Design, Improvement and Thrust Bearing Analysis of Oil Expeller Machine

Mehul.K.Modh[1*], J.R.Mevada[2]
[1] L.C. Institute of Technology, Bhandu,Mehasana, Gujarat, India.
[2]. V. Patel College of Engineering, Kherva, Mehasana, Gujarat, India

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 [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 [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)

Nomenclatures

TPD	=	Tones per Day
N	=	Revolutions per Minute Main Shaft (RPM)
N_1	=	Revolutions per Minute Motor shaft (RPM)
N_{max}	=	Maximum rotational speed
P	=	Motor Power
p	=	Pressure
D_1	=	Motor Pulley Diameter
D_2	=	Gear Box Pulley Diameter
T	=	Main Shaft Torque
τ	=	Shear Stress
Fa	=	Axial or Thrust Force
M	=	Minimum axial load factor
Cdyn	=	Dynamic Capacity of Bearing

The uses of various investigational techniques were described and illustrated with examples of work by D. Scott of the National Engineering Laboratory. [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

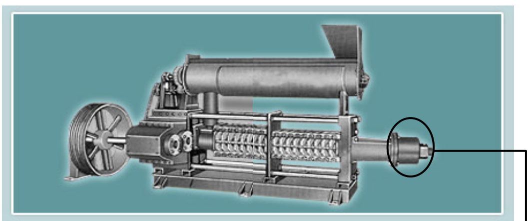

Fig.1. Double Chamber Oil Expeller Machine

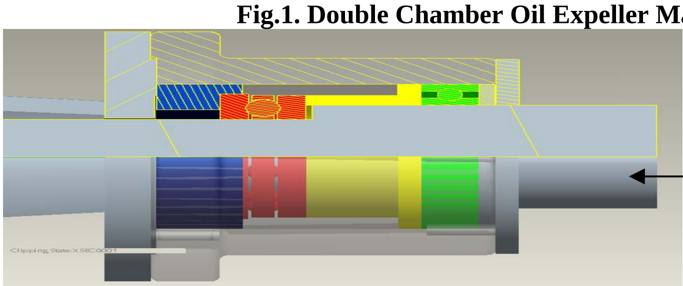

Fig.2. Location of Thrust Ball Bearing

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 [4]

- Find input speed for Gear Box
  For belt drive

$$\frac{D_1}{D_2} = \frac{N_2}{N_1}$$

$$\frac{240}{1000} = \frac{N_2}{940}$$

$$N_2 = 225.6 \text{ RPM}$$

- For Gear Drive

$$\text{Gear Ratio } G_1 = \frac{T_3}{T_2} = \frac{D_3}{D_2}$$

$$G_1 = \frac{54}{18} = \frac{48}{16} = 3$$

Similarly, $G_2 = \frac{T_5}{T_4} = \frac{63}{18} = 3.5$

Gear Ratio $G = G_1 * G_2 = 3 * 3.5 = 10.5$

Main Shaft Rotation N = I/P speed at Gear Box * Gear Ratio G

N = 225.6 * 10.5

N = 21.48 RPM

Say N = 22 RPM

2.4 Main Shaft Design. [4]

1 HP = 0.754699 KW

Motor Power P = 85 HP

P = 64.15 KW

$$\text{Power } P = \frac{2\pi NT}{60}$$

$$64.15 * 10^3 = \frac{2\pi * 22 * T}{60}$$

$$T = 27.84 * 10^3 \text{N m} = 27.84 * 10^6 \text{N mm}$$

Shear Stress for 45C8 Steel [20]

$$\tau_u = 500 \text{N/mm}^2$$

Say $\tau = 150 \text{ N/mm}^2$

Now Torque $T = \frac{\pi * d^3 * \tau}{16} = \frac{\pi * d^3 * 150}{16} = 27.84 * 10^6 \text{ N mm}$

Main Shaft Dia. d = 98.157 mm

Say d = 110 mm

2.5 Small Chamber Design [4]

$$T = 27.84 * 10^6 \text{N mm} = F * r = F * 55$$

$$F = 506.18 * 10^3 N$$

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

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

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

Chamber inner Dia. D = 147.79 mm
Take D = 170 mm

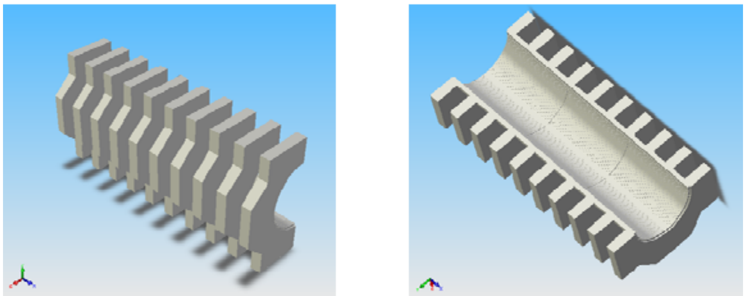

Fig.3. Small Chamber Casing

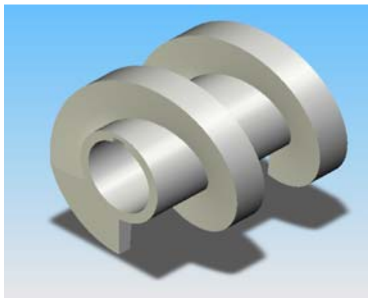

Fig.4. Warm inside the Chamber

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 Fa drops under minimum value. Minimum value Fa is calculated from equation:

Fa min = M $(N_{max} / 1000)^2$ [KN]

Fa min = 6.2 $(22 / 1000)^2$ = $3 * 10^{-3}$ [KN]

Where, (Refer Table1)
Fa min – minimum axial load [KN]
N_{max} – maximum rotational speed [min^{-1}]
M – Minimum axial load factor

If the axial load is smaller than Fa 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]

$$\text{Thrust Load Fa} = \frac{\pi*D^2*P}{4*10000} KN = \frac{\pi*170^2*100}{40000}$$

Fa = 226.865 KN

Calculate Nominal Thrust Bearing Life for Bearing No 51326M

Thrust Bearing Life,

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

$$L_{10} = \frac{10^6}{60 * 22} * (367 / 226.865)^{(10/3)}$$

L_{10} = 3765 hrs
L_{10} = 156.875 days
Lh_{10} = 5.2 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

Step 1

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

Thrust Bearing Life $L_{10} = \dfrac{10^6}{60 * 22} * \left(C_{dyn} / Fa\right)^{(10/3)}$ $\quad$ $26280\ \text{hrs} = \dfrac{10^6}{60 * 22} * \left(C_{dyn} / 226.865\right)^{(10/3)}$

C_{dyn} = 657.39 KN

Step 2

Identify candidate bearings with required rating

Bearing no 51426 is selected

C_{dyn} = 637 KN (See table No 1)

Table 1 Bearing Data [5]

Dimensions d	D	d_1	D_1	H	r_s min	Basic Load Rating Dynamic C_a	Static C_{0a}	Fatigue load limit P_u	Limiting Speed for Lubrication with Grease	Oil	Bearing Designation
mm						kN		kN	min^{-1}		
130	170	170	132	30	1.0	119.0	406.0	43.84	1400	1900	51126
	225	220	134	75	2.1	367.0	1070.0	109.85	750	1000	51326
	270	265	134	110	4.0	637.0	2010.0	199.10	560	750	51426

Abutment and Fillet Dimensions d	d_a min	D_a max	r_a max	Weight ~	Minimum Axial Load Factor
mm				kg	
130	154	146	1.0	1.870	0.650
	186	169	2.0	13.300	6.200
	213	187	3.0	32.000	18.000

Table 2 Bearing-Life Recommendations Table [5]

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

3.4 Bearing Analysis Result and Discussion

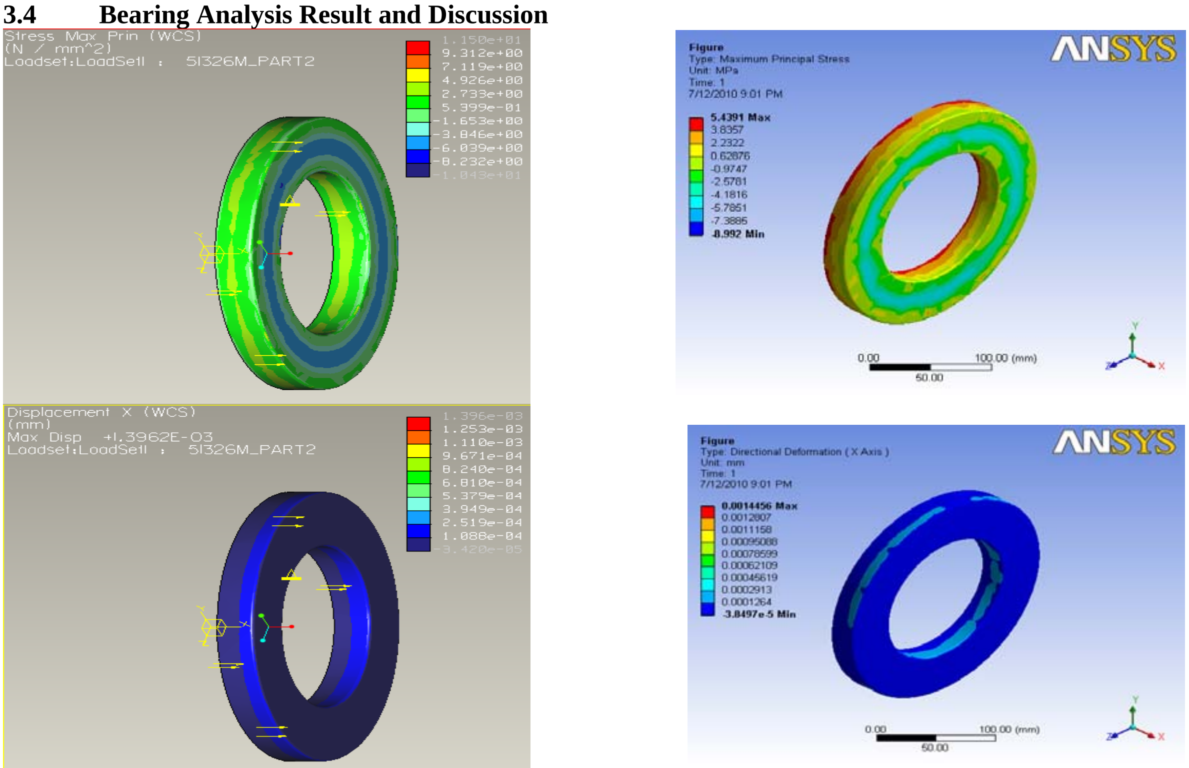

Fig.5. Analysis of Right side race in Pro e and Ansys work Bench (Selected Fig Only)

4 ANALYTICAL SOLUTIONS

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

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

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

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

Axial Deformation or Displacement $\delta_l=\frac{P*l}{A*E}$

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

$$\text{Strain Energy}=\frac{P\delta_1}{2}$$

4.1.1 Sample Calculation for Left side race of 51326M Bearing

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

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

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

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

Axial Deformation or Displacement

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

Principle Strain $e=\frac{\delta_l}{L}=\frac{1.1179*10^{-3}}{25}=4.4718*10^{-5}$

$$\text{Strain Energy}=\frac{P\delta_1}{2A}=\frac{230*10^3*1.1179*10^{-3}}{2*24.727*10^3}=0.01039\text{N/mm}$$

4.1.2 Formulas and sample calculations for Ball Part [7]

$\text{Area}=\frac{\pi}{4}$ (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 $\delta=\frac{P\,r^3}{E\,I}$

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

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

$$\text{Area} = \frac{\pi}{4}(11)^2$$

A = 94.985 mm^2

$$\text{Principle Stress } \sigma = \frac{P}{A} = \frac{16428.57}{94.985} = 172.959 \text{ N/mm}^2$$

$$I = \frac{\pi d^4}{64} = \frac{\pi * (35)^4}{64} = 73.62 \text{ mm}^4$$

$$\text{Deformation } \delta = \frac{P\,r^3}{EI} = \frac{16428.57*(17.5)^3}{2.08*10^5*73.62*10^3} = 0.0057496 \text{ mm}$$

$$\text{Principle Strain } e = \frac{\delta}{r} = \frac{0.0057496}{17.5} = 3.28548*10^{-4}$$

$$\text{Strain Energy} = \frac{P\delta_1}{2A} = \frac{16428.57*0.0057496}{2*94.985} = 0.0284 \text{ N/mm}$$

4.2Results

Table 3 Comparison of results obtained by Ansys and Pro-E for old and new bearing

Property	***Analytical***		***Software Analysis***			
	Old Bearing 51326M	***New Bearing 51426M***	***Ansys***		***Pro E***	
			Old Bearing	***New Bearing***	***Old Bearing***	***New Bearing***
Left Side Race						
Maximum Principle Stress	9.301	5.4945	7.7204	4.3734	13.89	11.26
Displacement (X Axes)	1.1179E-03	9.7738E-04	0.0015071	0.0014808	0.0016399	0.0016318
Maximum Principle Strain	4.4718E-05	2.6415E-05	0.00004532	0.000026626	6.134E-05	0.00005011
Strain Energy	0.01039	2.6851E-03	0.57292	0.58714	0.0167	0.03325
Right Side Race						
Maximum Principle Stress	8.9686	5.33259	5.4319	3.5906	11.50	5.282
Displacement (X Axes)	1.07795E-03	9.4858E-04	0.0014456	0.0012903	0.0013962	0.0012552
Maximum Principle Strain	4.3118E-05	2.5637E-05	0.000038212	1.1491E-07	0.00006191	6.911E-07
Strain Energy	4.833856E-03	2.52919E-03	0.77479	0.73595	0.001648	0.0004533
Ball						
Maximum Principle Stress	172.959	203.46	171.18	128.34	91.22	1.066E2
Displacement (X Axes)	5.7496E-3	5.634E-3	0.0056303	0.0057205	5.6152E-3	5.7257E-3
Maximum Principle Strain	3.28548E-4	7.076E-4	0.0011577	0.0007779	1.802E-3	1.442E-3
Strain Energy	0.0284	1.7996	0.26988	0.6978	6.207	3.966

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

[1] Bamgboye and Adejumo, “Development of a Sunflower Oil Expeller”, Agricultural Engineering International: the CIGR E-journal, Manuscript EE 06 015, Vol IX. September, 2007

[2] Alessandra Maria Sabelli, "Design of a Press for Oil Extraction from Moringa Seeds for Haiti" Thesis, Department of Mechanical Engineering, Massachusetts institute of technology, January 2004

[3] D. Scott, “Bearing failure Diagnosis and Investigation”, Nationall Engineering Laboratory, East Kilbride (Gt. Britain), Vol Wear 25, Page No 199-213, 1973

[4] R.S. Khurmi and J.K. Gupta, “A Text Book of Machine Design”, Chapter 14 Shaft and Chapter 27 Rolling Contact Bearings, 14th edition, 2005

[5] ZKL Bearings, “Thrust Ball Bearings”, Manual Page No 174-183

[6] Flender Extruder Ltd, “Selection Process Manual”, K21-1.0 EN, 2005

[7] Indrajeet M. Jain, “Advanced Strength of Materials”, Tech Max Publication, Chapter 1, Page No 1-48