Study of heat generation and dissipation mechanism for an exciter shaft and its impact

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Abstract: Compactor is a heavy-duty machinery that uses compaction to reduce the size of waste materials or biomass or compaction of load. A bearing is a machine element that restricts relative motion to only the desired motion, and decreases friction between moving parts. It is an essential component in a compactor and is used for supporting the exciter shaft. Compaction is achieved by vibration of roller drum, which in turn is achieved by means of generated at the exciter shaft with an eccentric mass. During this process of generation of forced excitation, heat is generated at shaft to bearing interface. The current heat generation and dissipation paths for the bearing on the exciter shaft of the compactor were studied and calculated. Suggestions for heat dissipation optimization are provided. Various optimization methods were devised for heat dissipation to improve the wear as well as efficiency of the bearing and further testing is required for implementation of same.

1. Introduction

Compactors are a type of heavy construction equipment used in applications such as compaction of soils and roads [1]. It works on the principle of a vibrating roller drum which is connected to a heavy drive which aids in its rotation. This vibration facilitates the compaction and its use is commonly seen for paving asphalt as well as dirt roads. These machines are generally operated on rough terrains which cause extreme vibrations which has a negative impact on the driver. These vibrations also cause a high amount of friction and heat is generated rapidly in the drum with temperatures sometimes reaching greater than 140°C. The conditions for compacting no longer remain favorable for the machine or the driver in these cases.

To ensure smoother functioning of these machines, bearings are used to connect the exciter shaft to the roller drive. Roller bearings are chosen for use here. They are commonly known as antifriction bearings due to their moving action and contact between the bearings is rolling. This means that the area in contact for rolling bearings is lesser, and hence friction is also lower. Their ability to provide high precision and low friction leads them to be a suitable choice for high speed applications while reducing noise and heat generated [2]. Among roller bearings, there are many kinds such as tapered roller bearings, cylindrical roller bearings, spherical roller bearings, needle roller bearings, and needle roller thrust bearings. Among these options, cylindrical roller bearings are chosen for their high load bearing capacity, high stiffness, low friction, high reliability, and long service life. Yu Wang, et al. [3] concluded that at high temperature, the bearings expand thermally and also cause heat generation of the bearing which affects the temperature field. Fangbo Ma et al. [4] theorized that with an increase in temperature, there is a decrease in the heat generated by a spherical roller bearing.
lubricated by grease. Fangbo Ma et al. [5] devised a method of calculating heat transfer using differential equations and Blanuša et al. [6] found the thermal behaviour of a cylindrical roller bearing using mathematical modelling.

2. Calculation of Heat Generation

2.1 Spherical Roller Bearings:
In spherical roller bearing, the main source of heat generation is contact between the roller and the raceway. The viscous friction in the middle of cages and inner ring land, pocket friction between roller and cage, and heat due to roller churning are also some sources of heat generation.

For a roller bearing, the heat generation rate of roller-raceway contact is given by:

$$H_{RRC} = \sum \sum H_{i,j}$$

(1)

Where,
- $H_{i,j}$ – heat generated when jth roller is in contact with inner raceway.
- $H_{o,j}$ – heat generated when jth roller is in contact with outer raceway.

Heat generation rate due to:

i) Contact between cage and inner ring land contact:

$$H_{cl} = 0.5 \times d_m \times F_{cl} \times (\omega_i - \omega_c)$$

(2)

ii) Pocket contact between roller and cage:

$$H_{p} = 0.5 \times Z \times D \times F_{p} \times \omega_r$$

(3)

iii) Roller churning:

$$H_{d} = 0.5 \times Z \times F_{d} \times d_m \times \omega_c$$

(4)

Total Heat generated:

$$H_{tot} = 2 \times (H_{RRC} \times H_{d} \times H_{cl} \times H_{p})$$

(5)

2.2 Palmgren Equation:
Taking into account Palmgren equation, we know that heat generated in a rolling bearing is based on i) Friction torque and ii) Bearing speed

Friction torque is given by:

$$M = M_1 + M_2$$

(6)

- Where $M_1$- moment depending on lubricant’s hydrodynamic losses

$$M_1 = 10^{-7} f_0 \times (\nu \times n)^{2/3} D^3$$

(7)

Where $f_0 = \begin{cases} 160 \times 10^{-7} f_0 D^3, \nu \times n < 2000 \\ \text{coefficient based on type of bearing and lubricant} \end{cases}$

- Where $M_2$-moment depending on loss due to friction

$$M_2 = f_1 \times P_1 \times D$$

(8)

In which,
- $f_1$ - coefficient based on type of bearing and load
- $P_1$ - Calculated load

The heat generated is given when the frictional moment is multiplied by the angular speed.

$$G = \pi \times M \times n / 30 \times 10^{-3}$$

(9)

2.3. Cylindrical Roller Bearings
Another way to calculate frictional moment is by calculating the various kinds of frictional moment which are frictional moment due to rolling motion, friction moment due to sliding motion, frictional moment due to presence of seal, frictional moment caused by churning and drag losses.

$$M = M_m + M_d + M_{seal} + M_{drag}$$

(10)

The rolling frictional moment is determined by following formula:

$$M_m = \Phi_{sh} \times \Phi_{rs} \times G_{fr} \times (\nu \times n)^{0.6}$$

(11)
Given that $M_{rr}$ is frictional moment due to rolling (Nmm), $\Phi_{rs}$ is the starvation factor, $G_{rr}$ is variable corresponding to bearing type, the mean diameter of the bearing (mm), axial load on bearing (N), and radial load on bearing (N), $n$ is speed of rotation (rpm) and $\nu$ is viscosity of the oil during the operation ((mm)$^2$/s).

The sliding frictional moment is calculated by following equation:

$$M_{sl} = G_{sl} \mu_{sl}$$ (12)

Given that $M_{sl}$ is frictional moment due to sliding (Nmm), $G_{sl}$ variable corresponding to bearing type, the mean diameter of the bearing (mm), axial load on bearing (N) and the radial load on bearing (N) and $\mu_{sl}$ is coefficient of friction for sliding.

The frictional moment of seals for double-sealed bearings is determined using:

$$M_{seal} = K_{S1} \times d_s \times b + K_{S2}$$ (13)

Given that $M_{seal}$ is frictional moment of the seal (Nmm), $K_{S1}$ is a constant corresponding to type of seal, bearing, and its dimensions, $d_s$ is the diameter of counterface of seal (mm), $b$ is an exponent and $K_{S2}$ is a constant both depending on type of seal, bearing, and its size.

2.4 Drag losses

When the chosen form of lubrication for bearings are partial or complete oil bath submersion, it leads to drag loss. It is significant to power loss. It is dependent on speed of bearing, viscosity of oil, level of oil, dimensions of oil bath as well as dependent on possible external oil agitation.

2.4.1 Drag loss in oil bath lubrication

- When the oil reservoir is big in size, then its effects on the agitation of oil from outside sources can be neglected.
- The shaft is placed horizontally.
- The speed of inner ring rotation is constant.
- The speed of inner ring rotation lesser than the allowed speed.
- The viscosity of the oil is within permissible limits depending on level of bearing submerged.

The position of lowest contact point is determined using:

- Outside diameter $D$ - for tapered roller bearings (mm)
- Outer ring mean diameter - for radial rolling bearings (mm) = 0.5*(D + D1)

The drag losses frictional moment is given:

$$M_{drag} = 4 * V_M * K_{roll} * C_w * d_m^{4*} n^2 + 1.093 * 10^{-7} * n^2 * d_m^{3*} (n^* d_m^{2*} f/v)^{1.779} * R_s$$ (14)

The rolling element related constants are:

$$K_{ball} = \frac{irw * K_z * (D + D1) * 10^{-12}}{(D - d)}$$ (15)

$$K_{roll} = \frac{K_t * K_z * (d + D) * 10^{-12}}{(D - d)}$$ (16)

The variables and functions used in to find frictional moment due to drag loss are:

$$C_w = 2.789 * 10^{14} * I_b - 2.786 * 10^{4} * I_b^2 + 0.0195 * I_b^2 + 0.6439$$ (17)

$$I_b = 5 * K_t * B / d_m$$ (18)

$$t = \sin(0.5\theta), when \ 0 <= t <= [\theta]$$ (19)

$$1, when \ [\theta] < t < 2[\theta]$$

$$R_s = 0.36 * d_m^2 * (t-sin t) * f_{sk}$$ (20)

$$t = 2 * cos^2(0.6 * d_m^2 - H / 0.6 * d_m) \ when \ H >= 1.2 d_m, use \ H = 1.2 * d_m$$ (21)

$$f_{sk} = 0.05 * K_t * (D + d) / (D - d)$$ (22)

Given that $M_{drag}$ is frictional moment caused due to drag loss (Nmm), $B$ is width of the bearing (mm), $V_M$ is factor due to drag loss, $d_m$ is the mean dia. of bearing (mm), $d$ is bore dia. of bearing (mm), $D$ is outside dia. of the bearing (mm), $H$ is height of the oil present or level (mm), $I_{rn}$ is number of ball rows, $K_z$ is constant related to the geometry of the bearing,
KL is constant related to the geometry of the roller bearing, n is speed of rotation (rpm) and n is viscosity of the lubricant during operation (mm)²/s.

3. Specification of Bearing:
   In this work we considered a cylindrical roller bearing which is shown in Figure 1. The dimensions of the bearing are given in Table 1.

Table 1. Dimensions of the Cylindrical Roller Bearing

| Outer Diameter (mm) | Inner Diameter (mm) | Width (mm) | Load at High Frequency (Radial) (kN) | Load at High Frequency (Axial) (kN) | Speed of Operation (rpm) |
|---------------------|---------------------|------------|-------------------------------------|------------------------------------|--------------------------|
| 320                 | 150                 | 108        | 260                                 | 1                                  | 2040                     |

Figure 1. Cylindrical Roller Bearing

4. Calculations of Power Loss

Considering the radial force, axial force, and speed at constant values of 260 kN, 1 kN, and 2040 rpm, the total frictional moment and power loss is calculated as shown in Table 2 and Table 3 for the left and right bearing respectively.

On the left bearing, at the inner ring temperature of 120°C and outer ring temperature of 100°C, we find an average total frictional moment of 30500 Nmm and power loss of 6527 W. At higher temperature range of inner ring temperature 140°C and outer ring temperature of 120°C, there is a total frictional moment of 27400 Nmm and a lower power loss of 5864 W and at lower temperature range of inner ring temperature 110°C and outer ring temperature of 90°C, there is a total frictional moment of 32600 Nmm and a higher power loss of 6973 W.

On the right bearing, the total frictional moment is found to be constant irrespective of the temperature range at 13800 Nmm. The power loss variation is also minimal and ranges from 2953 W to 2959 W.

5. Heat Dissipation

The following governing equations are used:

The heat dissipated by conduction mode:

\[ Q = kA \left( T_2 - T_1 \right) / L \] (23)

The heat dissipated by convection mode:

\[ Q = hA \left( T_2 - T_1 \right) \] (24)

The heat dissipated by radiation mode:

\[ Q = \varepsilon \sigma A \left( T_2^4 - T_1^4 \right) \] (25)

By considering the following factors, the heat dissipated through conduction and radiation is calculated as shown in Table 4 and Table 5.

The value of the constants are: \( k = 109 \text{ W/mK} \) - for brass; \( \varepsilon = 0.8 \) - for polished brass; \( \sigma = 5.6703 \times 10^{-8} \text{ W/m²K}^4 \)
Table 2. Power Loss on the Left bearing

| Radial force (kN) | Axial force (kN) | Speed (rpm) | Inner Ring Temp (°C) | Outer Ring Temp (°C) | Total Frictional Moment (Nmm) | Power Loss (W) |
|-------------------|------------------|-------------|----------------------|----------------------|----------------------------|---------------|
| 260               | 1                | 2040        | 110                  | 90                   | 32600                      | 6973          |
| 260               | 1                | 2040        | 120                  | 100                  | 30500                      | 6527          |
| 260               | 1                | 2040        | 140                  | 120                  | 27400                      | 5864          |

Table 3. Power Loss on the Right bearing

| Radial force (kN) | Axial force (kN) | Speed (rpm) | Inner Ring Temp (°C) | Outer Ring Temp (°C) | Total Frictional Moment (Nmm) | Power Loss (W) |
|-------------------|------------------|-------------|----------------------|----------------------|----------------------------|---------------|
| 260               | 1                | 2040        | 110                  | 90                   | 13800                      | 2959          |
| 260               | 1                | 2040        | 120                  | 100                  | 13800                      | 2956          |
| 260               | 1                | 2040        | 140                  | 120                  | 13800                      | 2953          |

Table 4. Heat Dissipated by Conduction

| Conduction to Bearing Shaft | Conduction to Drive Side Attachment |
|-----------------------------|-------------------------------------|
| 1391.588 W                 | 3969.518 W                          |

Table 5. Heat Dissipated by Radiation

| Radiation at Range of 140 °C & 120 °C | Radiation at Range of 120 °C & 100 °C | Radiation at Range of 110 °C & 90 °C |
|--------------------------------------|--------------------------------------|-------------------------------------|
| 165.21 W                             | 117.67 W                             | 96.47 W                             |

6. Results and Discussion

Radiation shows much better efficiency in vacuums and higher temperatures (of over 1000K). In the current scenario, it does not contribute much to heat dissipated in the system. An alternative I would suggest is to implement a system for conduction instead. A thrust washer can be used to fill the gap between the bearing and the casing.
According to calculations, it has been seen that oil with lower viscosity leads to a lower loss in power as well. The variation in power loss can be seen as shown in the table. Optimum viscosity oil can be decided upon using these findings. Currently, the oil bath is at a height of half the bearing height, but at lower heights, the power loss decreases significantly. The resulting power loss can be seen from Table. 6. to Table. 11.

6.1 Viscosity Dependent:

The variation in total frictional moment and power loss depending on variation in viscosity of oil lubricant for the temperature range of 120 °C to 100 °C is seen in Table. 6. The power loss decreases with decrease in viscosity with the highest power loss of 6616 W at a viscosity of 240 mm²/s and lowest power loss of 6329 W at a viscosity of 180 mm²/s.

| Viscosity (mm²/s) | Total Frictional Moment (Nmm) | Power Loss (W) |
|-------------------|-------------------------------|----------------|
| 220               | 30500                         | 6527           |
| 200               | 30100                         | 6432           |
| 180               | 29600                         | 6329           |
| 240               | 30900                         | 6616           |

Figure 2. shows the power loss vs. viscosity at an inner ring temperature of 120 °C and outer ring temperature of 100 °C. The power loss decreases with decrease in viscosity with the highest power loss of 6616 W at a viscosity of 240 mm²/s and lowest power loss of 6329 W at a viscosity of 180 mm²/s. It shows steady growth in power loss with higher viscosity of oil lubricant.

The variation in total frictional moment and power loss depending on the variation in viscosity of the oil lubricant for temperature range of 140 °C / 120 °C can be seen in Table. 7. The power loss decreases with decrease in viscosity with the highest power loss of 5929 W at a viscosity of 240 mm²/s and lowest power loss of 5724 W at a viscosity of 180 mm²/s.
Table 7. Variation of Total Frictional Moment and Power Loss with Viscosity
(at temperature 140°C to 120°C)

| Viscosity (mm²/s) | Total Frictional Moment (Nmm) | Power Loss (W) |
|------------------|-------------------------------|---------------|
| 220              | 27400                         | 5864          |
| 200              | 27100                         | 5797          |
| 180              | 26700                         | 5724          |
| 240              | 27700                         | 5929          |

Figure 3 shows the graph of power loss vs. viscosity at an inner ring temperature of 140 °C and outer ring temperature of 120 °C. The bar graph, like the previous figure shows a steady increase in power loss with viscosity of the oil lubricant.

![Power Loss vs. Viscosity (140°C - 120°C)](image)

Figure 3. Variation of Power Loss with Viscosity (at temperature 140°C to 120°C)

The variation in total frictional moment and power loss depending on the variation in viscosity of the oil lubricant for the temperature range of 110 °C/90 °C is seen in Table 8. The power loss once again decreases with decrease in viscosity with the highest power loss of 7078 W at a viscosity of 240 mm²/s and lowest power loss of 6742 W at a viscosity of 180 mm²/s.

Table 8. Variation of Total Frictional Moment and Power Loss with Viscosity
(at temperature 110°C to 90°C)

| Viscosity (mm²/s) | Total Frictional Moment (Nmm) | Power Loss (W) |
|------------------|-------------------------------|---------------|
| 220              | 32600                         | 6973          |
| 200              | 32100                         | 6863          |
| 180              | 31500                         | 6742          |
| 240              | 33100                         | 7078          |
Figure 4. shows the graph of power loss vs. viscosity at an inner ring temperature of 110 °C and outer ring temperature of 90 °C. The bar graph once again shows a steady increase in power loss with viscosity of the oil lubricant.

![Power Loss vs. Viscosity (110°C - 90°C)](image)

**Figure 4.** Variation of Power Loss with Viscosity (at temperature 110°C to 90°C)

Figure 5. shows a line graph comparing the power loss at various temperatures ranges with respect to viscosity and it can be seen that highest power loss is noted at lower temperature range of 110 °C/90 °C and the lowest power loss is noted at the higher temperature range of 140 °C / 120 °C. However, in all three temperature ranges, the power loss steadily increases with increase in viscosity.

![Viscosity Dependent](image)

**Figure 5.** Variation of Power Loss with Viscosity

6.2 Oil Bath Height Dependent:

The variation in total frictional moment and power loss depending on variation in oil bath height for the temperature range of 120 °C / 100 °C is seen in Table. 9. The power loss decreases with decrease in oil bath height with the highest power loss of 6807 W seen at a height of 70 cm and lowest power loss of 6326 W at a height of 58 cm.
Table 9. Variation of Total Frictional Moment and Power Loss with Oil Bath Height
(at temperature 120°C to 100°C)

| Oil Bath height (cm) | Total Frictional Moment (Nmm) | Power Loss (W) |
|----------------------|-------------------------------|----------------|
| 63                   | 30500                         | 6527           |
| 60                   | 29900                         | 6406           |
| 58                   | 29600                         | 6326           |
| 70                   | 31800                         | 6807           |

The variation in total frictional moment and power loss depending on variation in oil bath height for the temperature range of 140 °C / 120 °C is seen in Table. 10. The power loss once again decreases with decrease in oil bath height with the highest power loss of 6144 W seen at a height of 70 cm and lowest power loss of 5644 W at a height of 58 cm.

Table 10. Variation of Total Frictional Moment and Power Loss with Oil Bath Height
(at temperature 140°C to 120°C)

| Oil Bath height (cm) | Total Frictional Moment (Nmm) | Power Loss (W) |
|----------------------|-------------------------------|----------------|
| 63                   | 27400                         | 5864           |
| 60                   | 26800                         | 5744           |
| 58                   | 26500                         | 5644           |
| 70                   | 28700                         | 6144           |

The variation in total frictional moment and power loss depending on variation in oil bath height for the temperature range of 110 °C / 90 °C is seen in Table. 11. The power loss seen is the highest at a height of 70 cm with 7254 W and lowest power at 58 cm with 6773 W.

Table 11. Variation of Total Frictional Moment and Power Loss with Oil Bath Height
(at temperature 110°C to 90°C)

| Oil Bath height (cm) | Total Frictional Moment (Nmm) | Power Loss (W) |
|----------------------|-------------------------------|----------------|
| 63                   | 32600                         | 6973           |
| 60                   | 32000                         | 6853           |
| 58                   | 31700                         | 6773           |
| 70                   | 33900                         | 7254           |

Figure 6 shows a line graph comparing the power loss at various temperatures ranges with respect to oil bath height and it can be seen that the highest power loss is noted at the lower temperature range of 110°C to 90°C and the lowest power loss is noted at the higher temperature range of 140°C to 120°C. However, in all three temperature ranges, the power loss steadily increases with increase in oil bath height.
Figure 6. Variation of Power Loss with Oil Bath Height

The highest power loss was seen at a lower temperature range of 110°C to 90°C whereas the lowest power loss was seen at a higher temperature range of 140°C to 120°C in both conditions. Calculations were done for decreasing power loss using alternate means and the optimum power loss was seen at 58 cm height of the oil bath at the temperature range of 140°C to 120°C and at 180 mm²/s viscosity, once again at the temperature of 140°C to 120°C.

7. Conclusion

The current heat generation and dissipation paths for the bearing on the exciter shaft of the compactor were studied and calculated. Various optimization methods were devised for heat dissipation to improve the wear as well as efficiency of the bearing and further testing is required for implementation of the same.

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