Stress and fatigue analysis of the hydraulic motor shaft and conveyor joints of a tractor-mounted Chinese cabbage collector

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Abstract. Deflection and fatigue damage in a tractor-mounted Chinese cabbage collector might occur due to continuous operation. Therefore, the objective of this study is to predict the service life of the material used for the hydraulic motor shaft and conveyor joints based on stress and fatigue analysis. The torque produced by the motor was measured using the torque sensor for various tractor forward speeds and rotational speeds under unloaded and loaded conditions with three iterations. The relevant shear stress with the S–N curve on fatigue life was determined using the Smith–Watson–Topper equation and rain flow cycle counting method. Additionally, the severity level was investigated using Miner’s rule. Design variables of the shaft diameter and conveyor's joints were incrementally increased by 0.02 m. The experimental results of deflection and shear stress of the components were found to be negligible. The result of fatigue lifecycles was also generated over $10^7$. The maximum severity level of the motor shaft, conveyor joints 1 and 2 were 4.27 and 4.48, 4.26 and 5.58, and 4.12 and 4.46, respectively under unloaded and loaded conditions. When the values of design variable components were increased incrementally, shear stress and deflection decreased gradually but the fatigue life increased. Nevertheless, the results indicated that this machine could be safely used in the field.

1. Introduction

Hydraulic motor shaft is a rotating device under dynamic load for every revolution. Most welded structures of Chinese cabbage collector are subjected to multiaxial fatigue load and failure initiated from weld joints. Fatigue failure prediction is important for determining the effective design parameters for final production. The best-known fatigue analysis methods are stress-life, strain-life, and crack propagation methods [1]. The fatigue life of a material is estimated depending on the operating conditions, design assumptions, material properties and load conditions [2].
The operational characteristics of the cabbage collector were analyzed in terms of annual work hours, working speed, stress, load distribution, and fatigue life. Moreover, the rain flow cycle counting method considered the sequence effect for counting the cycles of the measured load loops [3]. Therefore, the main objective of this study is to perform stress and fatigue analysis of the Chinese cabbage collector and predict the reliable service life and the optimized dimension of the materials.

2. Materials and Methods

2.1. Structure of the Chinese cabbage collector

Figures 1 and 2 show the 3D structure of the Chinese cabbage collector and schematic diagram of the transfer conveyor, respectively. The conveyor comprises of four major components, i.e., collecting conveyor, transfer conveyor, hydraulic motor and hydraulic cylinders. To use this implement, farmers need to manually harvest and load the cabbages on the collecting conveyor. Then, the cabbages are packed using polypropylene bags and transported to the final destination. Before and after the operation, the collecting conveyor is unfolded or folded using hydraulic cylinders. The weight and dimension of the prototype collector were 850 kg and $7 \times 1 \times 1.24$ m (length $\times$ width $\times$ height), respectively. The average mass of the Chinese cabbage was 3 kg and its diameter and height were in the range of $202.5 \pm 10.4$ to $243.8 \pm 11.9$ mm and $281.9 \pm 1.46$ to $321.3 \pm 1.96$ mm, respectively [4].

2.2. Torque measurement of hydraulic motor shaft

The hydraulic motor shaft was subjected to stress and fatigue analysis using the experimental method. A torque sensor (Futek TRS-605, California, USA) was used to measure the torque of the hydraulic motor shaft (2.4 kW) at various speeds of 20, 60, and 120 rpm under unloaded and loaded conditions. The tests were conducted at various ground speeds of 1.36, 1.55, and 1.73 km/h using the 41-kW tractor engine. Each test was iterated thrice. The S–N curve of the motor shaft was simulated as per the ASTM standard for SCM420H carbon steel for various motor speed and tractor forward speed under both loading conditions using equation (1). The tensile and fatigue strengths of the material were estimated to be 1,817 and 700 MPa, respectively [5].

$$ N = 10^{ \left[ 6 - 6.097 \log \left( \frac{S}{223} \right) \right] } \quad \text{(1)} $$

The effects of the cabbage collector operations on the fatigue life of the motor shaft was estimated using the partial damage theory and compared in terms of severity of the fatigue strength of the material. The equivalent torque of the load spectrum was converted to shear stress as using equation (2).
where, \( N \) is the number of cycles when the shear stress is \( S \) (MPa), and \( D \) is the shaft diameter (mm).

2.3. Torque calculation of conveyor’s joints

The joints of the collecting conveyor and transfer conveyor (Joints 1 and 2 in Figure 1) were connected by welding as moving parts can cause serious or fatal injuries under continuous operations. The required hydraulic power for the conveyor was divided between two parts, i.e., collecting conveyor and transfer conveyor. The tensile load was applied to the conveyor due to the belt. The hydraulic power of the collecting conveyor [6] and transfer conveyor [7] can be expressed using equations (3-6).

\[
P_{cu} = \left( 0.06 \times f \times W \times v \times \frac{L + L_0}{367} \right) + \frac{L + L_0}{367} + \frac{H \times Q_y}{367} \tag{3}
\]

\[
P_{cl} = \left( 0.06 \times f \times W \times v \times \frac{L + L_0}{367} \right) + f \times Q_y \times \frac{L + L_0}{367} + \frac{H \times Q_y}{367} \tag{4}
\]

\[
Torque = \frac{P}{rpm} 
\]

\[
P_t = \frac{T_s \times v}{3.6} \tag{6}
\]

where, \( P_{cu} \) is the hydraulic power of the collecting conveyor under unloaded condition (kW), \( P_{cl} \) is the hydraulic power of the collecting conveyor under loaded condition (kW), \( P_t \) is the hydraulic power of the transfer conveyor (kW), and \( T_s \) is the effective belt tension at the drive pulley (kg).

The shear stress and deflection of the components were investigated from the experimental torque result using equations (7-8) under unloaded and loaded conditions.

\[
\delta = \frac{F \times L}{A \times E} \tag{7}
\]

\[
J = \frac{1}{3} d t_w^3 \tag{8}
\]

The S–N curve for the material of the conveyor joints was simulated using the Basquin equation as per the ASTM standard for S275 mild steel [8]. The yield strength and modulus of elasticity of the S275 mild steel were 0.275 and 40 GPa, respectively. To find the damage sum, the equivalent torque of the load spectrum of the material was converted to shear stress using equations (9-10).

\[
S = 413.69 N^{-0.069} \tag{9}
\]

\[
S = \frac{Torque \times t_w}{J} \tag{10}
\]

where, \( \delta \) is the deflection (mm), \( F \) is the force (N), \( E \) is the modulus elasticity (MPa), \( J \) is the torsion constant, \( d \) is the distance (m), \( t_w \) is the web thickness (m), and \( S \) is the maximum shear stress (MPa).

2.4. Severity calculation of damage

The operational load of the collector was analyzed by the load signal of the motor shaft as torque data, load spectrum, relevant stress, and related damage ratios under both load conditions. The effect of mean torque load was removed to find the spectrum magnitude using the Smith–Watson–Topper
equation and the rain flow cycle counting method [9]. The total number of cycles at a particular magnitude of torque for the overall life of the collector was calculated using equations (11-12).

\[
T_e = \sqrt{(t_m + t_a)H_a} \\
N_T = 3600 NLh
\]

The overall life of the collector was assumed as 10 years and 3000 h. The severity of the operation was represented by using equation (13) through the ratio of the damage sum. The lowest magnitude of the damage sum at all operational speeds was expressed by normalization using Miner’s rule [10].

\[
D_i = \sum_{i=1}^{k} \frac{n_i}{N_i}
\]

where, \( T_e \) is the equivalent torque (Nm), \( t_m \) is the mean torque, \( t_a \) is the measured torque amplitude (Nm), \( N_T \) is the total number of load cycles, \( N \) is the number of calculated cycles for the measured load (cycle/s), \( L \) is the entire life of the collector (year), \( h \) is the annual usage time hour (h/year), \( D \) is the damaged sum, \( n_i \) is the number of applied cycles at the \( i^{th} \) torque class, and \( N_i \) is the fatigue life at the \( i^{th} \) torque class (cycles).

2.5. Design consideration of the collector

An effective and efficient material handling system design is required that can increase productivity and minimize cost. The dimension of the design variables, such as shaft diameter and distance and web thickness of conveyor joints, were incrementally varied at the rate of 0.02 m from 0.1–0.3 mm, 0.05–0.25 mm, and 0.3–0.5 mm, respectively, to analyze the resulting shear stress, deflection, and fatigue life under the loaded condition. Table 1 lists the various parameter dimensions of the Chinese cabbage collector.

| Item | Specification (m) | Item | Specification (m) |
|------|-------------------|------|-------------------|
| \( L_c \) | 4.5 | \( D \) | 0.016 |
| \( L_{th} \) | 2.5 | \( d \) | 0.1 |
| \( H \) | 0.72 | \( t_w \) | 0.4 |

3. Results and Discussion

3.1. Stress behavior of hydraulic motor shaft

The maximum torque levels of the motor shaft for various rotational speeds at 120 rpm and 1.73 km/h tractor ground speed were 0.6579, 0.7443, and 0.8645 Nm, and 0.6853, 0.9985, and 1.4749 Nm, respectively, under unloaded and loaded conditions. Moreover, the minimum and maximum shear stresses of the motor shaft were determined to be between 0.1213 ~ 0.1266 \times 10^{-3} \text{ MPa} and 0.1164 ~ 0.1585 \times 10^{-3} \text{ MPa} at the highest rotational speed and tractor ground speed under unloaded and loaded conditions, respectively, as shown in Figures 3 and 4.
Figures 5 and 6 display the time history of the load and stress spectrum of the motor shaft for all rotational speeds and tractor forward speeds under unloaded and loaded conditions. The motor shaft could endure more than $10^7$ cycles under all operating conditions. Ten million cycles is a reasonable fatigue life for any kind of machinery [11]. The highest and lowest number of cycles was observed at 20 and 120 rpm, respectively, under both load conditions. The maximum shear stress at 120 rpm was 229 and 230 MPa, respectively. The maximum torque ratio obtained was 0.0175, 0.0186, and 0.0226, and 0.0171, 0.0194, and 0.0228, respectively, at 20, 60 and 120 rpm under the same conditions.

The relationship between the severity of the motor shaft and operational speed was described in Figure 7. The severity levels of 1, 1.28, and 4.27, and 1, 2.88, and 4.48, respectively, were represented by the ratio of the damaged sum to the smallest under the unloaded and loaded collections. When the motor speed increased, the damage level also increased up to 100, 128, and 427%, and 100, 288, and 448%, respectively at 20, 60, and 120 rpm under both load conditions.
3.2. Stress behavior of conveyor's joints

![Figure 8. Shear stress on Joint 1 for forward speeds under unloaded condition.](image1)

![Figure 9. Shear stress on Joint 1 for forward speed under loaded condition.](image2)

The experimental results showed that the shear stress of the conveyor Joint 1 was approximately $0.0001 \sim 0.0002 \times 10^{-3}$ MPa for all tractor forward speeds and operational speeds under both load conditions, as illustrated in Figures 8 and 9. Similarly, the shear stress of Joint 2 varied between $0.0083 \sim 0.0494 \times 10^{-3}$ MPa and $0.0097 \sim 0.0519 \times 10^{-3}$ MPa, respectively, under unloaded and loaded conditions.

The load and stress spectrum of Joints 1 and 2 were observed under both load conditions. The results suggested that all operating conditions could run more than $10^7$ lifecycles. The shortest and longest cycles of the joints at 120 and 20 rpm were related to the highest and lowest torque ratios, respectively. The maximum shear stresses at Joints 1 and 2 at 120 rpm were 161 and 163 MPa, and 160 and 162 MPa, respectively. The maximum torque ratio of Joint 1 was obtained at 0.0127, 0.0141, and 0.0161 and 0.0127, 0.0145, and 0.0163, respectively, at 20, 60, and 120 rpm. Similarly, the maximum rated torque ratio of Joint 2 was observed at 0.0007, 0.0053, and 0.0101, and 0.0009, 0.0063, and 0.0199, respectively under both load conditions.

The severity level of 1, 1.31, and 4.26, and 1, 1.32, and 5.58 of Joint 1 were obtained, respectively, by the ratio of the damaged sum to the smallest under unloaded and loaded conditions for the operational speed of 20, 60, and 120 rpm. Likewise, the severity level of Joint 2 was found at 1, 3.19, and 4.12, and 1, 4.12, and 4.46, respectively, under both load conditions at 20, 60, and 120 rpm. Moreover, severity level of the joints is related to an operational speed, as shown in Figures 10 and 11. When the motor speed was higher, the damage level of the joints was also increased by 100, 132, and 558%, and 100, 412, and 446%, respectively, for 20, 60, and 120 rpm under both load conditions.

![Figure 10. Damage severity of Joint 1 under various load conditions.](image3)

![Figure 11. Damage severity of Joint 2 under various load conditions.](image4)

3.3. Design consideration

The correct size of the shaft diameter helps selection of the belt width and pulley diameter [12]. When the shaft diameter was incrementally increased, shear stress and deflection continuously decreased but fatigue life increased, as displayed in Figures 12 and 13. Based on the Korean steel structure design
code and commentary (2018), one-third of the span can be considered as allowable deflection. However, the amount of shear stress and deflection was found to be negligible.

The correct joint structure can be implemented by determining the stress distribution at a conveyor's joint. When the web thickness and distance of the conveyor were sequentially increased, the result of shear stress and joint deflection gradually decreased until they became negligible. However, the fatigue lifecycles of joints 1 and 2 increased continuously up to $3 \times 10^{14}$ and $9 \times 10^{19}$, respectively.

4. Conclusions
The results of this study indicated that the shear stress and deflection experienced by the motor shaft and conveyor's joints were negligible during the operation. All operating conditions could perform more than $10^7$ cycles. Moreover, the maximum severity level of damage for the motor shaft, conveyor joints 1 and 2 were 4.27 and 4.48, 4.26 and 5.58, and 4.12 and 4.46 under unloaded and loaded conditions. With respect to design parameters, when the shaft diameter, distance, and web thickness of the conveyor were successively increased, shear stress and deflection gradually decreased. Additionally, the number of cycles also increased. According to the test results, the motor shaft and conveyor's joints were strengthened for the operation. Based on the results obtained, this machine can be safely used in the working field. Additionally, this study can be extended to understand stress and fatigue concentration due to unusual load conditions and environmental issues.

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