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REDESIGN OF 2010 HONDA CIVIC SEDAN TRANSMISSION TO PRODUCE TORQUE COMPARABLE TO AN AVERAGE PICKUP TRUCK

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ABSTRACT

The transmission of the 2010 Honda Civic Sedan was redesigned to produce torque of 400 ft-lb. The results were comparable to that of an average pickup truck. The final cost for the transmission redesign is $3,551.79. The final dimension of the transmission is 8.5714” × 8.5714” × 36.25”.

Figure 1. *2010 Honda Civic Sedan*

NOMENCLATURE

C	Load, kN
C_#_	Counter shaft gear, following number indicates its gear set (i.e. C1 = counter shaft gear, 1st gear)
d	Shaft diameter, in.
$\frac{dy}{dx}$	Shaft slope, radians
F	Radial force, lb
FOS	Factor of safety
G	Face width, in.
HP	Horsepower, hp
K/k	Factor
L	Life, cycles
N	Number of teeth, teeth
n	Speed, rpm
O_#_	Output shaft gear, following number indicates its gear set (i.e. O1 = output shaft gear, 1st gear)
R	Reliability, percentage
S	Strength, psi
T	Torque, ft-lb
V	Pitch line velocity, ft/min
W	Transmitted load, lb
x	Weibull variable
Y	Lewis Form Factor **[1]**
y	Deflection, in.

Greek

σ	Stress, psi
ϕ	Pressure angle, degrees
θ	Weibull characteristic parameter

Subscripts

0	Minimum for Weibull variable
10	Rated, for life and load
a	Amplitude for σ, surface factor denotation for k
allow	Allowable
b	Size factor denotation for k
c	Loading factor denotation for k
counter,1	Counter shaft gear paired with input shaft gear
counter,2	Counter shaft gear paired with output shaft gear
D	Desired
e	Endurance
f	Fatigue
input	Input
m	Mean
old	Previous iteration value
output	Output
T	Tangential
ut	Ultimate
V	Velocity factor denotation for K
y	Yield

INTRODUCTION

The 2010 Honda Civic Sedan is a capable vehicle but is at a disadvantage in a great portion of the American market. It lacks any significant torque capabilities that would allow for it to compete against other market options such as a pickup truck. A redesign of the transmission would improve its appeal to this new audience while retaining the other features that would set this vehicle apart.

Figure 2. *Modified 2010 Honda Civic Sedan Transmission, Isometric View*

BACKGROUND

A common problem on the road is having a vehicle have trouble hauling significant weight and having this bottleneck its speed. An expensive solution is to replace the vehicle with one designed from ground up to haul. A potentially cheaper solution however is to modify a transmission in an existing vehicle. While creating the transmission one-time can be an expensive endeavor, future streamlining would reduce the cost drastically and the vehicle would continue to hold many features that attracted many of its original customers all the while attracting new customers.

OBJECTIVE

The objective of this design is to redesign the 2010 Honda Civic Sedan transmission to produce a 400 ft-lb torque which would enable it to move modest loads such as a 2,000 lb travel trailer.

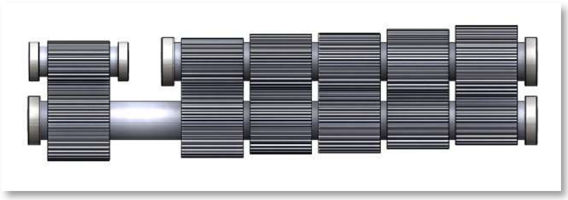

Figure 3. *Modified 2010 Honda Civic Sedan Transmission, Side View*

THEORY

Transmission gearing modifies the RPM of the engine to produce different output RPMs and their counterpart output torques. Each vehicle gear functions by having the input shaft connect to the counter shaft via a gear set, and then having the counter shaft connect to the output shaft via a second gear set. This gear train allows for accurate assessment of RPM reduction. [2]

$$n_{output} = n_{input} \cdot \frac{N_{input}}{N_{counter,1}} \cdot \frac{N_{counter,2}}{N_{output}} \quad (1)$$

The subsequent gear ratios chosen should provide reasonably smooth transitions between different torques.

When considering perfect spur gear torque transmission, the force transmitted is identified in tangential load. The tangential load can then be used to find other vital parameters such as stress and yield factor of safety.

$$W_T = T \cdot \frac{2}{d} \quad (2)$$

$$\sigma = \frac{W_T P}{GY} \quad (3)$$

$$V = 33000 \frac{HP}{W_T} \quad (4)$$

$$K_V = \frac{1200 + V}{1200} \quad (5)$$

$$FOS = K_V \frac{S_y}{\sigma} \quad (6)$$

Fatigue calculations can also be performed as provided as per instructions of Shigley's Mechanical Engineering Design [1].

$$S_e = S_e' k_a k_b k_c \quad (7)$$

$$FOS_f = \frac{1}{\frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_{ut}}} \quad (8)$$

The most notable forces at play are those committed by the gears, so tangential force and be converted to radial force given the gears' pressure angles. The calculation of the radial force solves the large unknown that is necessary for summation of forces. The summation of forces produces results for reaction forces which are produced by bearings.

$$F = W_t \sin \phi \quad (9)$$

$$C_{10} = a_f F \left(\frac{L_D / L_{10}}{x_0 + (\theta - x_0)(1 - R_D)^{1/b}} \right)^{1/a} \quad (10)$$

If the bearing rated load needs to be rounded, it should be rounded up to maintain conservative calculations. [3]

Lastly the shafts that the bearings hold must have shoulders for the bearings to retain their position. These shoulders introduce sharp fillets and act as stress concentrations. However, due to these fillets existing very close to the ends of the shafts they do not contribute any stress concentration factors. With no steps holding the gears in place (i.e. hypothetical tightening mechanisms), the largest moment is simply calculated for each shaft. The DE-Goodman equation can be used to find the factor of safety of the shaft.

$$\frac{1}{FOS_f} = \frac{16}{\pi d^3} \left\{ \frac{1}{S_e} \sqrt{4\left(K_f M_a\right)^2} + \frac{1}{S_{ut}} \sqrt{3\left(K_{fs} T_m\right)^2} \right\} \quad (11)$$

From there, maximum deflection and slope of the shafts can be found by referring to Shigley's

Mechanical Engineering Design [1]. Should the deflection and/or slope fall out of specification, they can be adjusted as follows:

$$d_{new} = d_{old} \left| \frac{n_d y_{old}}{y_{allow}} \right|^{1/4} \tag{11}$$

$$d_{new} = d_{old} \left| \frac{n_d \frac{dy}{dx}_{old}}{\frac{dy}{dx}_{allow}} \right|^{1/4} \tag{12}$$

DATA

Gear	Desired Ratio	Actual Ratio
1st	3	3
2nd	2.25	2.285714286
3rd	1.8	1.75
4th	1.65	1.636363636
5th	1.25	1.243243243

Table 1. *Initial Gear Ratios Chosen for Desired Torque*

Gears	Yield n	Goodman nf	Tangential Load	
Input	2.20143334	1.02129213	18529.056	lbf
Counter Input	4.54058278	2.106473711	9264.528	lbf
Counter 1st	2.672817414	1.239977309	15440.88	lbf
Output 1st	4.083556751	1.894449537	10293.92	lbf
Counter 2nd	3.161186629	1.466542259	13235.04	lbf
Output 2nd	3.619093415	1.678975036	11580.66	lbf
Counter 3rd	3.619093415	1.678975036	11580.66	lbf
Output 3rd	3.161186629	1.466542259	13235.04	lbf
Counter 4th	3.733753629	1.732168368	11229.73091	lbf
Output 4th	3.042549665	1.411504028	13725.22667	lbf
Counter 5th	4.201354783	1.94909857	10015.70595	lbf
Output 5th	2.559935384	1.187608915	16112.22261	lbf

Table 2. *Gears' Factor of Safety and Tangential Load*

Bearing	C10	C10 Rounded Up
Input (x2)	57.5830 kN	58.5 kN
Output (x2)	87.8625 kN	93 kN
Counter (x2)	92.5299 kN	93 kN

Table 3. *Bearing C10 Values*

Shaft	Diameter	Addition	FOS
Input	1.1811 in.	1 in.	1.0950
Output	1.5748 in.	0.9 in.	1.0594
Counter	1.5748 in.	0.3 in.	1.0129

Table 4. *Shafts' Factor of Safety (DE Goodman)*

ANALYSIS

All gears and shafts were made of AISI 4140 Q&T (400 F) steel. Material properties were either derived from Shigley's Mechanical Engineering Design [1] or MatWeb [5].

For this design, spur gears were initially chosen given their cheaper manufacturing costs and lack of thrust (radial forces present only). After multiple iterations of design, the spur gears demonstrated sufficient performance and were decided to be kept solely on their cheaper manufacturing costs.

Figure 4. *Counter Shaft Spur Gear*

Calculations all assumed worst case scenario: the transmission would be constantly receiving 140 HP from the engine [4]. 1st gear experiences RPMs between 1000 and 3000, 2nd between 2250 and 3000, 3rd between 2250 and 3000, 4th between 2250 and 3000, and 5th between 2250 and 6800. The max RPMs were chosen based on recommended RPMs where the driver would shift except for the 5th gear which is the redline RPM as provided by Honda [4].

Raw data (specifications) were initially interpreted by a gear solver written in MATLAB. The MATLAB program solved the required equations simultaneously and produced nominal gear teeth count on each gear to produce the desired gear ratios.

```
function SolveGears()
    P = 4;
    In = 20;
    CIn = 40;
    syms C1 O1 C2 O2 C3 O3 C4 O4 C5 O5

    eq1 = (In/CIn)*(C1/O1) == 1/3;
    eq2 = (In/CIn)*(C2/O2) == 1/2.25;
    eq3 = (In/CIn)*(C3/O3) == 1/1.8;
    eq4 = (In/CIn)*(C4/O4) == 1/1.65;
    eq5 = (In/CIn)*(C5/O5) == 1/1.25;
    eq6 = ((C1/P)+(O1/P)) == ((In/P)+(CIn/P));
    eq7 = ((C2/P)+(O2/P)) == ((In/P)+(CIn/P));
    eq8 = ((C3/P)+(O3/P)) == ((In/P)+(CIn/P));
    eq9 = ((C4/P)+(O4/P)) == ((In/P)+(CIn/P));
    eq10 = ((C5/P)+(O5/P)) == ((In/P)+(CIn/P));

    sol = solve([eq1,eq2,eq3,eq4,eq5,eq6,eq7,eq8,eq9,eq10],[C1 O1 C2 O2 C3 O3 C4 O4 C5 O5]);
    disp(round(sol.C1));
    disp(round(sol.O1));
    disp(round(sol.C2));
    disp(round(sol.O2));
    disp(round(sol.C3));
    disp(round(sol.O3));
    disp(round(sol.C4));
    disp(round(sol.O4));
    disp(round(sol.C5));
    disp(round(sol.O5));
end
```

Figure 5. *MATLAB Program*

With teeth count for the input, counter, and counter/output pair gears, solvers were created in Excel. The solvers were not automated, rather there was an input box and the desired result would be produced.

							Interpolation Chart	
Yield Check							N	40
Diameter	10	in.	Tangential Load	5294.016	lbf		N1	38
Teeth	20	teeth	Sy	77000	psi	[1045 CF Steel]	N2	43
Sigma	13602.30216	psi	Sut	91000	psi		Y1	0.384
Sigma*Kv	23494.38849	psi	Y	0.3892	[Table 14-2]		Y2	0.397
n	3.277378342		V	872.6834222	ft/min		x	0.4
			Kv	1.727236185	[Cut or milled profile]		Y	0.3892
Fatigue Check								
Se'	45500	psi	Addendum	0.25	Dedendum	0.3125	h	0.5625
ka	0.816991984		de	1.212				
kb	0.861226122							
kc	1							
Se	32014.47511	psi						
Sigma m	11747.19424	psi	Sigma a	11747.19424	psi	[Zero based]		
Goodman nf	2.016031949							

Figure 6. *Example Solver (Yield and Fatigue Check for Gears) in Excel*

The only calculations that were not automated by a solver were reaction force calculations and deflection/slope calculations. Those calculations can be found in Appendix A and Appendix B respectively.

Reaction forces on the shaft were calculated based on the worse-case scenario for radial loads. For the input shaft, this was just the radial force of the input gear. For the output shaft, this was just the radial force of the O1 (output, 1st gear) gear. For the counter shaft, the counter gear's radial force and C1's (counter, 1st gear) radial force were considered. The greater of the two reaction forces on a shaft was used to determine a sufficient C10 bearing value. The values were then rounded up to the manufacturer's closest available bearing with the desired value [3]. The values were rounded up to remain conservative.

The initial diameter of the shafts were set to their respective bearing's bore diameter. The diameter was increased as necessary to have a factor of safety over 1. As mentioned above, deflection/slope calculations had to be made. The output shaft and the counter shaft suffered from either deflection or slope that were out of specification, so where reevaluated. This is why the final diameters in the BOM is different from values in Table 4.

RESULTS

Gear	N	d
Input	20 teeth	2.8571 in.
Counter	40 teeth	5.7143 in.
C1	24 teeth	3.4286 in.
O1	36 teeth	5.1429 in.
C2	28 teeth	4.0000 in.
O2	32 teeth	4.5714 in.
C3	32 teeth	4.5174 in.
O3	28 teeth	4.0000 in.
C4	33 teeth	4.7143 in.
O4	27 teeth	3.8571 in.
C5	37 teeth	5.2857 in.
O5	23 teeth	3.2857 in.
All gears	P = 7 teeth/in.	G = 4.5 in.

Table 5. *BOM – Gears*

Bearings	SKF Name [3]	C10	Bore
Input (x2)	NUP 306 ECJ	58.5 kN	30 mm
Output (x2)	NU 308 ECML	93 kN	40 mm
Counter (x2)	NU 308 ECML	93 kN	40 mm

Table 6. *BOM – Bearings*

Shaft	Diameter	Total Length
Input	2.1811 in.	7.248 in.
Output	2.7817 in.	27.4055 in.
Counter	2.7981 in.	37.1555 in.

Table 7. *BOM - Shafts*

Using the provided manufacturer, the estimated cost of all materials is $3,551.79. The cost of the bearings is not provided by SKF so is unknown.

Gear	Volume		Cost		Shaft	W before		W after		W removed		Cost
Input	11.69	in^3	$83.50		Input	7.68	lb	6.56	lb	1.12	lb	$152.56
Counter Input	86.92	in^3	$475.07		Output	47.23	lb	45.11	lb	2.12	lb	$291.66
Counter 1st	13.43	in^3	$92.55		Counter	64.8	lb	62.64	lb	2.16	lb	$329.20
Output 1st	65.41	in^3	$363.11									
Counter 2nd	28.34	in^3	$170.16		Bearing	Name	Cost					
Output 2nd	45.88	in^3	$261.46		Input (2x)	NU 1016 ECM	N/A					
Counter 3rd	45.56	in^3	$259.79		Output (2x)	NU 2212 ECPH	N/A					
Output 3rd	28.67	in^3	$171.88		Counter (2x)	NUP 217 ECJ	N/A					
Counter 4th	50.23	in^3	$284.10									
Output 4th	24.72	in^3	$151.32									
Counter 5th	70.33	in^3	$388.72									
Output 5th	10.39	in^3	$76.73		Total	$3,551.79						

Table 8. *Breakdown of Cost*

DISCUSSION

Due to the entire process not be automated, there was plenty of human error along the way. When mistakes were caught, the numbers were ran again. This resulted in many iterations of attempting to design the transmission.

Because the solver was used for a large amount of the calculations, previous attempts that were not successful were not recorded. While the final design is influenced by the mistakes made along the way via the solver, the solver limits how much of the design history is kept (in this situation, mostly only the final design numbers were presented).

Lastly this situation was tackled as being a worst-case scenario. In actuality, the horsepower will not be at maximum 100% of the time and there will be external factors that cannot be predicted ahead of time such as thrust. Should this product be reevaluated, helical gears may be necessary for quieter action, smoother load transmission, and/or accountability of thrust.

CONCLUSIONS

- The transmission of the 2010 Honda Civic Sedan can be redesigned to produce torque comparable to that of an average pickup truck (400 ft-lb).
- Solvers helped calculations immensely. Further automation is possible and would allow for checking more possible configurations as opposed to being stuck on selecting an initial guess and iterating said guess.
- Solvers however do not record history and a lot of previous iterations (successes and failures alike) are lost.
- These calculations are unnecessarily stringent due to being absolute worst-case scenario. It is unlikely for the engine to perform at maximum horsepower 100% of the time.

ACKNOWLEDGEMENTS

Thank you to Dr. Sha for teaching us MED this semester. Thank you to the MEEG department for providing a great learning environment.

REFERENCES

[1] Budynas R., Nisbett J. “Shigley’s Mechanical Engineering Design Tenth Edition.” McGraw-Hill Education (2015).

[2] “Manual Transmission, How it works?” *YouTube*. https://www.youtube.com/watch?v=wCu9W9xNwtI.

[3] “Cylindrical roller bearings, single row.” *SKF*. http://www.skf.com/group/products/bearings-units-housings/roller-bearings/cylindrical-roller-bearings/single-row-cylindrical-roller-bearings/single-row/index.html.

[4] “Model Information.” *American Honda Motor Co., Inc.* http://owners.honda.com/vehicles/information/2010/Civic-Sedan/specs#mid^FA1E2AEW.

[5] “AISI 4140 Steel, Oil Quenched, 205°C (400°F) temper, 25 mm (1 in.) round.” *MatWeb, LLC.* http://matweb.com/search/DataSheet.aspx?MatGUID=757684bcff76455eaf94b20102617988&ckck=1.

APPENDIX A

Reaction Calculations

Input Shaft

Tangential Load: 18529.06 lbf
Radial (20-degree PA): 6337.310389 lbf

$$+\uparrow \sum F_y = 0: R_A + R_B = 6337.310389\ lbf$$
$$+\circlearrowleft \sum {M_z}^A = 0: -(4\ in.)(3621.319538\ lbf) + (8\ in.)R_B = 0$$
$$R_B = 3168.655194\ lbf$$
$$R_A = 3168.655194\ lbf$$

Output Shaft

Tangential Load: 16112.22261 lbf
Radial (20-degree PA): 5510.704686 lbf

$$+\uparrow \sum F_y = 0: R_A + R_B = 5510.704686\ lbf$$
$$+\circlearrowleft \sum {M_z}^A = 0: -(3.25\ in.)(5510.704686\ lbf) + \big(2(3.25\ in.) + 4(5\ in.)\big)R_B = 0$$
$$R_B = 675.8411408\ lbf$$
$$R_A = 4834.863546\ lbf$$

Counter Shaft

Tangential Load 1: 9264.53 lbf
Radial Load 1: 3168.655194 lbf

Tangential Load 2: 15440.88 lbf
Radial Load 2: 5281.091991 lbf

$$+\uparrow \sum F_y = 0: R_A + R_B = 3168.655194\ lbf + 5281.091991\ lbf$$
$$+\circlearrowleft \sum {M_z}^A = 0: (3.25\ in.)(3168.655194\ lbf) + (\ 4(3.25\ in.) + 4(5\ in.))(5281.091991\ lbf) - (36.25\ in.)R_B = 0$$
$$R_B = 5091.701105\ lbf$$
$$R_A = 3358.04608\ lbf$$

APPENDIX B

Deflection and Slope Calculations

Slope: Cylindrical roller, 0.0008 – 0.0012 rad
Deflection: Spur gear P < 10, 0.010 in.

Input Shaft

$$y_{max} = \frac{Fl^3}{48EI} = \frac{(6337.310389\ lbf)(6.5\ in.)^3}{48(29700000\ psi)\left(\frac{\pi(2.1811\ in.)^4}{64}\right)} = 0.0010991445\ in.$$

$$\frac{dy}{dx_{max}} = 0$$

Output Shaft

$$\begin{aligned} y_{max} &= \frac{Fab}{6EIl}(a^2 + b^2 - l^2) \\ &= \frac{(5510.704686\ lbf)(3.25\ in.)(23.25\ in.)}{6(29700000\ psi)\left(\frac{\pi(2.4748\ in.)^4}{64}\right)(26.5\ in.)}((3.25\ in.)^2 + (23.25\ in.)^2 - (26.5\ in.)^2) \\ &= -0.0058304302\ in. \end{aligned}$$

$$\begin{aligned} \frac{dy}{dx_{max}} &= \frac{Fb}{6EIl}(3x^2 - l^2 + b^2) = \frac{Fb}{6EIl}(3a^2 - l^2 + b^2) \\ &= \frac{(5510.704686\ lbf)(23.25\ in.)}{6(29700000\ psi)\left(\frac{\pi(2.4748\ in.)^4}{64}\right)(26.5\ in.)}(3(3.25\ in.)^2 - (26.5\ in.)^2 + (23.25\ in.)^2) \\ &= -0.0019155326\ rad \end{aligned}$$

$$d_{new} = d_{old}\left|\frac{n_d \frac{dy}{dx_{old}}}{\frac{dy}{dx_{allow}}}\right|^{1/4} = 2.4748\ in.\left|\frac{(1)(0.0019155326\ rad)}{0.0012\ rad}\right|^{1/4} = 2.781744146\ in.$$

Counter Shaft (Superposition)

$$\begin{aligned} y_{max,1} &= \frac{Fab}{6EIl}(a^2 + b^2 - l^2) \\ &= \frac{(3168.655194\ lbf)(3.25\ in.)(33\ in.)}{6(29700000\ psi)\left(\frac{\pi(1.8748\ in.)^4}{64}\right)(36.25\ in.)}((3.25\ in.)^2 + (33\ in.)^2 - (36.25\ in.)^2) \\ &= -0.0186077641\ in. \end{aligned}$$

$$y_{max,2} = \frac{Fab}{6EIl}(a^2 + b^2 - l^2)$$
$$= \frac{(5281.091991\ lbf)(33\ in.)(3.25\ in.)}{6(29700000\ psi)\left(\frac{\pi(1.8748\ in.)^4}{64}\right)(36.25\ in.)}((33\ in.)^2 + (3.25\ in.)^2 - (36.25\ in.)^2)$$
$$= -0.0310129402\ in.$$

$$y_{max} = y_{max,1} + y_{max,2} = -0.0496207043\ in.$$

$$d_{new} = d_{old}\left|\frac{n_d y_{old}}{y_{allow}}\right|^{1/4} = 1.8748\ in.\left|\frac{(1)(0.0496207043\ in.)}{0.01\ in.}\right|^{1/4} = 2.798147964\ in.$$

$$\frac{dy}{dx}_{max,1} = \frac{Fb}{6EIl}(3a^2 - l^2 + b^2)$$
$$= \frac{(3168.655194\ lbf)(33\ in.)}{6(29700000\ psi)\left(\frac{\pi(2.798147964\ in.)^4}{64}\right)(36.25\ in.)}(3(3.25\ in.)^2 - (36.25\ in.)^2$$
$$+ (33\ in.)^2) = -0.0010402098\ rad$$
$$\frac{dy}{dx}_{max,2} = \frac{Fb}{6EIl}(3a^2 - l^2 + b^2)$$
$$= \frac{(5281.091991\ lbf)(3.25\ in.)}{6(29700000\ psi)\left(\frac{\pi(2.798147964\ in.)^4}{64}\right)(36.25\ in.)}(3(33\ in.)^2 - (36.25\ in.)^2$$
$$+ (3.25\ in.)^2) = 0.001733683\ rad$$
$$\frac{dy}{dx}_{max} = \frac{dy}{dx}_{max,1} + \frac{dy}{dx}_{max,2} = 0.0006934732\ rad$$

APPENDIX C

V Diagrams, M Under Curve

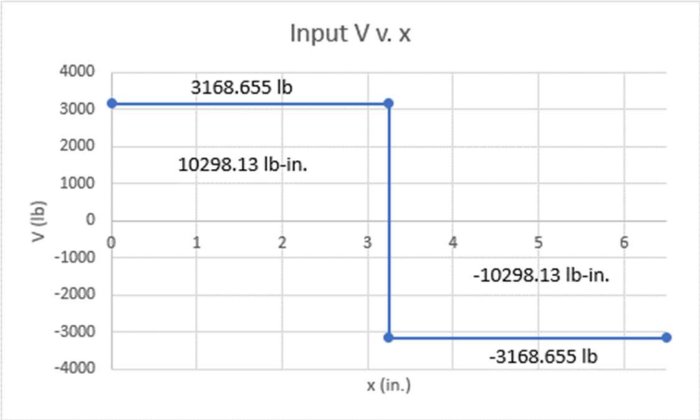

Figure 7. *Input Shaft V Values, M Under Curve*

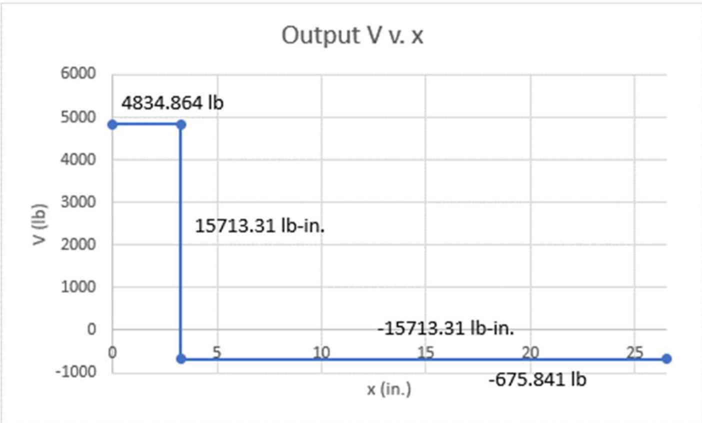

Figure 8. *Output Shaft V Values, M Under Curve*

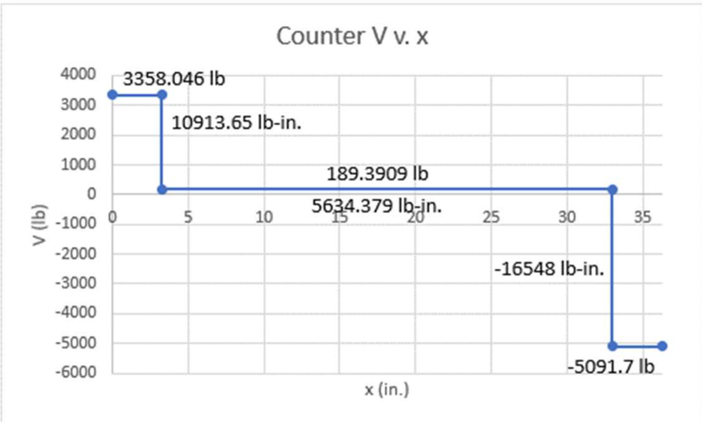

Figure 9. *Counter Shaft V Values, M Under Curve*