Crane Project

I. Description

The SkyCorp. crane was designed to run as smoothly and quickly as possible, using a large design factor to ensure reliability, and maintaining symmetry in the design. The overall ratio of load to exerted torque in the transmission system was 8:1, and the structural system was designed to bear a load almost twice the actual weight.

The transmission system uses a 4:1 compound gear train and a pulley system. The 10 pound weight is attached to a pulley, supported by a wire. One end of the wire is attached to the end of the beam, while the other end runs over a bushing-supported shaft at the end of the beam, and is tied around the third gear axle. The complete pulley system has a mechanical advantage of 2.

The gear train consists of four gears in total. The first gear axle, attached to the gearmotor assembly, holds a 20-tooth spur gear. This gear meshes with a 40-tooth spur gear on the second gear axle. On the same axle, another 20-tooth gear rotates at the same angular velocity and meshes with the final 40-tooth gear on the third gear axle. The mechanical advantage of the gear train is 4; the third gear axle rotates at ¼ the speed of the gearmotor, with 4 times the torque. As the third gear axle rotates, it puts tension in the cable before lifting the pulley and weight. All axles in the crane were supported with small nylon bushings to reduce friction.

The boom chosen for the crane was a ½" x 2" x 2' acrylic bar. A goal for the fabrication of the boom was to make as few changes to it as possible, in order to avoid compromising its mechanical properties. Two holes were drilled at the end of the beam, to which two ¼" acrylic supports were attached in order to hold the pulley axle in place. All fasteners used in the design were ¼-20 bolts and nuts, with some washers used.

At the supported end of the beam, four holes were drilled: two were used as attachment points for ⅛" load-bearing aluminum angle brackets, and the other two were used to hold together the support system. Adjacent to the boom, 2 ⅛" rectangular spacers, and 8 ⅛" circular spacers gave 1" of total space between two ⅛" thick, tall rectangular acrylic plates. These plates were made large in order to connect the beam supports and gear train with a solid piece of plastic. The structural system of the crane was designed to use few supports (the boom itself is only supported in 2 places) to minimize bending stresses in the boom.

II. Engineering CAD Drawings

See the following pages for engineering drawings 1-17. Drawing 16 is the exploded assembly, and Drawing 17 is the final assembly drawing.

III. Explanations/Justifications of Structural Design Factors

The design factor used in the design of the structure was 1.94. A structural design factor was calculated (see calculations below for justification) but the power transmission design factor was used as the structural design factor as well in order to ensure conservatism.

JOHNS HOPKINS UNIVERSITY DEPT OF MECHANICAL ENGINEERING 530.216 MBD LAB	CALCULATION STRUCTURAL DESIGN FACTOR (REFERENCE ONLY)		PAGE 1
	TEAM SKYCORP. INC.	TEAM MEMBERS PONCE, POLLARD, LAN	OF 1

$DF_{Structure} = DF_{Material} \cdot DF_{Stress} \cdot DF_{Geometry} \cdot DF_{Failure\ Analysis} \cdot DF_{Reliability}$

Material: 1.1 - Material properties are given on McMaster-Carr and MatWeb for all engineering materials used in the design.

Stress: 1.1 - Stress is well defined using basic beam bending analysis, no shocks are anticipated.

Geometry : 1.1 - Despite our best efforts, our machining will not be perfect - our ordered parts may also be under/oversized.

Failure Analysis: 1.0 - Very well-defined stress state.

Reliability: 1.4 - There is no margin for error; the crane must work perfectly every time, up to 100+ tests.

$DF_{Structure} = 1.86$

IV. Explanations/Justifications of Power Transmission Design Factors

The design factor used in the design of the power transmission was 1.94. See calculation below

JOHNS HOPKINS UNIVERSITY DEPT OF MECHANICAL ENGINEERING 530.216 MBD LAB	CALCULATION: Motor Design Factor Calculations		PAGE 1
	TEAM: SkyCorp. Inc	TEAM MEMBERS: Ponce, Pollard, Lan	OF 1

$$DF_{Motor} = DF_{Specs} DF_{Torque} DF_{Friction} DF_{Motor\ Power} DF_{Reliability}$$

Motor Specs: 1.05 - Fairly certain specs can be accurate - but there may be some variation between the given specs and the actual motor, which may have been in use for years

Torque: 1.0 - Known/Given, calculated with simple GR and pulley formulas

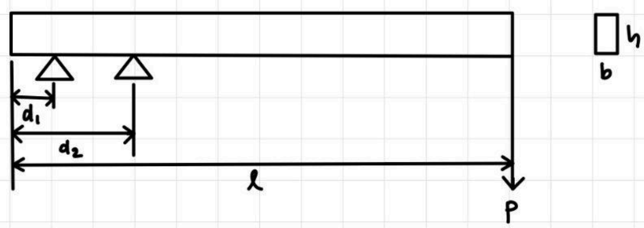
Friction: 1.1 - We have not accounted for friction in our design, but plan to use low-friction pulleys, bearings, etc.

Power: 1.2 - We may put the motor under variable loads during the lift, power needed is not 100% consistent.

Reliability: 1.4 - There is no margin of error, the crane must work perfectly during its demonstration, and must be available for many tests.

$$DF_{Motor} = 1.94$$

V. Design Calculations (see next page)



Knowns:

$P = 19.4$ lb
$b = 0.5$ in
$h = 2$ in
$d_1 = 1.5$ in
$d_2 = 5.25$ in
$l = 24$ in

Find: Maximum Bending Stress

Assumptions:

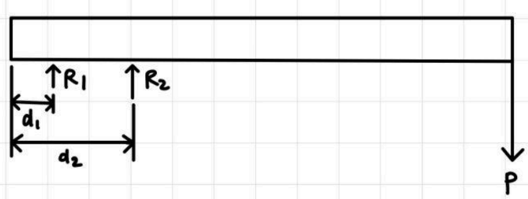
- the beam has a uniform cross section & weight
- the brackets that attach the boom to the mounting plate can be treated as simple supports

Design Factor = 1.94
(Weight of Load = 10 lb $\rightarrow$ (n.d.) $\times$ W = P = 19.4 lb)

Significant Equations:

$$\sigma_{max} = -\frac{M_{max} \cdot y}{I} \qquad I = \frac{1}{12} b h^3$$

Solution:



$$q(x) = R_1 \langle x - 1.5 \rangle^{-1} + R_2 \langle x - 5.25 \rangle^{-1} - P \langle x - 24 \rangle^{-1}$$

$$V(x) = R_1 \langle x - 1.5 \rangle^{0} + R_2 \langle x - 5.25 \rangle^{0} - P \langle x - 24 \rangle^{0}$$

$$M(x) = R_1 \langle x - 1.5 \rangle^{1} + R_2 \langle x - 5.25 \rangle^{1} - P \langle x - 24 \rangle^{1}$$

$$V(24) = R_1(1) + R_2(1) - P(1) = 0$$

$$R_1 + R_2 = P = 19.4 \text{ lb}$$

$$M(24) = R_1(24 - 1.5) + R_2(24 - 5.25) - P(0) = 0$$

$$R_1(22.5) + R_2(18.75) = 0$$

JOHNS HOPKINS UNIVERSITY DEPT OF MECHANICAL ENGINEERING 530.216 MBD LAB	CALCULATION Maximum Bending Stress on Boom	PAGE 2
	TEAM: Sky Corp Inc. / TEAM MEMBERS: Lan, Ponce, Pollard	OF 2

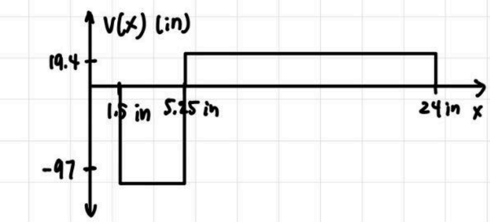
$R_1 + R_2 = 19.4 \rightarrow R_1 = 19.4 - R_2$

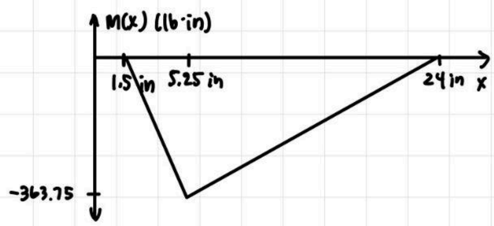
$22.5 R_1 + 18.75 R_2 = 0$

$R_2 = 116.4 \text{ lb} \uparrow \quad R_1 = -97 \text{ lb} = 97 \text{ lb} \downarrow$

$V(x) = -97\langle x-1.5\rangle^0 + 116.4\langle x-5.25\rangle^0 - 19.4\langle x-24\rangle^0$

$M(x) = -97\langle x-1.5\rangle^1 + 116.4\langle x-5.25\rangle^1 - 19.4\langle x-24\rangle^1$





$\sigma_{max} = -\frac{M_{max}\, y}{I} \quad I = \frac{1}{12} b h^3 = \frac{1}{12}(0.5'')(2'')^3 = 0.33 \text{ in}^4$

Top of beam:

$y = +\frac{h}{2} = +1 \text{ in}$

$\sigma_{max} = -\frac{(-363.75 \text{ lb·in})(1 \text{ in})}{0.33 \text{ in}^4}$

$= 1091.25 \text{ psi (tension)}$

Bottom of beam:

$y = -\frac{h}{2} = -1 \text{ in}$

$\sigma_{max} = -\frac{(-363.75 \text{ lb·in})(-1 \text{ in})}{0.33 \text{ in}^4}$

$= -1091.25 \text{ psi (compression)}$

∴ The maximum bending stress in the beam occurs at $x = 5.25$ in from the left end of the beam and has a magnitude of 1091.25 psi. The stress is compressive at the bottom of the beam and tensile at the top of the beam.

$$EI\,v(x) = -\frac{97}{6}\langle x-1.5\rangle^3 + \frac{116.4}{6}\langle x-5.25\rangle^3 - \frac{19.4}{6}\langle x-24\rangle^3 + 227.344\langle x\rangle^1 - 284.18\langle x\rangle^0$$

$$v(24) = \frac{1}{EI}\left[-\frac{97}{6}\langle 24-1.5\rangle^3 + \frac{116.4}{6}\langle 24-5.25\rangle^3 - \frac{19.4}{6}\langle 24-24\rangle^3 + 227.344\langle 24\rangle^1 - 284.18\right]$$

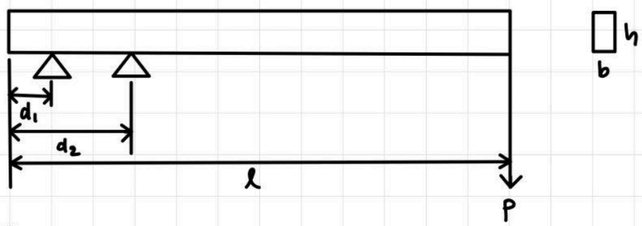
$E = 400$ ksi $\quad I = 0.33\ in^4$

$v(24) = -0.383$ in

$\therefore$ The maximum deflection at the end of the beam is 0.383 in downwards.

JOHNS HOPKINS UNIVERSITY DEPT OF MECHANICAL ENGINEERING 530.216 MBD LAB	CALCULATION Possibility of Yielding for Boom		PAGE 1
	TEAM Sky Corp Inc.	TEAM MEMBERS Lan, Ponce, Pollard	OF 1

Note: Follow-up calculation from Maximum Bending Stress



Knowns:

$P = 19.4$ lb

$b = 0.5$ in

$h = 2$ in

$d_1 = 1.5$ in

$d_2 = 5.25$ in

$l = 24$ in

$I = 0.33$ in^4

$\sigma_{max} = 1091.25$ psi

$E = 400$ ksi (Young's Modulus of an Acrylic Bar, source: MatWeb)

$S_y = 9400$ psi (Yield Strength of an Acrylic Bar, source: MatWeb)

Find: Possibility of Yielding

Assumptions:

- the beam has a uniform cross section & weight
- the brackets that attach the boom to the mounting plate can be treated as simple supports

Design Factor = 1.94 (Note: the design factor was already used in the calculation for maximum stress)

Significant Equations:

$(n.d.)(\sigma_{max}) \leq S_y$

Solution:

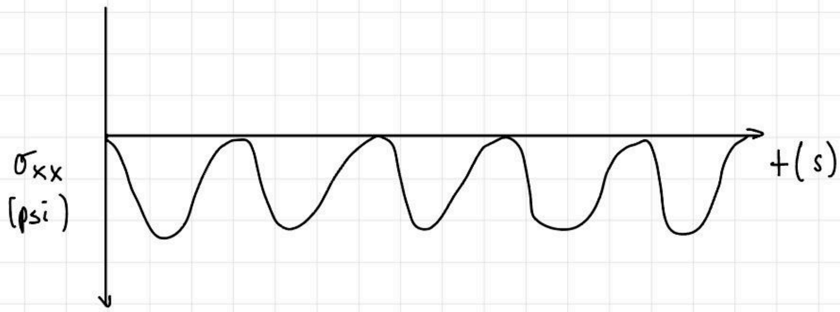
$\sigma_{max} \leq S_y$

$1091.25 \text{ psi} \leq 9400 \text{ psi}$ ✓

∴ The boom will <u>NOT</u> yield under the mechanical load applied.

JOHNS HOPKINS UNIVERSITY DEPT OF MECHANICAL ENGINEERING 530.216 MBD LAB	CALCULATION Beam Fatigue Calculation		PAGE 1 OF 1
	TEAM SkyCorp. Inc	TEAM MEMBERS Lan, Ponce, Pollard	

Non- fully reversed load



ASSUME - Light beam

- Use Goodman criteria (brittle material)

$$\left.\begin{array}{l}\sigma_{xx\,MAX} = 0 \quad \text{lbf} \\ \sigma_{xx\,MIN} = -1091.25 \quad \text{lbf}\end{array}\right\} \quad \begin{array}{l}\sigma_A = 545.63 \quad \text{lbf} \\ \sigma_m = -545.63 \quad \text{lbf}\end{array}$$

$S_{ut} = 14200$ psi (MatWeb)

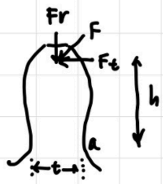
$K_f \approx 1.5$

$S_e \approx 15$ MPa $\cdot\sqrt{m} = 2175$ psi $\sqrt{i}$ (Shigley)

$$K_f \frac{\sigma_a}{S_e} + K_f \frac{\sigma_m}{S_{ut}} \leq \frac{1}{n.d.}$$

$$1.5 \cdot \frac{545.63}{2175} + 1.5 \cdot \frac{545}{14200} \leq \frac{1}{n.d} \rightarrow n.d = 3.1$$

JOHNS HOPKINS UNIVERSITY DEPT OF MECHANICAL ENGINEERING 530.216 MBD LAB	CALCULATION Gear Tooth Bending Stress Analysis		PAGE 1
	TEAM Sky Corp Inc.	TEAM MEMBERS Lan, Ponce, Pollard	OF 1



Find: the maximum bending stress at the root of a gear tooth

Significant Equations: $\sigma_{bending} = \frac{Mc}{I} = \frac{6F^t h}{bt^2} = \dots = \frac{F^t p}{by}$

(b = face width, y = "Lewis Form Factor", p = diametral pitch)

Knowns:

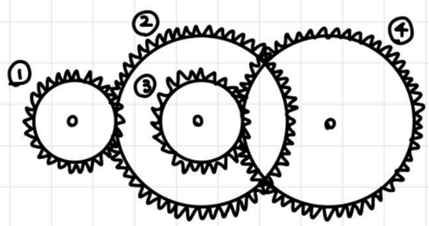
Large Gear	Small Gear
# teeth = 40	# teeth = 20
p = 10 teeth/in	p = 10 teeth/in
b = 0.25 in	b = 0.25 in
y = 0.283	y = 0.336

The Lewis factor (y) for each gear comes from a look-up table via EngineersEdge for a 14.5° pressure angle

Assumptions:

- the full load is applied at the tip of a single tooth
- the radial force component is negligible
- the load is uniformly distributed across the tooth surface
- the friction forces due to sliding are negligible
- the stress concentration at the fillet is negligible

Sketch of Gear Train:



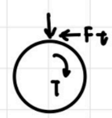
Solution:

Note: the torque is greatest at gear 1

Torque: 0.375 lb·in

F_t = 0.375 lb

$$\sigma_{b_1} = \frac{(0.375\text{ lb})(10/\text{in})}{(0.25\text{ in})(0.285)} = 52.63\text{ psi}$$



$T = \frac{d}{2} F^t$

$F^t = \frac{2T}{d}$

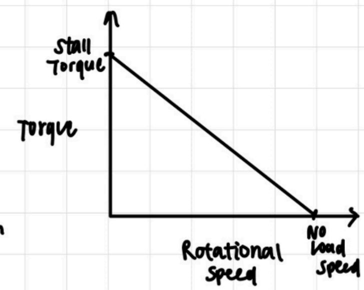
∴ The maximum bending stress at the root of a gear tooth is 52.63 psi.

JOHNS HOPKINS UNIVERSITY DEPT OF MECHANICAL ENGINEERING 530.216 MBD LAB	CALCULATION Motor Torque/RPM Required to Lift the Weight		PAGE 1
	TEAM Sky Corp Inc.	TEAM MEMBERS Lan, Ponce, Pollard	OF 1

Find: Motor Torque/RPM Required to Lift the Weight

Knowns:

- Stall torque of the gearmotor is rated at 12 oz-in at 6V input
- max speed of the gearmotor is 140 RPM
- $W = 10\,lb$
- $r = 0.125$ in (radius of rod)
- Gear Ratio = 4



Assumptions:

- there is no loss of energy in the transmission system

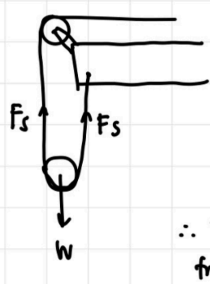
Design Factor = 1.94

Significant Equations:

$T = F \times r$

Solution:

The ideal amount of torque/speed the motor should be about ½ of the stall torque and maximum speed to optimize output.



$F_s = \frac{1}{2}(W \times n.d.) = 9.7\ lbf$

$F_r = \frac{T_{in}}{r} = \frac{T_{in}}{0.125\ in} = 32\,T_{in}$

$32\,T_{in} = 9.7$

$T_{in} = 0.303\ lb\cdot in \times \frac{16\,oz}{1\,lb} = \boxed{4.85\ oz\cdot in}$

$\frac{4.85\ oz\cdot in}{12\ oz\cdot in} = \frac{X\ RPM}{140\ RPM} \rightarrow \boxed{56.58\ RPM}$

$\frac{56.58\ RPM}{GR} = \frac{56.58}{4} = 14.1458\ RPM$

$14.1458\ RPM \times \frac{2\pi\ rad}{1\ rev} \times 0.125'' \times \frac{1\ min}{60\ sec} = 0.185\ in/s \rightarrow \boxed{1\ in\ in\ 5.4\ sec}$

∴ The input torque required from the motor is 4.85 oz·in and the corresponding RPM is 56.58 RPM (14.1458 RPM for a 4:1 gear ratio). The predicted time to lift the 10-lb weight is 5.4 seconds.

Material Properties Sources:

References

McMaster-Carr. (n.d.). McMaster-Carr. https://www.mcmaster.com/1227T439/

Overview of materials for acrylic, cast. (n.d.). Online Materials Information Resource - MatWeb. https://www.matweb.com/search/datasheet.aspx?bassnum=O1303

VI. Cost Analysis

Name	Dim	Qty	Cost	Value Used in Design
Beam	2"x.5"x2'	1	10.02	9.52
Pulley		1	3.03	3.03
Angle Aluminum	1.5"x1.5"x1'	1	5.11	2.56
Aluminum Rod	1/4"x 1'	1	2.53	1.37
Nylon Rod	1/2"x1'	1	6.72	0.55
Acrylic Sheet	1/8"x1'x1'	1	7.71	4.79
Shaft Collars	1/4" ID 1/2" OD	2	1.49	2.98
			TOTAL	USED TOTAL
			36.61	24.80

Individual Cost Assessment

The measuring tape function on SolidWorks was used to measure the length and width of the flat surfaces or the length of the long cylindrical machine components. These values were used to calculate the surface area or length of the total material used. The fraction of the dimension of the material used versus the dimension of the entire material was multiplied by the initial total price to find the new price of the material used.

VII. Performance Evaluation Results

The SkyCorp. crane lifted the ten-pound weight one inch in 11.06 seconds. The expected amount of boom end deflection was 0.383 inches while the actual amount of boom end deflection was 0.2 inches. However, it can be noted that the actual boom end deflection is not extremely precise, as the value was read from a gauge and not a digital device. Additionally, the calculated boom deflection was based on a weight of 19.4 lb (the actual load multiplied by the design factor). Therefore, the actual boom deflection was lower due not to any error in analysis, but due to the design factor providing an adequate margin for error.

During the crane performance evaluation, the Sky Corp. crane worked as expected since it was able to meet all the criteria for the design evaluation. In the week prior to the evaluation date, the crane was tested multiple times to ensure workability. However, this could have worn out the gearmotor, as it seemed like the crane was slower in lifting the weight 1 inch off the ground than when it was tested in the past. Furthermore, right before the official time trial, the wire was supposed to be pulled taut by pulling the weight just slightly off the ground. But when this operation was performed, the weight was pulled fully off the ground and was lifted over an inch in the air. It should be noted that this operation could have contributed to “tiring out” the motor before the official time trial and thus resulted in a slightly slower time to lift the weight than expected. Another unexpected aspect was that the retraction of the wire slowed down after a few seconds. This contributed to a delay in lifting the weight with the motor running at the desired speed. The gearmotor could have slowed down because it may have needed to increase the torque due to the heavy load. Additionally, the predicted time to lift the weight was 5.4 seconds, while the actual time was more than twice that number. This could be due to the fact that the gearmotor was passed down from years past and is no longer able to provide the speed and torque it was originally rated to output.

Overall, the crane performed successfully during the performance evaluation since it exceeded all the requirements, including being able to lift the 10-lb weight 1 inch from the ground in less than 2 minutes and deflecting less than 0.5 inches. All other aspects worked as expected in retracting the wire to pull up the weight, in particular the smooth interfacing of the gears in the gear train and the mechanical advantage of the pulley.