DESIGN OF RAM PRESS FOR PALM OIL MILLS

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ABSTRACT

A study carried out by the researcher at Rison Palm oil mill, Uhima-Rivers State shows that the press unit constitutes the heart of metal wear and breakdown in the mill. The press screws wear out rapidly. Unfortunately, the cost of importing the screws is extremely high, far beyond the reach of the local mills. Various attempts (reported elsewhere) have been made by the researcher to address the problem. The ram press provides a good alternative with comparatively less wear and breakdown problems, especially for small and medium level oil mills. A typical ram press design for a throughput capacity of 2.6976E-3 m³/s is hereby presented. Available local engineering materials were considered in all cases. Detailed design specifications are also provided.

Keyword: Palm oil mill, ram, design capacity, materials.

INTRODUCTION

The re-occurring problems of machine breakdown in the nation’s industries, and in the palm oil mills in particular have become an aspect of concern especially when one considers the contributions of this sub-sector to the national economy. Nigeria is blessed with abundant natural and human resources with population largely dependent on agriculture. Her economic survival therefore depended largely on agriculture and palm oil in particular, until the discovery of petroleum in the nation’s soil. Nigeria was until then, the world’s largest producer of palm oil despite the predominantly local technologies available in the country at that time. These technologies had required less maintenance cost owing to the associated minimal wear and breakdown problems (Hartley, 1977). With the growing world population however, the demand for oil commodities progressively increased and more advanced technologies were inevitably needed for increased bulk production. These sophisticated imported equipment however placed a stringent demand for high maintenance skills - a culture which was very lacking in our being. This resulted to incessant disruption of production such that the mills could only offer epileptic service and in extreme cases resulted to total breakdown. The use of power screws (which is more common) for power transmission is generally a combination of any of the following relative motion between the screw and the nut (Remy, 2004). These are:

1. Nut motion is translational and rotational, while screw is fixed.
2. Nut has only rotational motion while screw has translational without rotation.
3. Nut is fixed and screw has translational and rotational motion.
4. Nut has only translational motion, while the screw has only rotational.

In this case however, it is desired that the mash remains stationary while the screw is in a repeat reciprocating motion. This obviously does not satisfy any of the above conditions. Condition 3 above best explains the relative motion in a typical screw press. A combination of translational and rotational motion is known to induce more wear in the system. Also the translational force output is said to be equal to the product of the mechanical advantage and the rotational input force (Barbashov, 1984). The ram press, on the other hand, has been associated with less wear and breakdown problems. Maintenance cost is low and with less demand for skilled labour.

THE DESIGN CONCEPT

Figure 1.0 (last page) shows the assembly drawing of the press. The press cake is let into the ram chamber via the inlet orifice just as the reciprocal action of the ram allows the press cake to fall by gravity into the ram chamber when it (the ram) is at the extreme top position. The ram gradually comes down as the disc rotates thereby closes the cake inlet and presses on the cake already entrapped in the ram chamber. The ram action aids to further squeeze out any oil still entrapped in the cake. The liberated oil escapes via perforated casing into the oil outlet while the press cake is gradually released through the discharge gate at the springs compress after a certain pressure built-up. The springs return as the pressure releases thereby ensuring a continuous pressing action.
DESIGN ANALYSIS

Volumetric flow rate (or throughput capacity) \( = 2.6976 \times 10^6 \text{ m}^3 /\text{s} \)

INLET ORIFICE

Volumetric flow rate into the ram chamber \( = 2.6976 \times 10^6 \text{ m}^3 /\text{s} \)

Volume per second \( = 0.05 \times 0.2 \times 1 = 2.6976 \times 10^6 \text{ m}^3 /\text{s} \)

\( V = 2.6976 \times 10^6 \text{ m}^3 /\text{s} \times 0.05 \times 0.2 = 0.2678 \text{ m} \)

For a continuous flow system, the orifice size should be increased by 20%; hence;

\( I = 0.2678 \times 1.2 = 0.320 \text{ m} \)

THE RAM

Considering the following crank mechanism arrangement:

weight of ram + rod = \( w_1 + w_2 = W \)

rack radius, \( r = \) radius of rotating disk = 0.21 m

The ram has a simple harmonic motion represented by the following displacement – time equation (Bedford, 1995):

\[ x = r \cos \omega t \quad \text{ (1) where} \]

vertical distance moved by the ram, \( m \); \( \omega = \) angular velocity of rod, \( \frac{2 \pi}{60} \text{ rad/sec} \)

A rotational speed of the disk, rpm

from equation (1):

velocity, \( v = \frac{dx}{dt} = \omega r \sin \omega t \)

acceleration, \( a = \frac{dv}{dt} = -\omega^2 r \cos \omega t \)

resultant active force acting on the ram, \( F = Wz / x \times \omega / r \cos \omega t \)

It shows that the resultant active force acting on the ram is a function of time. When the ram is in the extreme top position, the angle \( \omega t \) becomes equal to \( \pi \); where \( i = \) odd integer

at extreme top position; \( \cos \omega t = -1 \) and \( F = Wz / x \omega \)

hence, \( F = 0 \)

Thus, maximum resultant force \( = Wz / x \omega \)

minimum resultant force \( = 0 \)

at result (Odionye, 2002) shows that 32.37 kPa is required to crush and press out oil from palm kernels. The compression pressure on the matrix should not exceed about 60% of this value as to ensure maximum recovery of the palm kernels.

Taking a conservative value of 60% of the above result;

ign impact pressure \( = 32.37 \times 0.6 = 19.422 \text{ kPa} \)

act pressure = impact force / ram surface area, let ram diameter = 0.460 m

Hence, \( Fz = \pi d / 4 = \pi (0.460) / 4 = 0.166 m \)

IMPACT FORCE ON THE RAM:

weight of ram, \( w_1 = \) mass of ram \( \times 9.8 \times 10^6 \text{ kg/m} \)

average density of ram material, \( \rho = 7.8 \times 10^6 \text{ kg/m} \)

volume, \( v = (\pi d / 4) \times \pi \) where \( \pi = \) ram thickness, 0.120 mm

\( \pi (0.46) / 4 \times 0.120 = 0.02 m \); \( \pi = 7.8 \times 10^6 \times 0.02 = 156 \text{ kg} \)

\( w_1 = m = 156 \times 9.81 = 1530.36 \text{ N} \)

weight of rotating disk; resultant active force on the ram,

\( Fz = (Wz / \omega) \times \cos \omega t \quad \text{ (2)} \)

\( w = w_1 + w_2 \) \( w_2 = \) weight of rod = \( m \)

\( \rho = \)
\[ v = \frac{\pi d}{4} \times 1 \quad \text{where} \quad l = \text{length of rod} ; \quad d = \text{diameter of rod} \quad 0.060 \text{ m} \]
\[ v = \pi (0.06) \times 4 \times 0.420 = 0.001188 \text{ m/s} ; \quad m = 7.8 \times 10^{-2} \times 0.001188 = 9.263 \text{ kg} \]
\[ W = 9.263 \times 9.81 = 90.87 \text{ N}; \quad \therefore W = 1530.56 + 90.87 = 1621.25 \text{ N} \]

From equation 2:
\[ \omega = \frac{(3.23 \times 10^3) \times 9.81}{1621.25 \times 0.21} = 93.068 \]
\[ \omega = \sqrt{(93.068)} = 9.65 \text{ rad/sec} \quad \text{but} \quad \omega = 2\pi n/60 \]

where \( n \) = rpm of rotating disk; \( n = (60 \times \omega)/2\pi = (60 \times 9.65)/2\pi = 92 \text{ rpm} \)

**POWER TRANSMISSION**

Figure 2.0 shows the power transmission network of the press.

![Power Transmission Network](image)

Electric Motor Speed ---- 2800 rpm;  Required Speed ---- 92 rpm

**SPEED REDUCTION BY PULLEYS**

Drive Pulley Diameter, \( D_1 = 0.080 \text{ m} \):
Diameter of Reduction Pulley, \( D_2 = 0.280 \text{ m} \)

From \( N_1 D_1 = N_2 D_2 \) where \( N_1 \) = rpm of motor; \( N_2 \) = required rpm
\[ N_2 = N_1 \frac{D_1}{D_2} = 2800 \times 80/280 = 800 \text{ rpm} \]

**CENTER DISTANCE:** \( D_2 \leq C \leq 3 \times (D_1 + D_2) \) (Shigley, 1989). Where
C = centre distance between the two pulleys; \( D_1 \) = diameter of small pulley, 0.080 m
\( D_2 \) = diameter of larger pulley, 0.280 m
C \leq 3 \times (80 + 280) \leq 1080 \text{ mm}; \quad \text{let} \quad C = 0.700 \text{ m} 

**LENGTH OF TRANSMISSION BELT**

Total length of belt, \( L_b = [4C - (D_2 - D_1)] + \frac{1}{2} (D_2 + D_1) \)
Where \( D_1 \) and \( D_2 \) are the drive belt angles of contact on the smaller and the larger pulleys respectively
\[ D_1 = 180 - 2 \sin \left( D_2 - D_1 / 2 \right) = 180 - 2 \sin \left( 90 - 45 \right) = 163.57 \text{ cm} \]
\[ = 163.57 \text{ cm} = 1.63 \text{ m} \]
\[ D_2 = 180 + 2 \sin \left( D_2 - D_1 / 2 \right) = 180 + 2 \sin \left( 90 - 45 \right) = 186.55 \text{ cm} \]
\[ = 186.55 \text{ cm = 1.87 m} \]
\[ L_b = [4(700) - (280 - 80)] + \frac{1}{2} (280 \times 3.26 + 80 \times 2.86) \]
\[ = \sqrt{(1960000 - 40000)} + 570.8 = 1956.44 \text{ mm = 1.96 m} \]

**SPEED REDUCTION BY GEARS**

Using bevel gear transmissions with shaft axes intersecting at right angles;
Pinion shaft speed = 800 rpm

1st Stage: Let number of teeth of pinion, \( T_1 = 20 \)
number of teeth of gear, \( T_2 = 36 \); speed of pinion, \( N_1 = 800 \text{ rpm} \)
speed of gear, \( N_2 = N_1 \frac{T_1}{T_2} = 800 \times 20/36 = 445 \text{ rpm} \)

2nd Stage: \( T_1 = 20 \text{ teeth, } T_2 = 36 \text{ teeth} \)
\( N_1 = 445 \text{ rpm}; \quad N_2 = 445 \times 20/36 = 248 \text{ rpm} \)

3rd Stage: \( T_1 = 20 \text{ teeth, } T_2 = 36 \text{ teeth} \)
\( N_1 = 248 \text{ rpm}; \quad N_2 = 248 \times 20/36 = 138 \text{ rpm} \)
Design of Ram Press For Palm Oil Mills

\[ T1 = 20 \text{ teeth; } N1 = 138 \text{ rpm} : N2 = 92 \text{ rpm} \Rightarrow \text{required speed} \]
\[ \therefore T2 = N1 \times T1 / N2 = 138 \times 20 / 92 = 36 \text{ teeth} \]

gear arrangement in the gear box is as shown in Figure 3.0.

Fig. 3.0  Power Transmission Gear Box

DESIGN

First two gears, we first determine which of the mating gears is weaker.

Strength is a function of the product \( S \times V \)

\( S \) = endurance limit of the gear material (approximately \( = 1/3 \) of the ultimate strength), 85 MN/m²

\( V \) = form factor

Number of teeth of gear, \( Nf = Ng \times (Np + Ng) / Np \)

\( Np \) = number of teeth of pinion; \( Ng \) = number of teeth of the gear

\[ 20 \times (20 + 36) / 20 = 74.13 \]

Bond (Hall, 1980);

0.115 for a 14 1/2 full-depth composite gear profile

Number of teeth of the pinion;

\[ Np \times (Np + Ng) / Ng = 20 \times (20 + 36) / 36 = 22. \]

Bonding to \( y = 0.093 \)

\[ (21) = 8.5 \times 0.115 = 9.78E6 \]

\[ (21) = 8.5 \times 0.093 = 7.86E6 \]

If \( S \) is smaller for the pinion gear, hence design is based on the pinion since it is the weaker.

Fitness for and in accordance with American Gear Manufacturers Association (AGMA) durability requirements;

The power transmitted \( 70.8CB \) b; where \( b = \) face width

\[ = \left[ \frac{Dp}{Np} \right] \times 0.032 \left( \frac{5.6}{5.6 (v)} \right) \]

Where \( v = \) pitch velocity

Fit of pinion, \( Dp = Tp \times m \)

Where \( Tp = \) number of teeth of pinion; \( m = \) module

On the pinion,

\[ 5 \times 60 / 2 \pi \times Np = 7 \times 10 \times 60 / 2 \pi \times 800 = 83.56 \text{ N-m} \]

Initial load on the pinion; \( F_t = 2T / Dp = 2T / m \times Tp \)

\[ = 2 \times 83.56 \text{ /m x 20} = 83.56 / \text{m} \cdot \text{N} \]

Height of pitch cone element (or slant height of the pitch cone);

\[ = Dp / 2 \sin \theta = m \times Tp / 2 \sin \theta \]

Shafts are at right angles, pitch angle for the pinion,

\[ \theta = \tan \theta = 1.8 \]

\[ \Rightarrow \theta = 29.05 \; \therefore L = \text{m x 20} / 2 \sin (29.05) = 20.59 \text{m} \]

\[ \theta = \text{slip} / \theta / 4 = 5.147 \times \text{module} \]

Form factor \( = 0.124 - 0.686 / \text{Nf} \)

Form factor for the pinion; \( \gamma_p = 0.124 - 0.686 / 22.88 = 0.094 \)

Initial load on the pinion,

\[ = S \times m \times \gamma_p (L \times \gamma_p / L) \text{ where } S = \text{allowable working stress} \]

39
\[ m = 3 \times \frac{8.356 + 0.625 \pi 7}{0.0045} = 0.0045 \text{ meters} = 0.0045 \text{m} \]

Thus, let gear module \( m = 0.006 \text{m} \)

- Diameter of pinion, \( D_p = x \times m = 20 \times 6 = 0.120 \text{m} \)
- Diameter of gear, \( D_g = T_g \times m = 36 \times 6 = 0.216 \text{m} \)
- \( L = 20.59 \times m = 20.59 \times 0.006 = 0.124 \text{m} \)
  - Also \( L / 4 \leq b \leq L / 5 \)
  - \( 123.54 / 4 \leq b \leq 123.54 / 3 \)
  - \( b = 30.89 \leq b \leq 41.18 \)
  - Use \( b = 0.04 \text{m} \)

Pitch line velocity:
\[ V = \frac{D_p}{2} \left( N_p \times 2 \pi \times 60 \right) = 0.120 / 2 \times 800 \times 2 \pi / 60 = 5.03 \text{ m/s} \]

- \( CB = \frac{\left( 0.120 \right)^2}{12 \times 2^3} \times 800 \times 0.032 \times 5.6 / 5.6 + \sqrt{5.03} = (10.3923)(0.714) = 7.42 \text{ Thus, } C_m = 7 / 0.8 \]
- \( 742 \times 0.04 = 3.0 \)

Thus, the selected material requires a minimum Brinell hardness of 210 for the pinion and 160 for the gear in order to ensure a durable service (Hall, 1980).

### Gear Specifications

<table>
<thead>
<tr>
<th>Number of Teeth</th>
<th>Pinion</th>
<th>Gear</th>
</tr>
</thead>
<tbody>
<tr>
<td>Speed</td>
<td>800 rpm</td>
<td>445 rpm</td>
</tr>
<tr>
<td>Diameter</td>
<td>120 mm</td>
<td>216 mm</td>
</tr>
<tr>
<td>Face Width</td>
<td>40 mm</td>
<td>40 mm</td>
</tr>
<tr>
<td>Pitch Angle</td>
<td>29.05</td>
<td>90 - 29.05 = 60.95</td>
</tr>
<tr>
<td>Length of Pitch Cone</td>
<td>123.54 mm</td>
<td>123.54 mm</td>
</tr>
<tr>
<td>Material</td>
<td>Steel</td>
<td>Steel</td>
</tr>
</tbody>
</table>

### Shaft Design

\[
W_T = \text{total weight acting at shaft end } D = \text{weight of (drive disk + rod + ram)}
\]

### Drive Disk

<table>
<thead>
<tr>
<th>Use</th>
<th>Diameter</th>
<th>Thickness</th>
<th>Material</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.400 m</td>
<td>0.050 m</td>
<td>Mild Steel</td>
</tr>
</tbody>
</table>

\[ w_3 = m \]
\[ m = 0.22 \times 9.81 = 149.112 N \]
\[ W_T = w_1 + w_2 + w_3 = 1550.56 + 90.87 + 149.112 = 1.771 kN \]
\[ w_1 = \text{weight of gear} \]
\[ \text{diameter} = 0.216 \text{m} \]
\[ \text{face width} = 0.040 \text{m} \]
\[ 216 - 2(40) = 136 \]

\[ \text{surface area} = \pi (0.216) / 4 - \pi (0.136) / 4 = 0.022 \text{ m} \]
\[ \text{volume} = 0.022 \times 0.022 \times 0.124 = 0.00274 \text{ m} \]
\[ m = 0.022 \times 7.8 \times 10 = 171.6 \text{ kg} \]
\[ W_4 = 171.6 \times 9.81 = 1.684 \text{ kN} \]
**Design of Ram Press For Palm Oil Mills**

**BEEN BODY DIAGRAM**

\[
\begin{align*}
B & = 1683.4 \text{ N} \\
A & = 1770.34 \text{ N} \\
D & = 770.34 \text{ N}
\end{align*}
\]

Where \( R_A \) and \( R_B \) are reactions at bearing supports.

\[\text{Lx} = 0, \quad R_A + R_B = 1770.34 + 1683.4 = 3.454 \text{ kN}\]

\[\text{LMA} = 0, \quad 1770.34 (0.23) + 0.3 R_B - 1683.4 (0.42) = 0\]

\[0.3 R_B = 1683.4 (0.42) - 1770.34 (0.23) = 707.03 - 407.178 = 3.00 \text{ kN}\]

\[R_B = 299.85 / 0.3 = 1000 \text{ kN}; \quad R_A = 3453.74 - 999.50 = 2.454 \text{ kN}\]

It is easily observed from the force diagram that the shear force at point A is zero, thereby indicating point of maximum bending moment (Allen, 1980).

Thus, bending moments about A:

\[1770.34 (0.23) = 407.18 \text{ N} \]

\[R_B (0.3) = 999.50 (0.3) = 299.85 \text{ N}\]

\[1683.4 (0.42) = 707.03 \text{ N}\]

Maximum bending moment, \( M = 707.03 \text{ N-m}\)

The weight developed by the shaft (Allen, 1980):

\[t = 9550 \times K \times \text{rpm} = 9550 \times 6.6 / 92 = 685.11 \text{ N-m}\]

The popular shaft equation:

\[d = 16 / (\pi s) \sqrt{(K_b M) + (K_t T)}\]

\[s = \text{allowable shear stress}; \quad s = 5.516 \times 10^6 \text{ N/m}\]

\[K_b = \text{shear factor for bending moment}; \quad 1.7\]

\[K_t = \text{shear factor for torsional moment}; \quad 1.3\]

\[d = 16 / (\pi s) \sqrt{(1.7 \times 707.03) + (1.3 \times 685.11)} = 1444678.03 + 793244.95 = 0.0001368 \text{ m} = 0.050 \text{ m}\]

**RING SELECTION**

**Diameter** --- 0.050 m

**Shaft Speed** --- 92 rpm

**Linear Speed, \( v = \pi DN/60 = \pi (0.050) \times 92 / 60 = 0.24 \text{ m/s}\)**

**AD CALCULATIONS**

Weight of shaft:

\[s = \rho v \quad \text{but} \quad v = \pi D \times (0.05) = 0.00127 \text{ m}\]

\[76.80 \times 10 \times 0.00127 = 98 \text{ kg}\]

Weight of driveshaft:

\[98 \times 9.81 = 961.55 \text{ N}\]

Weight on shaft:

\[V_1 + V_2 + W_3 + W_4 = WT + W_4 = 1770.34 + 270 = 2.04 \text{ kN}\]

Shear force value is mainly due to radial load on the shaft. Hence a radial or journal bearing is selected quite suitable for the design

**Table (Tyler, 1986):** choosing a bearing sleeve material of porous iron:

<table>
<thead>
<tr>
<th>Minimum rated load</th>
<th>80,000 psi</th>
<th>5.5 E07 N/m</th>
</tr>
</thead>
<tbody>
<tr>
<td>Minimum speed</td>
<td>600fpsm</td>
<td>4.16 m/s</td>
</tr>
<tr>
<td>Minimum operating temperature</td>
<td>176E06 N/m-s</td>
<td>65.6 C</td>
</tr>
</tbody>
</table>

For a maximum operating pressure of 4.137 E06 N/m (600 psi),

| Length of bearing sleeve, \( l = L / (p_d) \) where \( L = \text{load} \)|
|--------------------------|-----------|--------|
| Maximum pressure load    | \( d = \text{shaft diameter} \)|
| 0.34 / 4137 x 10         | 0.05     | 0.00986 m |

Value of a bearing is given by

\[V = \text{operating bearing pressure} \times \text{shaft surface speed} \]

\[4137 \times 10 \times 0.24 = 993 \times 10 \text{ N/m-s}\]
This gives a safe value for the chosen bearing material since it is obviously less than the recommended pv limit of 1760 x 10^6 N/m-s.

<table>
<thead>
<tr>
<th>Bearing Selection for a Force Fit</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Use</strong></td>
</tr>
<tr>
<td>Material</td>
</tr>
<tr>
<td>Bore</td>
</tr>
</tbody>
</table>

**SIZE OF PRESS CHAMBER**

If the press cake discharge gate opens at 2 minutes intervals:

Volume of mash in the ram chamber after 2 minutes

\[ = 2.6976 \times 10^6 \times 60 \times 2 = 0.324 \text{ m}^3 \]

Let height of ram chamber from discharge gate to the matrix inlet = \( h \)

Volume \( = \pi D \cdot h \cdot l / 4 = 0.324 \text{ m}^3 \)

\( h \cdot l = 4 (0.324) / \pi (0.80) = 0.645 \text{ m} \)

**CONCLUSION / RECOMMENDATION**

Design of ram press of any capacity can easily be done with the method specified in the work. Ram press is recommended for small-to-medium sized palm oil mills. This is because the oil extraction efficiency of ram press is found to be comparatively lower. It is advisable that material be based on what is locally available. Sliding or contact surfaces should be surface treated for hardness.

**REFERENCES**
