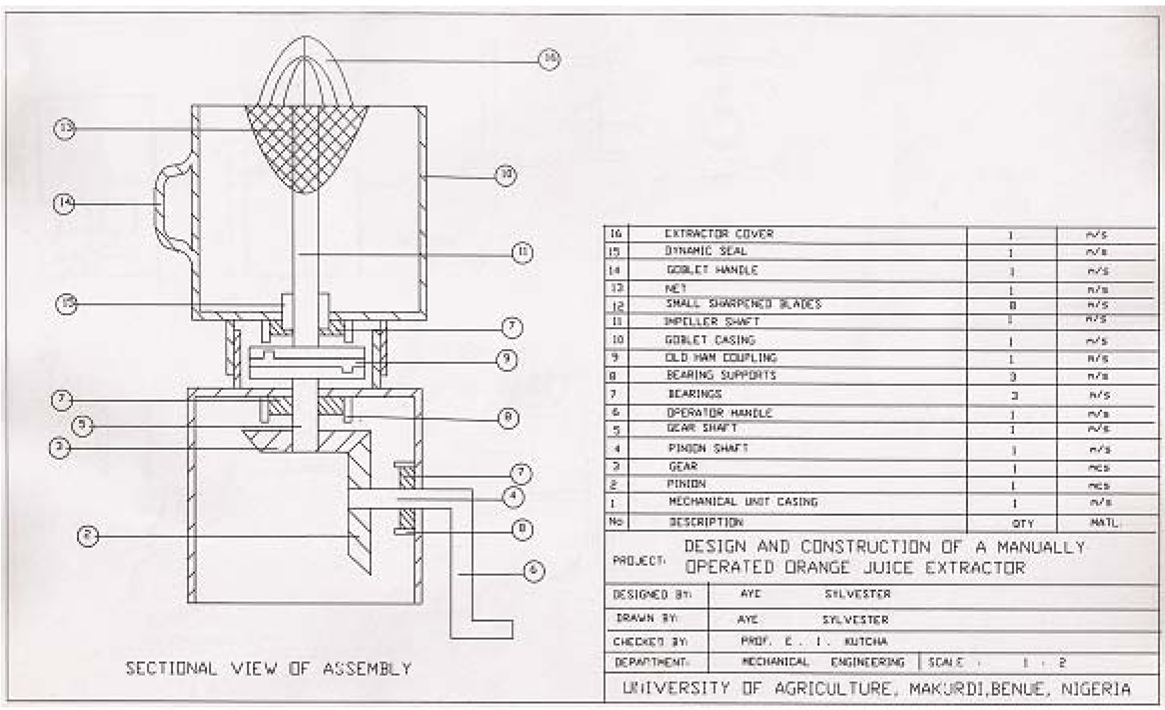

Figure 1: Sectional view of the assembly

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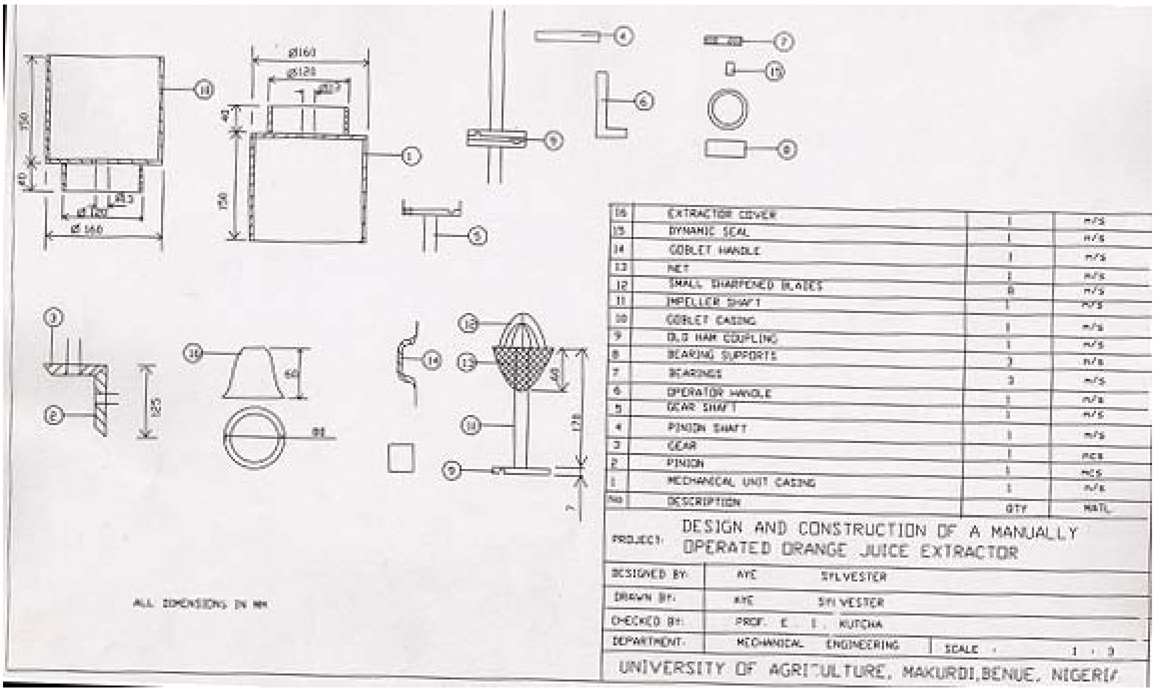

Figure 2: Component parts

APPENDIX

Dessert *(n)* - Sweet course served at the end of meal.
Goblet *(n)* - Hopper where oranges are fed
Juice *(n)* - Liquid part of vegetables, fruit or meat.
Rind *(n)* - Outer coating of fruits, cheese or bacon

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