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# Design and Stress Analysis of Screw Shaft for Peanut Oil Screw Press Expeller

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***Abstract* – The screw press oil expellers are reported to be more efficient than all other methods of oil extraction. The oil seeds fed through the hopper are crushed and transported by a rotating screw in a press barrel. Continuous transport of material by the screw shaft causes the pressure to increase to a level needed, which increases the friction inside the screw press and generates heat which lowers viscosity of the oil in the crushed seeds thereby increasing the oil flow rate. The oil and cake are usually collected at the oil outlet and press cake exit.**

**Screw shaft is the main component of the oil extraction machine. Existing screw press oil expeller is the JY 200 machine with 220W motor power and 1500rpm. The von-Mises stress of existing screw design is 94.11 MPa and the modified screw design is 89.876 MPa for theoretical. In this research, the modified screw design is considered in order to improve the oil capacity, the efficiency and number of cycles to failure and then to reduce the von-Mises stress, deformation and fatigue stress of automatic oil press machine by changing the pitch of screw shaft for peanut oil screw press expeller. Peanut oil screw press expeller is easy to repair and easy to maintenance. This machine is very suitable for local production and operation.**

***Keywords* – screw shaft, efficiency, oil capacity, Von-Mises stress, deformation and fatigue stress, number of cycles.**

## I. INTRODUCTION

The oil content of groundnut can contain up to 50% oil (although the usual range is 40% to 45%) and 25 % to 30 % protein. Oil is extracted from groundnut through either traditional means (mostly dependent on human energy with about 20-30% of the oil extracted) or mechanical means with over 30% of the oil which have been extracted. Most vegetable oils are recovered by grinding, cooking, expelling and pressing, or by solvent extraction of the raw materials [1].

The most common method of extracting edible oil from oilseeds is mechanical pressing of oilseeds. Extraction of groundnut oil could be done in two major ways that is the mechanical and improved methods. The improved method consists of chemical extraction and mechanical expression. The chemical extraction method requires the use of organic solvents to recover the oil from the products. Mechanical method involves the application of pressure to already pre-treated oil-bearing products. It employs the use of devices like screw and hydraulic presses as a means of applying the pressure [1].

Screw type oil expeller is advanced oil processing machinery, characterized by their high oil output rate with good quality, simple design, easy to use and continuous operation. Expeller's use a horizontally rotating metal screw, which feeds oil-bearing seeds into a barrel shaped outer casing with perforated walls. The seeds are continuously fed to the expeller, which grinds, crushes and presses the oil out as it passes through the machine. The residue of the material from which oil has been extracted, exits from the machine, and is known as the cake. The percentage of oil expressed by expellers is nearly 90% depending upon the type and kind of products as well as the expeller being employed [4].

The oil seeds fed through the hopper are crushed and transported by a rotating screw inside a barrel. Continuous transport of material by the screw shaft causes pressure to increase to a level needed, which increases friction inside the screw press and generates heat which lowers viscosity of the oil in the crushed seeds thereby increasing the oil flow rate. The machine components are the speed reduction gear, expellant unit, drains collector, driving and driven gears, and the hopper. The groundnut seeds were pre-heated by roasting before extraction of the oil in it. The machine gave a better performance at the speed of 60 revolutions per minute and improvement in the design can be achieved by incorporation a heating chamber along the expeller barrel.

In this paper, the author intends to design the screw shaft and to calculate the stresses of peanut oil screw press oil expeller. Figure 1 shows main components of the oil expeller [11].

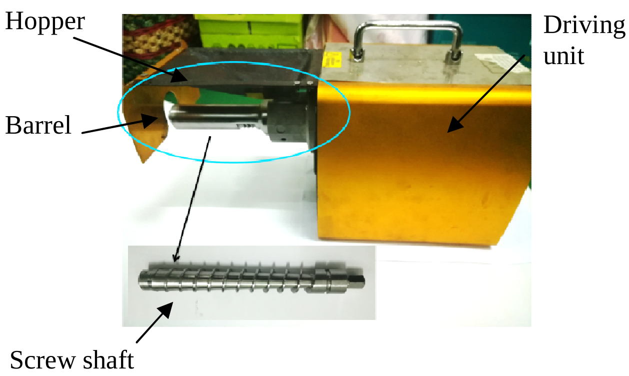


Figure 1. Main Components of the Oil Expeller

## II. Material And Method

The screw press consists of the following components: screw shaft, cylindrical barrel, feeding hopper, electric motor, cake outlet, oil outlet and main frame. The cylindrical barrel was made from a stainless steel of length 143 mm, inside diameter of 22 mm and thickness of 5 mm. The screw shaft was made from a stainless-steel solid rod of diameter 20 mm and length 156 mm, which was machine on their lathe at decreasing screw depth. The worm shaft is housed in the cylindrical barrel at a clearance of 1 mm between the screw diameter and inside diameter of the barrel. For AISI 1035 steel shaft, the ultimate strength is 550 MPa and yield strength is 283 MPa. Young modulus is 210 GPa and factor of safety is 4 are used in this research. The screw shaft rotates in the cylindrical barrel and conveys the peanut from the feeding section towards the discharge section. Pressure is achieved in the machine by the operation of the screw shaft which was designed to have a decreasing depth thus reducing the area available for the peanut in the barrel. Raw materials enter one side of the screw shaft through the hopper and waste products exit the other side. The expeller screw shaft rotates inside a cylindrical barrel. The peanut to be pressed is between the screw and the barrel and propelled by the rotating screw in a direction parallel to the axis. Expeller pressing is a mechanical method for extracting oil from raw materials. The raw materials are squeezed under high pressure in a single step [10]. In operation, the oilseed is introduced into the machine through the feeding hopper; the machine conveys, crushes, grinds and presses the oilseed inside the cylindrical barrel with the aid of the screw shaft until oil is squeezed out of the seed. The oil extracted is drained though the oil channel into the oil tray where it is collected; the residual cake is discharged at the cake outlet and collected at the cake tray [1,10].

## III. Design Consideration Of Screw Press Oil Expeller

While designing the machine consideration included: high oil yield and high oil extraction efficiency. Other considerations included the simplicity in design and easy to fabricate the machine, be usable by anybody even without any previous technical training. Specification data for screw shaft shown in following Table I.

TABLE I. Specification Data For Screw Press Oil Expeller

| Description | Symbol | value | unit |
|---|---|---|---|
| Length of screw shaft | L | 133 | mm |
| Power | P | 220 | W |
| Outside diameter of barrel | $d_o$ | 30 | mm |
| Speed of motor | $N_p$ | 1500 | rpm |
| Outside diameter of screw shaft | $d_o$ | 20 | mm |
| Clearance | c | 1 | mm |
| Coefficient of friction | μ | 0.515 (0.3~0.73) | - |
| Diameter of pinion (helical gear) | $D_{p1}$ | 6 | mm |
| Diameter of gear (helical gear) | $D_{g1}$ | 28 | mm |
| Diameter of pinion (spur gear) | $D_{p2}$ | 11 | mm |
| Diameter of gear (spur gear) | $D_{g2}$ | 60 | mm |

Calculation of velocity ratio,

$$V.R = \frac{N_p}{N_g} = \frac{D_g}{D_p} \tag{1}$$

Power,

$$P = \frac{2\pi NT}{60} \tag{2}$$

Shear Stress,

$$\tau = \frac{0.5S_y}{N} \tag{3}$$

N = factor of safety for steel

$$N = \begin{cases} 4 & \text{Steady Load} \\ 8 & \text{Live Load} \end{cases}$$

Calculation of Diameter,

$$\tau = \frac{16T}{\pi d_c^3} \tag{4}$$

Mean Diameter,

$$d_m = \frac{d_0 + d_c}{2} \tag{5}$$

Where, $d_c$ = core diameter

Number of Screw Turn,

$$n_t = \frac{l}{p}$$

Helix Angle,

$$\tan\alpha = \frac{p}{\pi d_m} \tag{6}$$

Friction Angle,

$$\tan\phi = \mu \tag{7}$$

$$\phi = \tan^{-1}\mu$$

Effort, P

$$P = \frac{T}{r_m} \tag{8}$$

Weight, W

$$W = \frac{P}{\tan(\alpha+\phi)} \tag{9}$$

Torque required to raise and lower the load,

$$T_R = W\frac{d_m}{2}\tan(\alpha+\phi) \tag{10}$$

$$T_L = W\frac{d_m}{2}\tan(\phi-\alpha) \tag{11}$$

Calculation of efficiency,

$$\eta = \frac{\tan(\alpha)}{\tan(\alpha+\phi)} \tag{12}$$

Calculation of oil capacity,

$$Q = \frac{\pi}{4}\times d_m^2\, pN\eta \tag{13}$$

TABLE II. Result Data For Screw Press Oil Expeller

| Parameters | 1. | 2. | 3. | unit |
|---|---|---|---|---|
| Pitch | 12 | 13 | 14 | mm |
| Helix Angle | 14.3 | 15.4 | 16.55 | degree |
| Weight | 5365.7 | 5162.6 | 4959 | N |
| Torque (rise) | 35.67 | 35.66 | 35.65 | N |
| Torque (lower) | 9.25 | 8.12 | 7.02 | N |
| Efficiency | 28.8 | 29.9 | 31 | % |
| Oil Capacity | 2.159 | 2.428 | 2.71 | L/hr |

| Parameters | 4. | 5. | 6. | unit |
|---|---|---|---|---|
| Pitch | 15 | 16 | 17 | mm |
| Helix Angle | 17.7 | 18.8 | 19.84 | degree |
| Weight | 4763.8 | 4584.4 | 4584.4 | N |
| Torque (rise) | 35.64 | 35.63 | 35.62 | N |
| Torque (lower) | 6 | 5.1 | 4.31 | N |
| Efficiency | 32 | 32.83 | 33.6 | % |
| Oil Capacity | 2.998 | 3.2814 | 3.5682 | L/hr |

Calculation of Stresses on the Screw,

Normal stresses,

$$\sigma_{z1} = -\frac{F}{A} = -\frac{4W}{\pi d_c^2} \tag{14}$$

Calculation of allowable stress,

$$\sigma_{all} = \frac{S_u}{N} \tag{15}$$

Bearing pressure,

$$P_b = \frac{W}{\pi d_m n_t t} \tag{16}$$

Where, $P_b$ = bearing pressure ($N/m^2$)

$n_t$ = number of threads

$t$ = depth of thread

Bearing stress,

$$\sigma_B = -\frac{F}{A} = -\frac{2W}{\pi d_m n_t p} \tag{17}$$

$$\sigma_B = \sigma_{z_2}$$

Bending stress,

$$\sigma_b = \frac{Mc}{I} = \frac{6W}{\pi d_r n_t p} \tag{18}$$

$$\sigma_b = \sigma_x$$

$$\sigma_z = \sigma_{z1} + \sigma_{z2}$$

Equivalent stress (Von-Mises stress)

$$\bar{\sigma}=\frac{1}{\sqrt{2}}[(\sigma_x-\sigma_y)^2+(\sigma_y-\sigma_z)^2+(\sigma_z-\sigma_x)^2]^{\frac{1}{2}}$$

Calculation of Deformation on the Screw,

$$\delta=\frac{FL}{AE} \tag{19}$$

TABLE III. Result Data Of von-Mises Stress And Deformation

| Parameters | 1. | 2. | 3. | unit |
|---|---|---|---|---|
| Pitch | 12 | 13 | 14 | mm |
| Pressure | 20.7 | 21.9 | 22.05 | MPa |
| Normal Stress | -68.32 | -65.73 | -63.14 | MPa |
| Bending Stress | 7.76 | 7.58 | 6.765 | MPa |
| Allowable Stress | 137.5 | 137.5 | 137.5 | MPa |
| Von-Mises Stress | 96.25 | 94.17 | 91.74 | MPa |
| Deformation | 0.01923 | 0.0185 | 0.0178 | mm |
| Parameters | 4. | 5. | 6. | unit |
| Pitch | 15 | 16 | 17 | mm |
| Pressure | 22.46 | 24.32 | 23.45 | MPa |
| Normal Stress | -60.65 | -58.37 | -56.29 | MPa |
| Bending Stress | 6.74 | 6.58 | 6.21 | MPa |
| Allowable Stress | 137.5 | 137.5 | 137.5 | MPa |
| Von-Mises Stress | 89.89 | 88.2 | 86.47 | MPa |
| Deformation | 0.01707 | 0.01643 | 0.01584 | mm |

Mean stress,

$$\sigma_m=\frac{\sigma_{max}+\sigma_{min}}{2} \tag{20}$$

Alternating stress,

$$\sigma_a=\frac{\sigma_{max}-\sigma_{min}}{2} \tag{21}$$

$$\sigma_{max}=\sigma_x$$

$$\sigma_{min}=\sigma_z$$

Estimating the Theoretical Endurance Limit,

An approximate endurance limit,

$$S'_e=0.504S_u \quad S_u\le 1400\text{MPa}$$

Calculation of surface factor,

$$K_a=aS_u^b \tag{22}$$

Calculation of size factor,

$$K_b=(\frac{d}{7.62})^{-0.1133} \tag{23}$$

For axial loading, there is no size effect.

$$K_b=1$$

Loading factor,

$$k_c=\begin{cases}1 & \text{bending}\\ 0.85 & \text{axial}\\ 0.577 & \text{torsion}\end{cases}$$

Consideration of temperature factor,

$$k_d=1$$

Consideration of reliability factor,

$K_e=0.702$ [13 Nor]

Consideration of miscellaneous-effects factor,

$$k_f=1$$

Calculation of Actual Endurance Strength,

$$S_e=k_a\times k_b\times k_c\times k_d\times k_e\times k_f\times S'_e \tag{24}$$

The fatigue strength,

$$S_f=\frac{\sigma_a}{1-\frac{\sigma_m}{S_u}} \tag{25}$$

For AISI 1035 steel,

$$S_m=0.9S_u \tag{26}$$

TABLE IV. Result Data Of Endurance And Fatigue Stress

| Parameters | 1. | 2. | 3. | unit |
|---|---|---|---|---|
| Pitch | 12 | 13 | 14 | mm |
| Maximum Stress | 7.76 | 7.58 | 6.765 | MPa |
| Minimum Stress | -70.045 | -67.415 | -64.64 | MPa |
| Mean Stress | -31.14 | -29.91 | -28.94 | MPa |
| Alternating Stress | 38.9 | 37.5 | 35.7 | MPa |
| Endurance Strength | 88.085 | 88.085 | 88.085 | MPa |
| Fatigue Stress | 36.82 | 35.566 | 33.91 | MPa |
| Parameters | 4. | 5. | 6. | unit |
| Pitch | 15 | 16 | 17 | mm |
| Maximum Stress | 6.74 | 6.58 | 6.21 | MPa |

| | | | | |
|---|---|---|---|---|
| Minimum Stress | -62.148 | -59.89 | -57.67 | MPa |
| Mean Stress | -27.7 | -26.66 | -25.73 | MPa |
| Alternating Stress | 34.44 | 33.235 | 31.94 | MPa |
| Endurance Strength | 88.085 | 88.085 | 88.085 | MPa |
| Fatigue Stress | 32.79 | 31.7 | 30.51 | MPa |

Fatigue Life,

The coefficient,

$$a = \frac{(f\,S_u)^2}{S_e} \tag{27}$$

The exponent,

$$b = -\frac{1}{3}\log\left(\frac{f\,S_u}{S_e}\right)$$

Consideration of Fatigue strength fraction,

$f = 0.9$

Number of cycles to failure,

$$N = \left(\frac{S_f}{a}\right)^{\frac{1}{b}} \tag{28}$$

Calculation of Safety Factor,

Mean torque,

$$T_m = \frac{T_{max}+T_{min}}{2} \tag{29}$$

Alternating torque,

$$T_a = \frac{T_{max}-T_{min}}{2} \tag{30}$$

$T_{max} = T_R$ , $T_{min} = T_L$

Mean shear stress,

$$\tau_m = \frac{16T_m}{\pi d_c^{\,3}} \tag{31}$$

Alternating shear stress,

$$\tau_a = \frac{16T_a}{\pi d_c^{\,3}} \tag{32}$$

Ultimate shear strength,

$$S_{us} = 0.67S_u \tag{33}$$

Where, $S_u$ = Ultimate Stress (MPa)

TABLE V. Result Data For Screw Press Oil Expeller

| Parameters | 1. | 2. | 3. | unit |
|---|---|---|---|---|
| Pitch | 12 | 13 | 14 | mm |
| Number of cycles | 32.577 | 37.42 | 45.28 | ( $10^6$ ) |
| Mean Shear Stress | 114.387 | 111.5 | 108.66 | MPa |
| Alternating Shear Stress | 67.28 | 70.13 | 72.91 | MPa |
| Parameters | 4. | 5. | 6. | unit |
| Pitch | 15 | 16 | 17 | mm |
| Number of cycles | 51.79 | 59.3 | 69.1 | ( $10^6$ ) |
| Mean Shear Stress | 106.04 | 103.72 | 101.68 | MPa |
| Alternating Shear Stress | 75.5 | 77.74 | 79.73 | MPa |

Figure 2. Modified Screw Shaft Design (15mm Pitch)

## IV. RESULT AND DISCUSSION

In this research, the results of efficiency, oil capacity, fatigue stress, number of cycles to failure and alternating shear stress are computed by theoretical analysis. When the analysis each AISI 1035 steel shaft is studied for six different pitch 12mm, 13mm, 14mm, 15mm, 16mm and 17mm but we compared 13mm, 14mm and 15mm pitch. Shaft length, major diameter and minor diameter are kept constant. Table II shows the comparison of efficiency and oil capacity. According to the theoretical results, 15mm pitch is optimum capacity than 13mm and 14mm pitch. Table III shows the comparison of allowable stress and von-Mises stress by changing three different pitch. The maximum von-Mises stress is less than the allowable stress. So, the design is safe. By comparing the theoretical results, the von-Mises stress and deformation of 15mm pitch are less than other two pitch. Table IV shows the minimum fatigue stress and table V shows the maximum cycles to failure by changing from 13mm pitch to 15mm pitch screw shaft of JY 200 oil expeller.

## V. Conclusion

The screw shaft is the main components of the screw press oil expeller because it conveys, grinds and crushes the peanut inside the cylindrical barrel. The original screw shaft design is 13 mm pitch screw design. The outside diameter is 20 mm and inlet depth is 5 mm. In order to get high oil capacity, the modified screw design is constructed. Screw designs are calculated and compared for 13 mm pitch, 14 mm pitch, and 15 mm pitch. 15 mm pitch screw design is higher oil capacity, efficiency, endurance limit, number of cycles than others. According to theoretical results, oil capacity of modified and existing screw design is 2.997 L/hr and 2.428 L/hr.

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