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Development and Performance of a Sugar Centrifuge

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A centrifuge having capacity to produce 800kg of sugar per day was designed and fabricated in the National Cereals Research Institute, Badeggi. It handles one of the unit processing operations in the Institute's cottage brown sugar processing plant. The machine is made up of a perforated internal basket which is suspended on a vertical shaft that obtains drive from a 20hp vertical mounted electric motor through pulleys and belts. The whole machine rests on a U-channel frame. Test result of the machine shows that it has maximum moisture extraction efficiency of 89-92.4% for a loading capacity of 8-10kg of massercuit per batch. The machine could also run about 160 cycles per day. The operation of the machine is simple and its maintenance is also easy. It is thus found useful in the small scale sugar processing industry.

KEYWORDS : Centrifuge, Development, Performance, Brown sugar, Cottage industry

Centrifuge is one of the equipment used in processing sugar cane into sugar. It is used for separating the sugar crystal from the molasses after crystallization. All over the world, sugar centrifuges are classified into two basic categories: continuous and batch types.

The continuous type centrifugal are useful in advanced (Vaccum pan) processing technologies because of it's complicated features resulting from automation of the system. Unlike low grade continuous centrifugals, perfection of the development of a high grade continuous centrifugal was realized recently in 1995 by Australian researchers (Greig and Bellate, 1996). Previous efforts to modify low grade continuous centrifugals for curing high grade massercuits throughout the world yielded only little success (Credoz *et al.*, 1977; de Robillard and Journet, 1980; Swindlells and Kinby, 1981). According to Greig *et al.*, (1992) and Kirby *et al.* (1990), the breakthrough in the development of the prototype continuous centrifugal called STG became possible after Goodacre *et al.* (1984) suggested some possible design modifications in the existing ones to overcome the difficulties mentioned above.

Batch type centrifugals operate intermittently between cycles. These have been in the sugar industries before 1960 and are more useful in the third world because of its comparable lower sophistication (Gbabo, 1991). Chapman (1963) outlines mechanical factors such as strength, corrosion and reliability as parameters to be considered in the development of centrifugals. Hugot (1972) also had extensive study on batch type centrifugals and came up with models for predicting theoretical and practical volumetric capacities for different (Conical and Flat bottom) shapes of centrifuge baskets. Also time per cycle, gravity factor, stress per unit area of the basket due to centrifugal force, permissible thickness of material for the basket and power requirement at starting and at full speed of the machine.

Since batch type centrifugals have been observed to be more useful in the third world, engineers in the National Cereals Research Institute Badeggi undertook the development of an adoptable model for utilization in small sugar processing industries in developing countries. Accordingly, this paper highlights the design, machine features and performance of the batch type centrifugal machine that was designed and fabricated in the National Cereals Research Institute, Badeggi as one of the component equipment in the Nigeria's indegeneous brown sugar processing plant.

Design Analysis

Design analysis aimed at determining the necessary design parameters for selection of the various machine parts were carried out. This was done in order to avoid failures of machine parts during the required working life of the machine.

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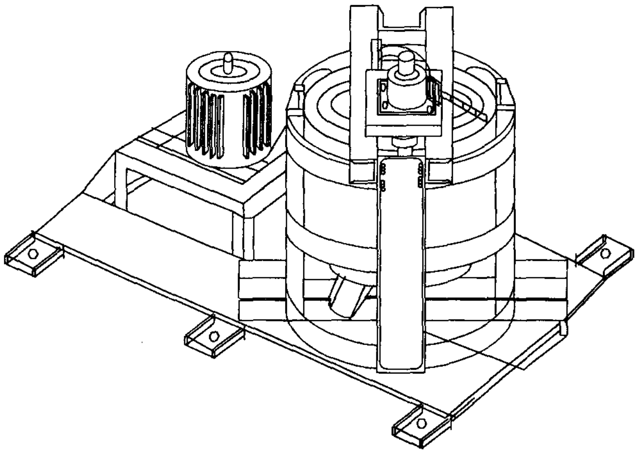

Fig. 1 : Isometric view

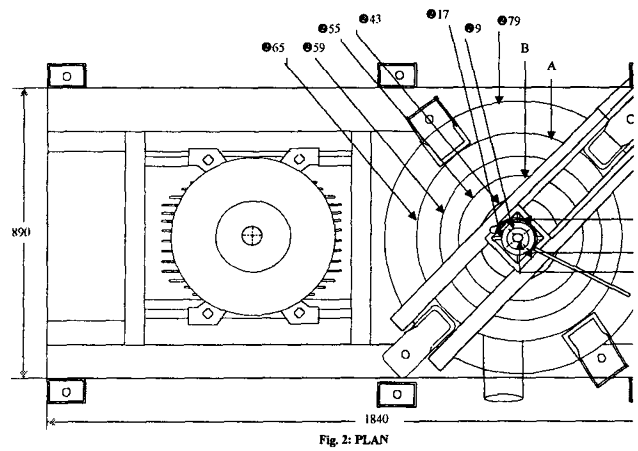

Fig. 2 : Plan

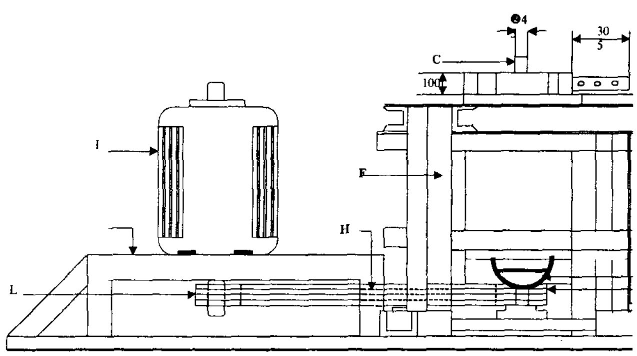

Fig. 3 : Left hand side view

Legend for Figures

A - Outer basket	G - Mollasses outlet	D_b – Diameter of inner basket
B - Inner basket	H - Belts	P_c – Central pipe
C - Central shaft	I - Electric motor	B_p – Base plate
D - Braking linkage	J – Electric motor frame	D_1 – Diameter of machine pulley
E – Braking drum	K – centrifuge pulley	D_2 – Diameter of machine pulley
F - machine frame	L – Electric motor pulley	C – Center to center spacing between Machine and motor pull
	M - Flange	

Power Requirement

The power required to drive the machine (internal basket) is a function of the mass of the internal basket, its content, flanges and the central shaft that transmits power from the electric motor to the basket through pulleys and belts as shown in Figs 5 and 7.

Hence the power required to rotate or drive the

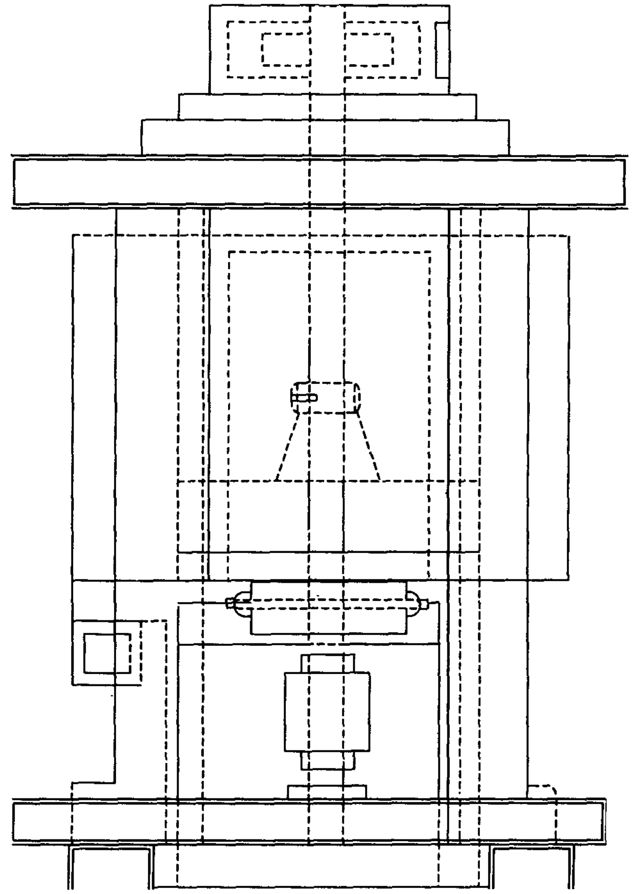

Fig. 4 : Front view

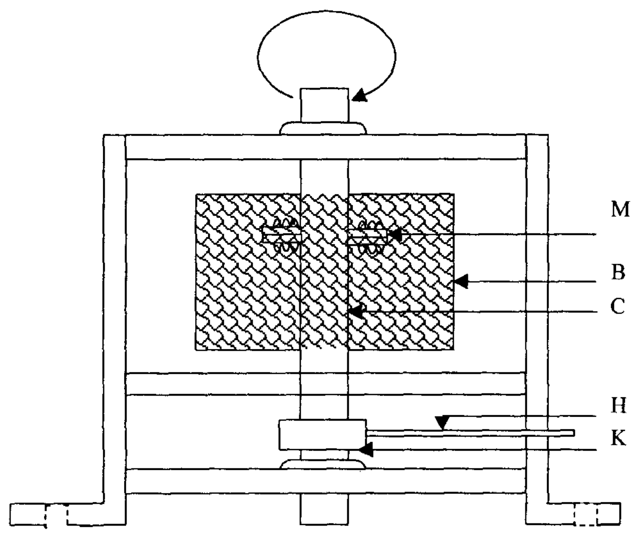

Fig. 5 : X-Sectional Sketch of Centrifuge Showing Power Transmission System

basket for separation of the sugar crystals is obtained by using the generally established equation.

$$P = F_T V \qquad \text{.............} (1)$$

Where P is the power required.

F_T is the total force of the basket, its content, the central shaft and flanges (N)

V is the velocity of the basket at full speed (m/sec).

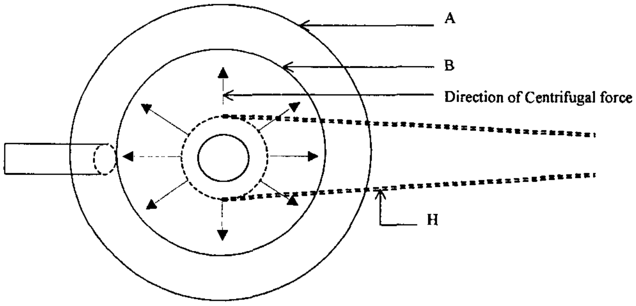

Fig. 6 : Sketch (plan) showing the Direction of Centrifugal Force

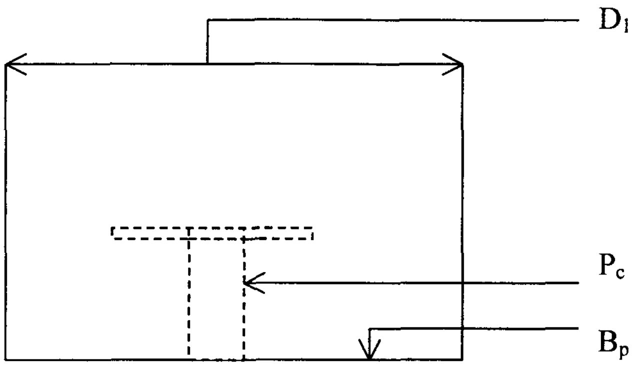

Fig. 7 : Sketch of Internal Basket

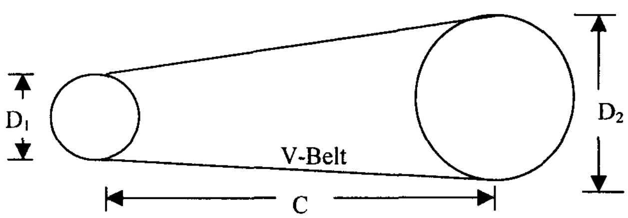

Fig. 8 : Sketch of Transmission System Using V-Belt

Centrifugal force generated in the internal basket

A centrifugal force is set up in the basket due to the rotation of the internal basket while working. In line with the principle of centrifugal force, the direction of action of the force is from the center towards the periphery of the basket as shown in fig. 6.

The centrifugal force generated was determined by applying the equation

$$Fc = \frac{M_T w^2 D}{2} \qquad \text{.............} (2)$$

where Fc is centrifugal force generated in the internal basket (N/m)

M_T is the total mass of basket, its content, shaft and the two flanges used in (Kg) coupling the internal basket to the central shaft.

W is angular velocity of internal basket (rad/sec)

D is diameter of the internal basket (m)

Stress in the internal basket

The walls of the internal basket is subjected to stress (shown in Fig. 6) due to the action of the centrifugal force. Thus the strength of the material chosen for the fabrication of the internal basket was expected to be adequate enough to withstand the expected stress in order to avoid possible basket failure while in operation.

The formula for determining the stress per unit area on the wall of the basket established by Kreg (1975) was utilized.

$$\sigma_b = \frac{M_T \omega^2 r}{\pi DH} = \frac{M_T \omega^2 D}{2\pi DH} \qquad \text{.............} (3)$$

$$= \frac{M_T \omega^2}{\pi H} \qquad \text{.............} (4)$$

where σ_b is stress on the wall of the internal basket (N/m^2)

r is the radius of the basket (m).

M_T is total mass of basket, its content and shaft assembly in kg.

H is height of basket (m).

ω and D have been defined.

Allowable tickness of material for the construction of the internal basket

In order to withstand the expected stress on the walls of the basket, an appropriate thickness of the material (stainless steel sheet) was determined to prevent avoidable basket failures. Kreg (1976) stated that the thickness of the wall of the basket to withstand the stress is a function of the unit stress that acts on the wall, the diameter of the basket and the maximum permissible stress of the material as shown below:

$$t_b = \frac{\sigma_b D}{2\sigma_p} \qquad \text{.............} (5)$$

where t_b is the thickness of the basket (m).

σ_p is the permissible stress of the material of the basket (N/m^2).

σ_b and D have been previously defined.

Twisting Moment

The high rotating speed of the shaft which is attached to the internal basket is subjected to a twisting moment. In order for the shaft not to fail, the value of the twisting moment generated is expected to be within the permissible limit in order to avoid failure of the shaft. The expression of Juvinall (1976) was used to determine the expected twisting moment of the shaft as shown below.

$$M_t = \frac{60W}{2\pi N} \qquad \text{.............} (6)$$

$= 9.55W$

N .... (7)

Where Mt is twisting moment (Nm).

N is speed of rotation of the shaft (rev. per sec.)

W is power transmitted (watts).

Torsional shear stress

The Torsional shear stress of the shaft was computed as a function of the twisting moment, diameter and second moment of area as expressed by Webb (1982):

$$T = \frac{M_t D}{2J} \quad \text{............. (8)}$$

where T is the torsional shear stress (N/m^2)

m_t is twisting moment (Nm.)

D is diameter of shaft (m)

J is polar moment of area (m4) = $\frac{\pi d^4}{32}$

By substituting J = $\frac{\pi d^4}{32}$ in equation .... (8)

$$T = \frac{16 m_t}{\pi d^3}$$

Bending stress

The bending stress at any section of the shaft was calculated once the bending moment diagram was obtained from the usual bending stress equation:

$$\sigma_b = \frac{M_y}{d^3} \quad \text{............. (9)}$$

Where σ_b is bending stress at section under consideration (Nm).

M_y is bending moment at section under consideration (Nm)

$Y = \frac{d}{z}$ is distance from axis of shaft (m)

I is second moment of area = $\frac{\pi d^4}{64}$ for solid shaft (m^4).

Substituting I = $\frac{\pi d^4}{64}$ in equation .... (9)

$$\sigma = \frac{32\ M}{\pi d^3} \quad \text{............. (10)}$$

Radial deformation in shaft

The shaft that transmits power to the internal basket was expected to experience deformation due to the torsional shear stress it is subjected. The extent of distortion (deformation) depends on: the torsional shear stress, length, modulus of rigidity and radius of the shaft (Ryder 1982).

$$\phi = \frac{TL}{Gr} \quad \text{............. (11)}$$

Where f is the radial deformation, shear strain or angle of twist.

T is the maximum torsional shear stress of the shaft (N/m^2).

L is the length of the shaft (m).

G is the modulus of rigidity (N/m).

R is the radius of the shaft (m).

Diameter of Shaft

The diameter of the shaft to transmit power to the internal basket is dependent on the twisting moment (Torque) on the shaft and the permissible shear stress of the material of make (stainless steel) of the shaft as shown (Holman 1969).

$$d_s = \frac{(16\ m_t\ d_b)^{0.33}}{(\pi \rho_s)} \quad \text{............. (12)}$$

Where ds is diameter of shaft.

m_t is twisting moment (Torque) on the shaft developed due to the rotation of the internal basket fastened onto the shaft with the aid of a lock bolt (Nm).

ρ_s is permissible shear stress of stainless steel = 115×10^6 N/m2 (Anon, 1978 and 1979)

d_b is diameter of internal basket (m).

Pulleys and belt for power

transmission V-belts and pulleys were used to transmit power from the electric motor to the machine due to their ability to absorb vibration, suitability for transmitting power at relatively larger distance, quietness while in operation, long trouble free life span, and easy detection of fault.

Length of belt and angle of contact

The length of belt required to transmit power from the motor to the machine is a function of the diameters of the two pulleys and center to center distance between the two pulleys and is expressed by the following established formulae.

$$L_T = \frac{\pi}{2}(D_2 + D_1) + 2C + \frac{(D_2 - D_1)^2}{4} \quad \text{...... (13)}$$

where L_T is total belt length required in m.

D_1, D_2 and (have been defined).

The angle of contact between the belt and pulley depends on the diameters of the two pulleys (drive and driven), the center to center distance between the two pulleys and the orientation or mode of usage of the belt (either open or cross). In this design, open belt is employed (Fig. 9) for the transmission of power from the electric motor to the machine because of the absence of rubbing action of the two sides of the belt that might cause wear while working unlike the cross belt hence would ensure durability.

The angles of contact of the belt with the drive and driven pulley is determined from the equation.

$$\phi_1 = 180^\circ + 2\sin^{-1} \frac{(D_2 - D_1)}{2c} \quad \text{............. (14)}$$

$$\phi_2 = 180^\circ - 2\sin^{-1} \frac{(D_2 - D_1)}{2c} \quad \text{............. (15)}$$

where ϕ_1 is the angle of contact between the belt and motor pulley (degrees).

ϕ_2 is the angle of contact between the belt and machine pulley (degrees).

D_1 is the diameter of motor pulley in m.

D_2 is the diameter of machine pulley in m.

c is the center to center distance between the motor and machine pulleys (m).

Performance Assessment

The machine was tested to assess it's efficiency in separating massercuit into sugar crystal and molasses.

MATERIALS AND METHOD

Four samples of sugar massercuit weighing 8.0 kg, 10.0 kg, 12.0 kg and 14.0 kg in three replications were introduced into the centrifuge. The machine was put on for a period of 45 seconds to separate the massercuit into sugar crystal and molasses. The spinning period of 45 seconds was chosen because preliminary tests conducted on the machine shows that complete purging of molasses from the massercuit was accomplished within 30-35 seconds. The wet sugar crystal and molasses were weighed with a weighing balance (BAW 7210). The sugar was then dried with a laboratory electric rotary dryer from 55% to 10% moisture content. The weight of the dried sugar was recorded and the quantity of evaporated moisture and centrifugal efficiency of the machine were computed as follows:

(1) Mass of evaporated moisture :

$$Mm = M_T - Md \quad \text{.............} \ (16)$$

Where Mm = Mass of evaporated moisture (kg)

M_T = Total Mass of moist sugar (kg)

Md = Mass of dried sugar (kg).

(2) Centrifugal efficiency

$$\eta_{cy} = \frac{Me}{Me + mm} \times 100\% \quad \text{.............} \ (17)$$

Where Mo = Mass of molasses (kg)

Mm and M_T have been previously defined.

RESULTS AND CONCLUSION

The result of the centrifugation test is shown in Tables 1 and 2. It indicates that the centrifugal efficiency of the machine decreased with increasing load. Also some quantity of aggregated sugar were obtained for the 12kg and 14kg masseercuit. In addition, some vibration tendency of the machine was clearly observed for the higher levels of sugar massercuit. It is the vibration tendency of the machine at the 12kg and 14kg load that must have been responsible for the lower centrifrugal efficiency because the centrifugal force developed under this vibratory condition are distorted. Thus less centrifugal force is generated to expel the void and bound molasses between and within the sugar crystals. From this observation, a maximum massercuit load of 10kg is found to be more suitable for the operation of the centrifuge. Also, considering a maximum sugar

Table - 1 : Centrifugation data

Mass of massercuit (kg)	Mass of moist sugar (kg)	Mass of molasses (kg)	Time taken to separate sugar	Mass of dried sugar (kg)	Mass of evaporated moisture during drying
8.0	4.0	3.9	128	3.52	0.48
8.0	4.2	3.6	135	3.77	0.43
8.0	3.9	3.8	131	3.5	0.4
10.0	5.6	4.2	120	5.1	0.5
10.0	5.2	4.7	140	4.7	0.5
10.0	5.5	4.3	125	4.9	0.6
12.0	7.0	4.8	122	6.0	1.0
12.0	7.3	4.5	130	5.8	1.5
12.0	7.1	4.8	132	6.2	0.9
14.0	9.0	4.8	118	7.3	1.7
14.0	9.01	4.8	123	7.0	2.1
14.0	8.8	5.0	136	7.3	1.5

Table - 2 : Average Values of Centrifugation data and machine efficiency

Mass of massercuit (kg)	Mass of moist sugar (kg)	Mass of mollasses (kg)	Sugar Discharge period (sec)	Mass of dried sugar (kg)	Mass of evaporated sugar (kg)	Total Mass of mollasses and moisture (kg)	Centrifugal efficiency (%)
8.0	4.03	3.77	3.6	131.33	0.48	4.08	93.4
10.0	5.43	4.4	4.9	128.3	0.53	4.93	89.25
12.0	7.13	4.7	6.0	128.0	1.13	5.83	80.62
14.0	8.96	4.8	7.2	125.7	1.77	6.64	73.3

discharge period of 136 seconds the machine can run 160 cycles per day of 8hrs working period. At this rate, about 800kg of sugar can be produced per day.

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