Design modification and failure analysis of scissor jack

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ABSTRACT

This paper relates to design modification and failure analysis of crank-operated scissor jack which is generally used for lifting light motor vehicles (L.M.V) during maintenance. This project is focused on designing and finding various stresses and expected a life of various parts of scissor jack like power screw, base plate, etc using different modeling and analytical software.

Keywords: Scissor Jack, Failures, Analysis.

1. INTRODUCTION

The scissor jack is the one of the most important mechanical component used for lifting of load in application such as cars, lifts. The cost of jack is a major concern and has forced people to look for alternative jacks that are available abundantly and are cheaper and compact. The different alternatives available are like scissor jack, hydraulic jack, pneumatic jack, floor jack, etc. Amongst these alternatives available, scissor jack is widely used for lifting vehicles. A scissor jack is the most common type of jack which may have encountered as there jacks comes along with cars. It is used for road side repairs. They are light in weight and the cost of the scissor jack is reasonably small and scissor jacks are easy to store and easy to operate at any time and any place in the different environment condition. It is use for the average consumer / vehicle owner. It is simple in design and working, it is having some component like carrier plate, upper arms, lower arms, base plate, middle pins and power screw. The jack is working on the scissor lifting mechanism. The power screw is the heart of scissor jack and scissor jack lift the load with the help of screw and nut interaction of power screw and middle pin. Scissor jack lift the load due to which the rotation motion is converted into linear motion and due this motion it is easy to lift the load and different range of scissor jacks are available according to the different load.

Figure 1 Scissor Jack
2. PROBLEM DESCRIPTION
In the real scenario, it has been seen that many people face the problem that scissor jack fails during maintenance. According to the observations so far it has been found out that two out of four scissor jacks fail in short time. There are many issues related to the scissor jack which are forcing to find out the reason of jack failure. The disadvantage of the scissor jack is worn out the thread of power screw so the scissor jack fails during the time of maintenance.

Figure 2 Problem Description

3. OBJECTIVES
- To make such a modified scissor jack that is very stable and it can take enough load on uneven surfaces and somewhat inclination is also allowed.
- To find out the component in which initial failure occurs.
- To find the type of failure.
- To find the reason for failure.
- To find out the flaws existing in the design of power screw.

4. MEASUREMENTS AND CALCULATIONS
A. Analytical Calculations:

B. Measured design Parameters:
- By measuring linear dimensions, it is found that
  - Mean diameter of screw = 10 mm
  - Core diameter of screw = 7 mm
  - Outer diameter of screw = 12 mm
  - Pitch of screw = 3 mm
  - Length of nut = 15 mm
  - Number of screw threads engage with nut = 5
  - Maximum allowable load \( W = 700 \text{Kg} = 6867 \text{ N} \)
  - Coefficient of friction = 0.15

C. Calculated Design Parameters: By elementary static analysis from figure 4,
- \( \sin \theta = h/2l \)
• \( \sin \theta = \frac{55}{150} \)
• Thus, \( \theta = \left[ 21.51 \right]^\circ \)
• Also, \( \tan \theta = \frac{w}{2t} \)
• Thus, \( t = \frac{w}{2 \tan \theta} \)
• \( t = 8711.90 \text{ N} \)
• Now, from the figure it is seen that \( t \) acts in both directions. Thus, total load on screw \( p = 2t = 17,423.8 \text{ N} \).
• Torque \( T' = \frac{(p \cdot d \cdot m \cdot \tan \theta (\theta + \lambda))}{2} \) where, \( \tan \lambda = \mu \)
• \( T' = 29.20034 \text{ Nm} \).

D. Design Calculations:
• From figure 4,
• \( N = p / ((\cos \theta - \mu \sin \theta)) \)
• Now the value of \( \theta \) and \( \mu \) is very small,
• so \( \mu \sin \theta \) can be neglected
• And \( \cos \theta = 1 \)
• Thus, taking \( N = P \)
• \( F_t = N (\sin \theta + f \cos \theta) \)
• \( F_t = N (\cos \theta + \mu \sin \theta) \)

E. Stress Calculations:
On screw, there are mainly 6 types of failures. They are as follows:-

I) For screw body
• Tensile failure
• Buckling failure
• Twisting failure

II) For screw thread
• Crushing failure
• Bending failure
• Shear failure

Thus the stresses induced in components for the above-listed failure are as calculated below.

<table>
<thead>
<tr>
<th>Sr No.</th>
<th>Element</th>
<th>Material</th>
<th>Ultimate Tensile Strength (Mpa)</th>
<th>Yield Tensile Strength (Mpa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.</td>
<td>Screw</td>
<td>Mild/low carbon steel</td>
<td>440</td>
<td>370</td>
</tr>
<tr>
<td>2.</td>
<td>Nut</td>
<td>Cast iron</td>
<td>820</td>
<td>----</td>
</tr>
</tbody>
</table>
I) For Screw

a) Tensile stress,
\[ \sigma_t = \frac{F}{A} = \frac{4F}{\pi D_m^2} \]
Thus, \( \sigma_t = 221.84 \text{ N/mm}^2 \)

b) Torsional shear stress,
\[ \tau = \frac{16T}{\pi D_m^3} \]
Thus, \( \tau = 148.74 \text{ N/mm}^2 \)

II) For Screw Thread,

For power screw, it is seen that the force exerted on the 1st thread will be 38% of total load and after it will be reduced for other consecutive threads. It is seen that the force after the 7th thread is negligible. So to get better design the value has been taken as \( n = 1 \) and \( P = 0.38P_{\text{total}} \).

a) Bearing pressure,
\[ P_b = \frac{P \pi h D_m}{n} \]
Thus, \( P_b = 140.5025 \text{ N/mm}^2 \)

b) Shear stress,
\[ \tau_{\text{avg}} = \frac{P_b}{h} \]
For square threads, \( h = b = \text{pitch}/2 \)
Thus, \( \tau_{\text{avg}} = P_b = 140.5029 \text{ N/mm}^2 \)
Maximum shear stress induced is
\[ \tau_{\text{max}} = 3\tau_{\text{avg}}/2 = 210.753 \text{ N/mm}^2 \]

c) Bending stress,
\[ \sigma_b = \frac{M_y}{I} = \frac{P_b}{(4b^2 \times 12h^2 b)} \]
Thus, \( \sigma_b = 421.5087 \text{ N/mm}^2 \)

Thus, following result from analytical calculations are obtained.

- For screw body.

<table>
<thead>
<tr>
<th>Table 2 Stresses in the screw body</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Type</strong></td>
</tr>
<tr>
<td>Axial tensile stress</td>
</tr>
<tr>
<td>Torsional shear stress</td>
</tr>
</tbody>
</table>

- For screw thread.

<table>
<thead>
<tr>
<th>Table 3 Stress in the screw thread</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Type</strong></td>
</tr>
<tr>
<td>Bearing pressure</td>
</tr>
<tr>
<td>Shear stress</td>
</tr>
<tr>
<td>Bending stress</td>
</tr>
</tbody>
</table>
5. SOLID MODELING

Figure 5 Scissor Jack assembly

6. STATIC STRUCTURAL ANALYSIS IN ANSYS

Boundary conditions:
The induced stress by analytical calculations is now checked using Ansys in workbench static structural module.

Analysis of stress induced in screw body:
The required components for screw body analysis are nut, washer, and screw. Thus other components are suppressed and the various boundary conditions required for specified components are shown in below figure.

Figure 6 Arrangement and boundary condition for analysis of screw

The solutions obtained from the analysis are shown below.

Figure 7 Stresses induced in the screw body
Figure 8 Total deformations in the screw body

Analysis for screw thread:

If the analysis of screw thread is done directly on the 3D model then it will be time-consuming and will require high computer memory for processing. Thus the analysis of screw thread is generally done by alternate methods which are accurate and requires less time and memory. The analysis of screw thread is generally done by using 3 methods which are:

- Analysis by using bolt thread feature.
- Analysis by using bolt pretention feature.
- Analysis by doing 2D analysis in the workbench.

Here the 2D analysis of screw thread is used as it is very accurate than other methods. The required components for screw body analysis are a 2D nut and 2D screw. The arrangements and various boundary conditions required for analysis are shown in below figure.

Figure 9 Arrangement and boundary condition for analysis of screw thread

The solutions obtained from the analysis are shown below.
CONCLUSION

Hypothetically, it was considered that the scissor jack was failed at some load under the prescribed limit. The induced stress calculated by ANSYS, which shows that the scissor jack fails under the prescribed limit of the load. Further, the result validated with analytical calculations.

So, there must be a requirement of modification in the design. To sustain the scissor jack under the prescribed limit of the load.

FUTURE WORK TO BE DONE

As it has been seen that the scissor jack fails under the critical load the problem has been defined.

After the definition of problem, a specific material is to be decided to reduce the cost and increase the strength of scissor jack up to the given load.

A new material should be then tested analytically and experimentally and then that can be concluded the problem of the solution.

WORK PLAN
10. REFERENCES