Power Transmission Design & Analysis Report

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I. Introduction

ATV Solutions is developing a mobile, portable scissor lift for at home use. Most scissor lifts are primarily sold for commercial use and are much too large and expensive for the common consumer. However, repairs and renovations are a critical part of home ownership and are made easier through the use of a scissor lift. These tasks often involve hard-to-reach places, whether it's a routine job such as cleaning the gutters or a more ambitious project like repainting a home's exterior. Any job high above the ground is inherently dangerous, especially for older adults and those with physical impairments. This risk is amplified when using a ladder, which can be extremely unstable if improperly secured. This product provides the ordinary homeowner with a safer and easier way to carry out these tasks for a reasonable cost.

In this report, the ATV Solutions team outlines the design for the power screw and gearbox power transmission system for the scissor lift. Since the linear vertical speed of the scissor lift must be low in order to ensure safety, a gear ratio of 4 to 1 was chosen for the design. Since it is operating outdoors, all components of the power transmission system are plated to ensure corrosion resistance. This report analyzes the selected power screw in order to calculate the required output torque and speed of the gearbox, which dictates the chosen motor specifications. For the power screw calculations, the forces calculated in the components report were utilized. It is also assumed that the greatest load required to lift is 900 lb.

Lastly, these calculations were used to determine the final factors of safety for the design of this scissor lift in order to ensure that this product will be both sufficiently safe and affordable.

II. Power Screw

a. Power Screw Overview

The power screw is a crucial component in the power transmission system as it is what drives the scissor lift assembly. The power screw attaches to both sides of the linkage assemblies to drive the horizontal motion of the scissor lift, which in turn raises the lift. The power screw can be seen in the assembly, shown in Figure II.1 below.

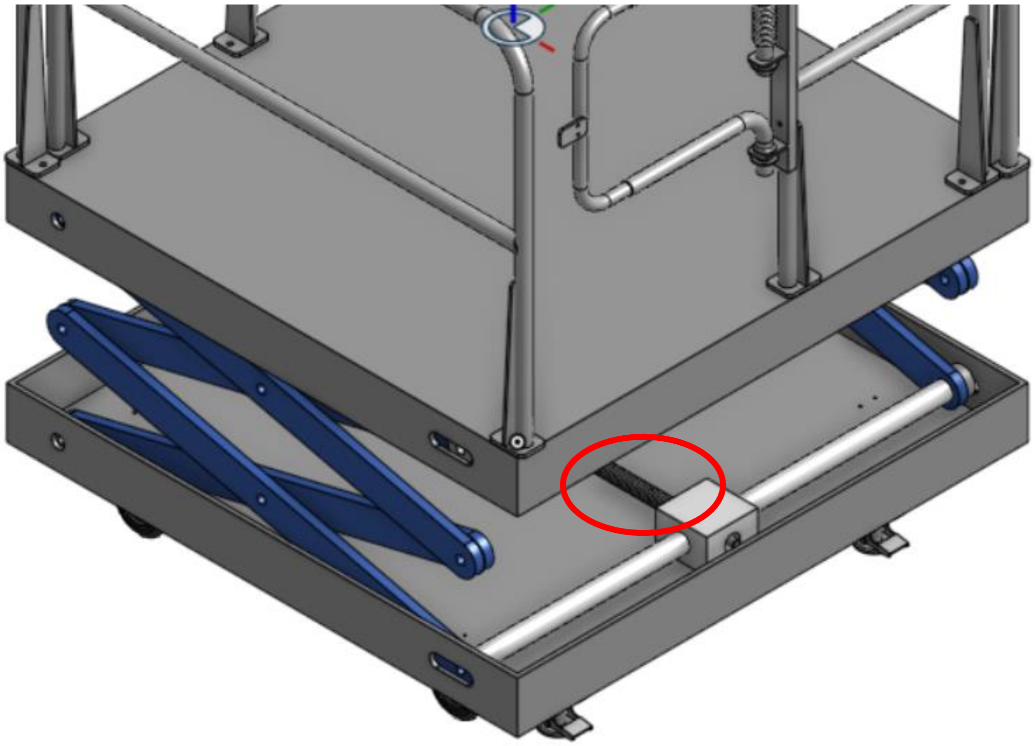

Figure II.1: Lead Screw Pictured in Full Assembly

b. Assumptions

The following assumptions were made during the power screw calculations:

- Axial force in the power screw for worst-case loading, based on components design report is 9.26 kip.
- Assume a mean diameter of the power screw of 44.45 mm (1.75 inches), due to standard size and availability
- Utilize power screw with 6.35 mm (0.25 inches) lead
- Assume 1 thread start for selected power screw
- Assume coefficient of friction of 0.15
- Assume no collar friction
- Lift at a vertical speed of 5 inches/minute
- 3.56 inches required horizontal motion of the scissor bar ends, based on components design analysis

- 29.562 inches required vertical motion of the scissor bar ends, based on components design analysis

c. Calculations

To calculate the linear speed of the lead screw, the desired time to raise the lift was found:

$$\frac{d_y}{v_y} = \frac{29.562\ in}{5\ in/min} = 5.9124\ minutes\ to\ raise$$

$$v_x = d_x t = \frac{3.56\ in}{5.9124\ min} = 0.602\ in/min$$

From here, the lead screw calculations can be completed

ACME Thread, mean diameter of 1.75 in, single thread, 14.5-degree pressure angle

$$d = 1.75\ in$$

$$n_t = 1$$

$$\alpha = 14.5°$$

From Table 7.1:

$$p = 0.25\ in$$

$$L = n_t p = 1 * .25 = 0.25\ in$$

$$d_r = d - p = 1.75 - 0.25 = 1.5\ in$$

$$d_m = \frac{1}{2}(d + d_r) = \frac{1}{2}(1.75 + 1.5) = 1.625\ in$$

$$\lambda = \tan^{-1}\left(\frac{L}{\pi d_m}\right) = \tan^{-1}\left(\frac{0.25}{\pi(1.625}\right) = 2.8036°$$

$$\alpha_n = \tan^{-1}(tan\alpha cos\lambda) = \tan^{-1}(\tan(20) * \cos(2.8036)) = 19.978°$$

$$T_R = \left(\frac{F_D d_m}{2}\right)\left(\frac{f\pi d_m + L cos\alpha_n}{\pi d_m cos\alpha_n - fL}\right) + \left(\frac{F_D f_c d_c}{2}\right)$$
$$= \left(\frac{9260 * 1.625}{2}\right)\left(\frac{0.15 * \pi * 1.625 + 0.25 * \cos(14.5)}{\pi * 1.625 * \cos(14.5) - 0.15 * 0.25}\right) = 1545.77\ lb * in$$

$$n = \frac{v_x}{L} = \frac{0.602}{0.25} = 2.408\ rpm$$

$$P = \frac{2\pi T_R n}{60 * 550 * 12} = \frac{2 * \pi * 1545.77 * 2.408}{60 * 550 * 12} = 0.059\ hp$$

d. Results

From these calculations, the required torque to drive the horizontal motion of the connecting rod is 1545.77 lb*in. The power required to drive the power screw was then calculated from 0.059 hp. These values will be used to calculate the required motor power in the following section.

III. Gearbox

a. Gearbox Overview

In order to transmit a reasonable amount of power from the motor to the lead screw, a gearbox is required. Gears were selected based on reasonable assumptions and available stock on common websites. These gears were selected to be spur gears with a gear ratio of 4 to 1 was selected with respective number of teeth of 48 and 12. This gear configuration can be seen below in Figure III.1

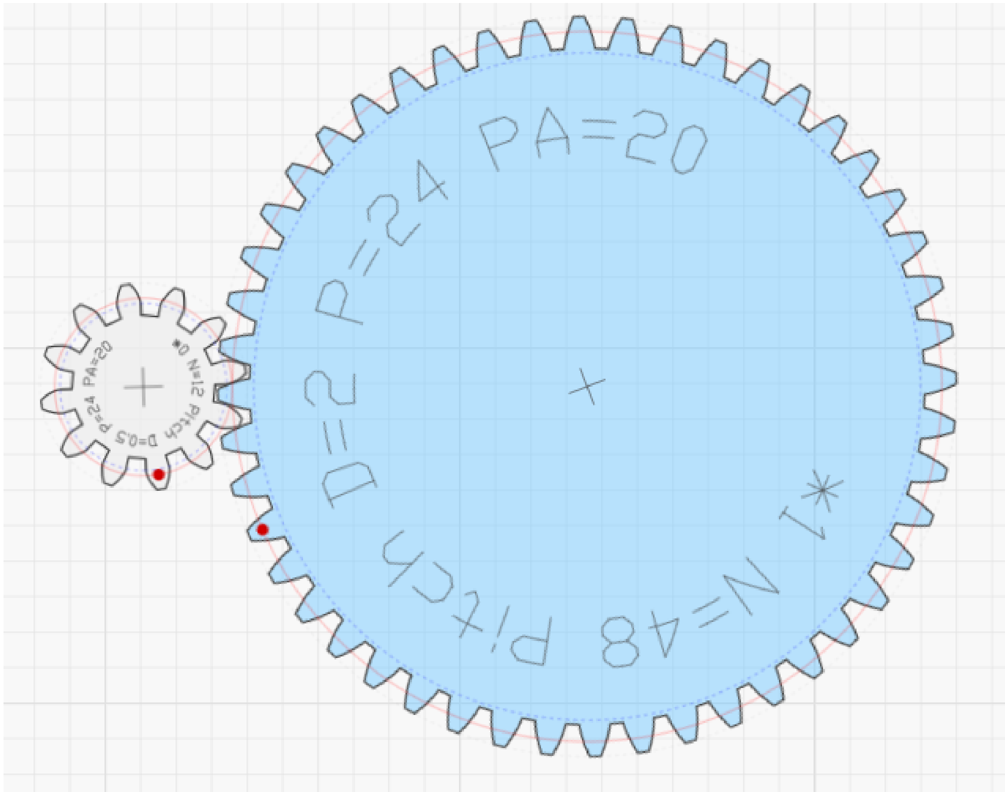

Figure III.1: Spur Gear Ratio Visual

b. Assumptions

The following assumptions were made before the gear box calculations were carried out.

- The material of the gears is 1144 Carbon Steel
- No temperature factor
- Moderate shocks and Impact
- Precision Ground and Shaved Gears
- Size Factor of 1
- Lifetime of 10 million cycles
- Reliability of 99%
- Pressure angle of 20 degrees

c. Calculations

The following calculations on the gearbox were carried out in order to solve for the power requirement of the motor and the safety factors in bending stress and contact stress respectively.

From selected gears, assumptions, and previous calculations:

$N_1 = 12\ teeth$

$N_2 = 48\ teeth$

$n_2 = 2.408\ rpm$

$p = 12\ teeth/in$

$b = 0.25\ in$

$n_1 = \frac{N_2}{N_1} n_2 = \left(\frac{48}{12}\right) * 2.408 = 9.632\ rpm$

Required motor power:

$P = \frac{2\pi T_R n_1}{60 * 550 * 12} = \frac{2 * \pi * 1545.77 * 9.632}{60 * 550 * 12} = 0.242\ hp$

Calculate bending stress:

Using the Pinion:

$$K_o = 1.75$$

$$B = 0.25(12 - 7)^{2/3} = 0.9148$$

$$A = 50 + 56(1 - B) = 54.770$$

$$V = \frac{\pi dn}{60} = \frac{\pi(0.5)(9.632)}{60} = 0.2522$$

$$K_v = \left(\frac{A + sqrt(V)}{A}\right)^b = 1.00838$$

$$K_s = 1$$

$$K_m = 1.6$$

$$J = 0.22$$

$$F_{t,pinion} = \frac{T}{d} = \frac{1.545}{.5 * \ .5} = 6.183\ lb$$

$$\sigma = F_t K_o K_v \frac{p}{b}\frac{K_s K_m}{J} = (6.183)(1.75)(1.00838)\frac{12}{0.25}\frac{(1)(1.6)}{0.22} = 1.904\ ksi$$

From Figure 14.2:

$$s_t = 2.5\ ksi$$

$$\sigma_{allow} = \frac{S_t Y_N}{K_t K_R} = \frac{(2.5)(1)}{(1)(1)} = \ 2.5\ ksi$$

$$n_{bending}\frac{\sigma_{allow}}{\sigma} = \frac{2.5}{1.904} = \mathbf{1.313}$$

Calculate bending stress:

Using the Pinion:

$$E_p = E_g = 200\ ksi$$

$$V_p = V_g = 0.28$$

$$C_p = \left[\frac{1}{\pi\left(\frac{1-v_p^2}{E_p} + \frac{1-v_g^2}{E_g}\right)}\right]^{1/2} = \left[\frac{1}{\pi\left(\frac{1-.28^2}{200000} + \frac{1-.28^2}{200000}\right)}\right]^{1/2} = 5.877\ (185.846?)$$

$$I = \frac{\cos(\varphi)\sin(\varphi)}{2}\frac{m_g}{m_g+1} = 0.1286$$

$$C_f = 1.25\ (assume\ rough\ finish)$$

$$\sigma = C_p\left[F_t K_o K_v K_s \frac{K_m}{db}\frac{C_p}{I}\right]^{\frac{1}{2}} = (185.846)\left[6.183(1.75)(1.00838)(1)\frac{1.6}{(0.5)(0.25)}\frac{1.25}{0.1286}\right]^{\frac{1}{2}}$$
$$= 6.847 ksi$$

$$S_c = 0.349HB + 34.300 = 0.349(140) = 83.16\ ksi$$

$$\sigma_{allow} = \frac{S_c Z_N C_H}{K_T K_R} = \frac{83.16(1)(1)}{(1)(1)} = 83.16\ ksi$$

$$n_{contact} = \frac{\sigma_{allow}}{\sigma} = \frac{83.16}{6.847} = \mathbf{12.145}$$

d. Results

For Bending Stress in the gear teeth, the overall safety factor is 1.313. For the Contact Stress in the gear teeth, the overall safety factor is 12.145. Both of these are within a normal safety range without being excessively wasteful.

IV. Conclusion

In conclusion, the lead screw and gearbox mechanism for the scissor lift will effectively raise and lower the load of the assembly. Both of the lead screw and the gears have safety factors against fatigue that are above 1, and thus infinite life is achieved. Additionally, the torque required is achieved and the scissor lift is able to raise and lower at the correct speed, 5in/min. ATV Solutions is confident in our design and its mechanical viability.

V. Appendix

Appendix A – Group Member Contributions

This contribution distribution was agreed on by all group members.

PJ Fries: 25% Contribution: Power Screw Calculations, Introduction, Report Writing

Emilie Hardel: 25% Contribution: Gear Calculations, Summary, Report Writing

Lauren Kreder: 25% Contribution: Power Screw Calculations, Gear Calculations, Report Writing

Justin Miller: 25% Contribution: Gear Stress Calculations, Report Writing