Design, Improvement and Thrust Bearing Analysis of Oil Expeller Machine

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Abstract

In this paper we will design and make improvement of some of the components of the oil expeller machine also carried out the thrust ball bearing analysis and results are compared with analytical results.

Key Words: Oil Expeller Machine, Design, Thrust Ball Bearing, Ansys, Pro-E

1. INTRODUCTION

Screw type oil presses are advanced oil processing machinery, characterized by their high oil output rate with good quality, simple design, easy to use and continuous operation. They can use for various raw materials, such as peanut, beans, rape seeds, cotton seeds, sesame, sunflower seeds, copra etc.

According to the online resources, Bamgboye and Adejumo \cite{1} developed an expelling machine for extracting oil from decorticated sunflower seeds. The machine was tested at auger speeds of 30, 40, 50 rpm respectively and three throughputs. Results showed that performance efficiencies increased with auger speed and throughput. Expelling efficiency of over 70\% was obtained.

Alessandra Maria Sabelli \cite{2} represented a first step into the development of an extraction tool that maximizes the extraction of oil from moringa seeds, and consequently the consumption of the seeds themselves, not exploited so far. The author designed the press and carried out an experiment to investigate the effect of pressure applied on seed vs. oil extracted (i.e. efficiency of the oil press).

The uses of various investigational techniques were described and illustrated with examples of work by D. Scott of the National Engineering Laboratory. \cite{3} Also methods of diagnosis of impending bearing failure and its prevention are outlined.

The main objective of this work is to design and improvement by replacing the existing bearing. Also carry out analytical and software analysis of thrust ball bearing used in machine.

2. PRODUCT DESIGN

![Fig.1. Double Chamber Oil Expeller Machine](image1)

![Fig.2. Location of Thrust Ball Bearing](image2)

<table>
<thead>
<tr>
<th>Nomenclatures</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>TPD</td>
<td>Tones per Day</td>
</tr>
<tr>
<td>N</td>
<td>Revolutions per Minute Main Shaft (RPM)</td>
</tr>
<tr>
<td>N\textsubscript{1}</td>
<td>Revolutions per Minute Motor shaft (RPM)</td>
</tr>
<tr>
<td>N\textsubscript{max}</td>
<td>Maximum rotational speed</td>
</tr>
<tr>
<td>P</td>
<td>Motor Power</td>
</tr>
<tr>
<td>p</td>
<td>Pressure</td>
</tr>
<tr>
<td>D\textsubscript{1}</td>
<td>Motor Pulley Diameter</td>
</tr>
<tr>
<td>D\textsubscript{2}</td>
<td>Gear Box Pulley Diameter</td>
</tr>
<tr>
<td>T</td>
<td>Main Shaft Torque</td>
</tr>
<tr>
<td>\tau</td>
<td>Shear Stress</td>
</tr>
<tr>
<td>F\textsubscript{a}</td>
<td>Axial or Thrust Force</td>
</tr>
<tr>
<td>M</td>
<td>Minimum axial load factor</td>
</tr>
<tr>
<td>C\textsubscript{dyn}</td>
<td>Dynamic Capacity of Bearing</td>
</tr>
</tbody>
</table>
2.1 Problem Description
1. Thrust ball bearing of m/c fails once in 2 to 3 months.
2. Type of failure It breaks into pieces

2.2 Technical Details
These details are taken from the machine catalogue
1. Steam rate 2 Kg/Cm²
2. Quantity to be crushed 25 to 30 TPD
3. Main Shaft rotation \( N = 22 \) RPM
4. Motor rotation \( N_1 = 940 \) RPM
5. Motor Capacity 85 HP
6. Thrust bearing No (ZKL Company) 51326 M
7. Pr. required expelling oil from mustard seeds 10 MPa to 15 MPa.
8. Motor Pulley Dia. \( D_1 = 240 \) mm
9. Gear Box Pulley Dia. \( D_2 = 1000 \) mm

2.3 Calculate Gear Ratio and Verify Main Shaft Rotation
- Find input speed for Gear Box
  For belt drive
  \[
  \frac{D_1}{D_2} = \frac{N_1}{N_2} = \frac{240}{1000} = 0.24 \Rightarrow N_2 = 225.6 \text{ RPM}
  \]
  For Gear Drive
  \[
  \frac{T_1}{T_2} = \frac{D_1}{D_2} = \frac{22}{18} = \frac{11}{9} \Rightarrow \text{Gear Ratio } G = 3.5
  \]
  Similarly, \( G_2 = \frac{T_3}{T_4} = \frac{63}{18} = 3.5 \)
  Gear Ratio \( G = G_1 \cdot G_2 = 3 \cdot 3.5 = 10.5 \)
  Main Shaft Rotation \( N = \frac{I/P \text{ speed at Gear Box} \cdot \text{ Gear Ratio } G}{G} = 225.6 \cdot 10.5 \)
  \( N = 21.48 \text{ RPM} \)
  Say \( N = 22 \text{ RPM} \)

2.4 Main Shaft Design
1 HP = 0.754699 KW
Motor Power \( P = 85 \) HP
\( P = 64.15 \) KW

\[
\text{Power } P = \frac{2 \pi NT}{60}
\]
\[
64.15 \cdot 10^3 = \frac{2 \pi \cdot 22 \cdot T}{60}
\]
\( T = 27.84 \cdot 10^3 \) N m = 27.84 \( \times 10^6 \) N mm

Shear Stress for 45C8 Steel \[20\]
\( \tau = 500 \text{N/mm}^2 \)
Say \( \tau = 150 \text{N/mm}^2 \)

\[
\text{Now Torque } T = \frac{\pi \cdot d^3 \cdot \tau}{16} = \frac{\pi \cdot 110^3 \cdot 150}{16} = 27.84 \cdot 10^6 \text{ N mm}
\]
Main Shaft Dia. \( d = 98.157 \) mm
Say \( d = 110 \) mm

2.5 Small Chamber Design \[4\]
\( T = 27.84 \cdot 10^6 \text{N mm} = F \cdot r = F \cdot 55 \)
$F = 506.18 \times 10^3 \text{N}$

$\frac{Pr.}{A} = \frac{F}{\pi d_h L} = 140 \text{N/mm}^2 = \frac{506.18 \times 10^3 \text{N}}{\pi \times 140 \text{L}}$

Where, $d_h = \text{Warm’s core diameter}$
$Pr. = \text{Pressure required to expel oil from mustard seeds 10 MPa to 15 MPa.}$
Hence taking $Pr. = 14 \text{ Mpa} = 140 \text{ N/mm}^2$ ($\because 1 \text{ MPa} = 10 \text{ N/mm}^2$)
Length of Chamber $L = 822.4 \text{ mm}$
Take $L = 850 \text{ mm}$

$\frac{Pr.}{A} = \frac{F}{\pi D (D - d_h)} = 140 \text{ N/mm}^2 = \frac{506.18 \times 10^3 \text{N}}{\pi D (D - 140)}$

Chamber inner Dia. $D = 147.79 \text{ mm}$
Take $D = 170 \text{ mm}$

Fig.3. Small Chamber Casing
Fig.4. Warm inside the Chamber

3 BEARING DIAGNOSIS

3.1 Minimum Axial Load Required [5]
At higher rotational speeds danger of ball sliding between ring raceways can occur because of centrifugal forces, if axial load $F_a$ drops under minimum value. Minimum value $F_a$ is calculated from equation:

$F_a \text{ min} = M (N_{max} / 1000)^2 \text{ KN}$

$F_a \text{ min} = 6.2 (22 / 1000)^2 = 3 \times 10^{-3} \text{ KN}$

Where, (Refer Table1)

$F_a \text{ min} – \text{minimum axial load [KN]}$
$N_{max} – \text{maximum rotational speed [min}^{-1}]$
$M – \text{Minimum axial load factor}$

If the axial load is smaller than $F_a \text{ min}$, or if bearing relieving comes into being during operation, e.g. of one ball row in double direction bearing, or of one bearing when using a pair of single direction thrust bearings, it is necessary to secure minimum load, e.g. by means of springs.

3.2 Calculate Thrust Load [6]

$F_a = \frac{\pi D^2 P}{4 \times 100000} \text{ KN} = \frac{\pi \times 172^2 \times 100}{40000}$

$F_a = 226.865 \text{ KN}$

Calculate Nominal Thrust Bearing Life for Bearing No 51326M

Thrust Bearing Life,

$L_{10} = \frac{10^6}{60 \times 22} \times (C_{dyn} / F_a)^{10/3}$

$L_{10} = \frac{3765 \text{ hrs}}{10^6}$

$L_{10} = 156.875 \text{ days}$

$L_{10} = 5.2 \text{ Months}$

Here we get the life of bearing 5.2 months but actually when machine is operated 24 hours its designed life should be 50 to 60 Kh. So it is desirable to select a newer bearing with the desired life. (Refer Table 2)

3.3 Bearing Selection Process

- Calculate the dynamic load rating, $C_{dyn}$ For the required bearing life.
- Identify candidate bearings with required rating
- Select bearing with most convenient geometry, also considering cost and availability

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Step 1
Select the bearing life of 3 years = 26280 hrs (See the Table No 2)

\[ \text{Thrust Bearing Life} \ L_{10} = \frac{10^6}{60} \frac{(C_{\text{dyn}} / F_a)^{10/3}}{22} \]

\[ 26280 \text{ hrs} = \frac{10^6}{60} \frac{(C_{\text{dyn}} / 226.865)^{10/3}}{22} \]

\[ C_{\text{dyn}} = 657.39 \text{ KN} \]

Step 2
Identify candidate bearings with required rating
Bearing no 51426 is selected
\[ C_{\text{dyn}} = 637 \text{ KN} \]

### Table 1 Bearing Data [5]

<table>
<thead>
<tr>
<th>Dimensions</th>
<th>d</th>
<th>D</th>
<th>d₁</th>
<th>Dₙ</th>
<th>H</th>
<th>r₁</th>
<th>rₙ</th>
<th>Cdyn</th>
<th>Rating</th>
<th>Fatigue</th>
<th>Limiting Speed</th>
<th>Bearing Designation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Basic Load</td>
<td>150</td>
<td>170</td>
<td>102</td>
<td>90</td>
<td>10</td>
<td>1.0</td>
<td>1.0</td>
<td>110.0</td>
<td>400.0</td>
<td>40.94</td>
<td>1400</td>
<td>1900</td>
</tr>
<tr>
<td>Dynamic Rating</td>
<td>226</td>
<td>220</td>
<td>134</td>
<td>75</td>
<td>21</td>
<td>267.0</td>
<td>1070.0</td>
<td>876.85</td>
<td>750.0</td>
<td>5560</td>
<td>750.0</td>
<td>51326</td>
</tr>
<tr>
<td>Fatigue Limit</td>
<td>270</td>
<td>265</td>
<td>134</td>
<td>110</td>
<td>4.0</td>
<td>657.0</td>
<td>2010.0</td>
<td>199.10</td>
<td>560.0</td>
<td>51426</td>
<td>51426</td>
<td></td>
</tr>
</tbody>
</table>

### Table 2 Bearing-Life Recommendations Table [5]

<table>
<thead>
<tr>
<th>Sr.</th>
<th>Type of Application</th>
<th>Life, Kh</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Instruments and apparatus for infrequent use</td>
<td>Up to 0.5</td>
</tr>
<tr>
<td>2</td>
<td>Aircraft engines</td>
<td>0.5 – 2</td>
</tr>
<tr>
<td>3</td>
<td>Machines for short or intermittent operation where service interruption is of minor importance</td>
<td>4 – 8</td>
</tr>
<tr>
<td>4</td>
<td>Machines for intermittent service where reliable operation is of great importance</td>
<td>8 – 14</td>
</tr>
<tr>
<td>5</td>
<td>Machines for 8-h service which are not always fully utilized</td>
<td>14 – 20</td>
</tr>
<tr>
<td>6</td>
<td>Machines for 8-h service which are fully utilized</td>
<td>20 – 30</td>
</tr>
<tr>
<td>7</td>
<td>Machines for continuous 24-h service</td>
<td>50 – 60</td>
</tr>
<tr>
<td>8</td>
<td>Machines for continuous 24-h service where reliability is of extreme importance</td>
<td>100 – 200</td>
</tr>
</tbody>
</table>

3.4 Bearing Analysis Result and Discussion

Fig.5. Analysis of Right side race in Pro e and Ansys work Bench (Selected Fig Only)
4 ANALYTICAL SOLUTIONS

4.1 Formulas for Right Side and Left Side Races [7]

Left Side Race  Area = $\frac{\pi}{4} \left( d_1^2 - d^2 \right)$

Right Side Race  Area = $\frac{\pi}{4} \left( D^2 - D_1^2 \right)$

Principle Stress $\sigma = \frac{P}{A}$

Axial Deformation or Displacement $\delta = \frac{P*1}{A*E}$

Principle Strain $e = \frac{\delta}{L}$

Strain Energy $= \frac{P\delta}{2}$

4.1.1 Sample Calculation for Left side race of 51326M Bearing

Area of Left Race = $\frac{\pi}{4} \left( d_1^2 - d^2 \right)$

$A = \frac{\pi}{4} \left( 220^2 - 130^2 \right)$

$A = 24.727 \times 10^3 \text{ mm}^2$

Principle Stress $\sigma = \frac{P}{A} = \frac{230 \times 10^3}{24.727 \times 10^3} = 9.301 \text{ N/mm}^2$

Axial Deformation or Displacement

$\delta = \frac{P*1}{A*E} = \frac{230 \times 10^3 \times 25}{24.727 \times 10^3 \times 2.08 \times 10^5} = 1.1179 \times 10^{-3} \text{ mm}$

Principal Strain $e = \frac{\delta}{L} = \frac{1.1179 \times 10^{-3}}{25} = 4.4718 \times 10^{-5}$

Strain Energy $= \frac{P\delta}{2A} = \frac{230 \times 10^3 \times 1.1179 \times 10^{-3}}{2 \times 24.727 \times 10^3} = 0.01039 \text{ N/mm}^2$

4.1.2 Formulas and sample calculations for Ball Part [7]

Area = $\frac{\pi}{4}$ (Diameter of Contact Surface with Race Groove)$^2$

Diameter of Balls is 35mm and 50 mm and groove are 11 mm and 12 mm respectively for old and new bearings

Principle Stress $\sigma = \frac{P}{A}$

Deformation $\delta = \frac{P \cdot r^3}{EI}$

Where

$I = \frac{\pi d^4}{64}$

Principle Strain $e = \frac{\delta}{r}$

Using these equations values of stress, strain, deformation and strain energy are calculated and plotted into the Results Tables

4.1.3 Sample Calculation for Ball part of 51326M Bearing

Area = $\frac{\pi}{4}$ (Diameter of Contact Surface with Race Groove)$^2$
\[ \text{Area} = \pi \left(\frac{11}{2}\right)^2 \]
\[ A = 94.985 \text{ mm}^2 \]

Principle Stress 
\[ \sigma = \frac{P}{A} = \frac{16428.57}{94.985} = 172.959 \text{ N/mm}^2 \]
\[ I = \frac{\pi d^4}{64} = \frac{\pi \times (35)^4}{64} = 73.62 \text{ mm}^4 \]

Deformation 
\[ \delta = \frac{P r^3}{2EI} = \frac{16428.57 \times (17.5)^3}{2 \times 10^3 \times 73.62 \times 10^3} = 0.0057496 \text{ mm} \]

Principle Strain 
\[ \epsilon = \frac{\delta}{r} = \frac{0.0057496}{17.5} = 3.28548 \times 10^{-4} \]

Strain Energy 
\[ \frac{P \delta}{2A} = \frac{16428.57 \times 0.0057496}{2 \times 94.985} = 0.0284 \text{ N/mm}^2 \]

### 4.2 Results

<table>
<thead>
<tr>
<th>Property</th>
<th>Analytical</th>
<th>Software Analysis</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Left Side Race</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Maximum Principle Stress</td>
<td>9.301</td>
<td>7.7204</td>
</tr>
<tr>
<td>Displacement (X Axes)</td>
<td>1.1179E-03</td>
<td>0.0015071</td>
</tr>
<tr>
<td>Maximum Principle Strain</td>
<td>4.4718E-05</td>
<td>0.00004532</td>
</tr>
<tr>
<td>Strain Energy</td>
<td>0.01039</td>
<td>0.57292</td>
</tr>
<tr>
<td><strong>Right Side Race</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Maximum Principle Stress</td>
<td>8.9686</td>
<td>5.4319</td>
</tr>
<tr>
<td>Displacement (X Axes)</td>
<td>1.07795E-03</td>
<td>0.0014456</td>
</tr>
<tr>
<td>Maximum Principle Strain</td>
<td>4.3118E-05</td>
<td>0.000038212</td>
</tr>
<tr>
<td>Strain Energy</td>
<td>4.833856E-03</td>
<td>0.77479</td>
</tr>
<tr>
<td><strong>Ball</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Maximum Principle Stress</td>
<td>172.959</td>
<td>171.18</td>
</tr>
<tr>
<td>Displacement (X Axes)</td>
<td>5.7496E-3</td>
<td>0.0056303</td>
</tr>
<tr>
<td>Maximum Principle Strain</td>
<td>3.28548E-4</td>
<td>0.0015777</td>
</tr>
<tr>
<td>Strain Energy</td>
<td>0.0284</td>
<td>0.26988</td>
</tr>
</tbody>
</table>

### 5.0 CONCLUSIONS

- In this dissertation work we designed some of the components of oil expeller such gear ratio, main shaft rpm, main shaft diameter and crushing chamber dimensions. Design of all these components matches with the components actually used in oil expeller machine. Therefore we can say that the design components used is safe.
- We also calculated the thrust force acting upon the thrust ball bearing, which is quite high which a bearing can withstand. Calculated life of existing bearing is about 5.2 months. Hence new bearing must be selected and hence we select bearing no 51426M which gives satisfactory life of 2.5 years by calculation. This bearing has been also replaced by the company and it is giving satisfactory result and has not failed yet.
- Further we have done static analysis of the Part of bearing on Pro-E wild Fire 4 and Ansys Workbench 11. In analysis results the Principle stress, Principle strain and axial deformation is reduced. Further the results obtained by different software are approximately matching the results obtained by the analytical solutions. From the comparison of the result data we found that results obtained by the Ansys Workbench are closer to the analytical results.
6.0 REFERENCES


