UNIVERSITY OF NOTRE DAME

SENIOR DESIGN PROJECT

AME 40463

Gearbox Trade Study

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Team 12

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Executive Summary

The purpose of this Trade Study is to choose a gearbox to incorporate with an autonomous warehouse vehicle. The vehicle is meant to move large inventory in a warehouse as quickly as possible without leaving skid marks on the factory floor. In particular, the vehicle is limited to a maximum weight of 21 lb, and it must move a 29 lb payload a distance of 20 ft. The vehicle-payload combination must stop before reaching 25 ft. The setup is detailed in Figure 1.

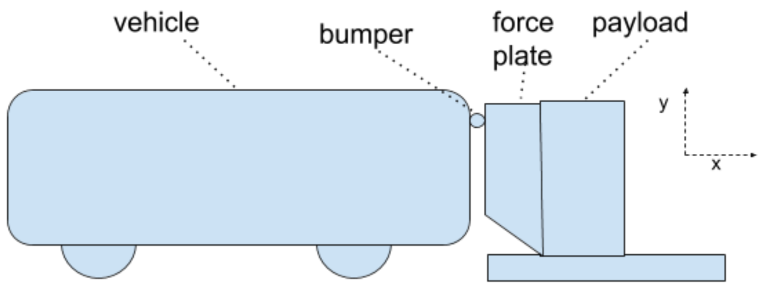

Figure 1: Vehicle-payload setup in the warehouse.

The significance of the overall project is that a successfully-designed vehicle could save time and labor on the warehouse floor, resulting in greater output and more efficient spending. The company could dedicate its factory-floor workers to more intellectually or socially intensive activities that could not be performed by a robot. The significance of choosing the gearbox to match with the two Banebots RS775 24V motors is that the correct choice will ensure that the vehicle will travel as fast as physically possible given the floor-tire tractive conditions. To accomplish the task quickly, it is desirable to provide the maximum tractive force at the wheels, which occurs at the slip condition. The company prohibits the formation of any skid marks, so the engineering team designs for the vehicle wheels to operate at 95% of the maximum tractive force.

The team collected friction data for the wheels and payload and focused calculations around the maximum wheel static coefficient of friction found $\mu_{ws} = 0.68$ and the maximum payload kinetic coefficient of friction found $\mu_{pk} = 0.40$. A force analysis with these friction values determined a maximum tractive force of 12.9 lb. Dividing this tractive force equally between the two rear drive wheels and using 95% of this maximum gearmotor torque revealed the desired operating torque of 0.51 ft·lb. Constant acceleration equations revealed the maximum acceleration, maximum deceleration, switching position, maximum velocity, and the time to complete the task.

The gearmotor assembly needed to provide enough torque to exceed the slip condition torque, but also needed to have enough angular speed headroom so that it would not reach its maximum speed before having to decelerate. Additionally, a set of two gearboxes could not exceed $150. When using a wheel-surface static friction coefficient of $\mu = 0.68$, no gearbox, the Banebots P61 3:1 gearbox, the Banebots P61 4:1 gearbox, and the the Banebots P61 11:1 gearbox all satisfied the constraints. When the wheel-surface static friction coefficient increased to $\mu = 0.90$, only the Banebots P61 3:1 and 4:1 gearboxes remained as viable options. The Banebots P61 4:1 gearbox was chosen because it met the constraints and it would enable the motor to operate at a higher RPM and lower torque, lengthening its lifetime and reducing that amount of current pull from the batteries. The 4:1 gearbox is also cheaper than the 3:1 gearbox, with a set of two costing $94.20.

Engineering Analysis

The vehicle under analysis is an autonomous machine with two drive wheels meant to move large loads of inventory in a warehouse. Each drive wheel is a 2 inch diameter, shore 60 wheel, powered by a Banebots RS775 24VDC motor. The robot vehicle weighs 21 pounds and is capable of moving a 29 pound load in a straight line a distance of 20 feet. The vehicle must contact a force plate on the load with a rubber bumper at a height of 3 in. The setup is detailed in Figure 2. The warehouse floor

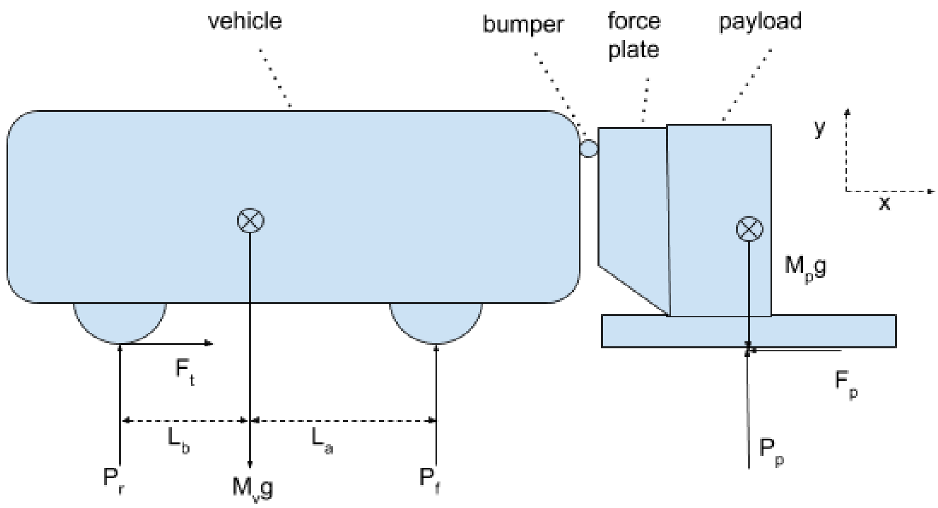

Figure 2: Free-body diagram of the vehicle and payload.

is unknown, so the coefficient of friction between the wheels and the floor must be estimated. The Trade Study design variables, state variables, and constraints, as well as the previously submitted Trade Study Proposals, can be found in Section 1 of the Appendix. The following steps were taken to select the gearbox of the drive system of the autonomous warehouse machine, used as part of a Senior Design Project. A test vehicle previously designed for the application was used as part of the design process for the new vehicle.

Step 1: Determine reasonable coefficient of friction values for the wheels and the payload. The test vehicle was placed on a level surface such that its drive wheels were on the force plate. The load on the wheels were recorded. The vehicle was pushed on the classroom tile, carpet, and marble stairway in Stinson-Remick Hall, and the force required to move the vehicle was recorded. The pushing experiment was repeated with the payload. The force values were then used to determine the static friction coefficient of the wheels, the static friction coefficient of the payload, and the kinetic friction coefficient of the payload, using the relationship

$$F = P\mu, \tag{1}$$

where F is the measured force value, P is the upward normal force acting against either the drive wheels or the payload, and μ is the coefficient of friction. The data and calculations detailed in Section 2 of the Appendix produced the coefficient of friction values for each surface, displayed in Table 1.

Table 1: Coefficients of friction for the wheel and payload on different surfaces.

surface	wheel static	payload static	payload kinetic
classroom tile	0.41	0.33	0.26
carpet	0.58	0.38	0.35
marble stairwell	0.68	0.51	0.40

Step 2: Find the maximum tractive force of the drive wheels. Assume that the longitudinal distance $L = L_a + L_b$ between the front and back wheels of the vehicle under design is the same as that of the test vehicle, that is 6.25 in. Assume that the center of gravity is 10% of this distance away from the back wheel. Assume that the coefficient of friction between the wheel and the floor is the maximum coefficient of friction found in Step 1, $\mu_{ws} = 0.68$. The maximum tractive force of the drive wheels,

$$F_{tmax} = M_v \cdot g \cdot \frac{L_a}{L} \cdot \mu_{ws} \tag{2}$$

is found to be F_{tmax}=12.9 lb. The derivation and calculation are detailed in Section 3 of the Appendix.

Step 3: Find the maximum required operating torque of the gearmotor at each drive wheel. Figure 3 details a free-body diagram of one drive wheel. A moment analysis, detailed in Section 4 of

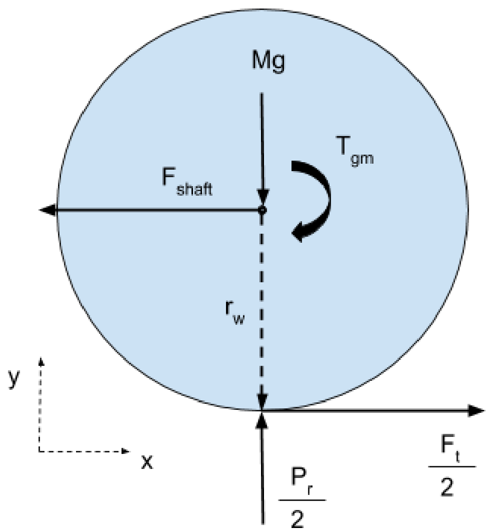

Figure 3: Free-body diagram of one drive wheel.

the Appendix, reveals the maximum torque of the gearmotor, T_{gmax}. The team desires for the wheels to operate at 95% of this maximum slip condition torque, so the maximum operating torque of the gearmotor,

$$T_{gm} = 0.95 \cdot T_{gmax} = 0.95 \cdot \frac{F_{tmax}}{2} \cdot r_w \tag{3}$$

is determined to be $T_{gm} = 0.51$ ft $\cdot$ lb.

Step 4: Calculate the maximum acceleration, the maximum deceleration, the switching position, the maximum velocity, and the time to complete the task of the vehicle-payload combination. Assume that both gearmotors operate at T_{gm} for as long as possible until required to reverse the torque to achieve a maximum deceleration torque of T_{gm} in the opposite direction. The maximum acceleration $a_{max} = 0.41\frac{ft}{s^2}$. The maximum deceleration $d_{max} = -12.9\frac{ft}{s^2}$. The switching position is the distance from the start at which point the vehicle transitions from maximum acceleration to maximum deceleration. Assuming that it is desirable to have the vehicle-payload combination stop at a position of 22.5 ft, which is halfway between the $20 \leq x \leq 25$ ft acceptable range, the switching position $x_{switch} = 20$ ft. The maximum velocity $v_{max} = 4.04\frac{ft}{s}$. This corresponds to an angular wheel rotation of 463 RPM. The time for the vehicle to travel 20 ft is $t = 9.91$ s. The calculations are detailed in Section 5 of the Appendix.

Step 5: Plot the torque-speed curve and the maximum operating torque of the motor given the gear ratio R. Table 2 details RS775 24V motor data from the Banebots website [2] that is necessary for further calculations.

Table 2: Banebots RS775 24V motor specifications.

specification	variable	value from website	converted value
no load speed	ω_0	12600 RPM	–
no load current	I_0	0.85 A	–
stall current	I_{stall}	59 A	–
stall torque	T_{stall}	119.43 oz·in	0.6220 ft·lb

Results

Due to *Design Variable 1*, which requires the transmission to be compatible with the Banebots RS775 24V motor, only Banebots gearboxes that ensured compatibility with the RS-700 motors were considered. Plotting torque (ft·lb) vs. angular speed (RPM) for the gearmotor enabled for an analysis of *State Variable 1*, which required the stall torque of the gearmotor to be higher than the maximum operating gearmotor torque $T_{gm} = 0.51$ ft·lb. The torque vs. angular speed plot also enabled for an analysis of *State Variable 2*, which required the maximum angular speed at T_{gm} to be greater than the maximum angular velocity of the wheel. The Banebots website, which listed the price of each gearbox, enabled for an analysis of *State Variable 3*.

The proposed gearmotors included the RS775 24V motor linked with no gearbox, with the Banebots 3:1 gearbox, the Banebots 4:1 gearbox, and the Banebots 11:1 gearbox. The torque-speed curves of the four gearmotor combinations are detailed in Figure 4 and Figure 5.

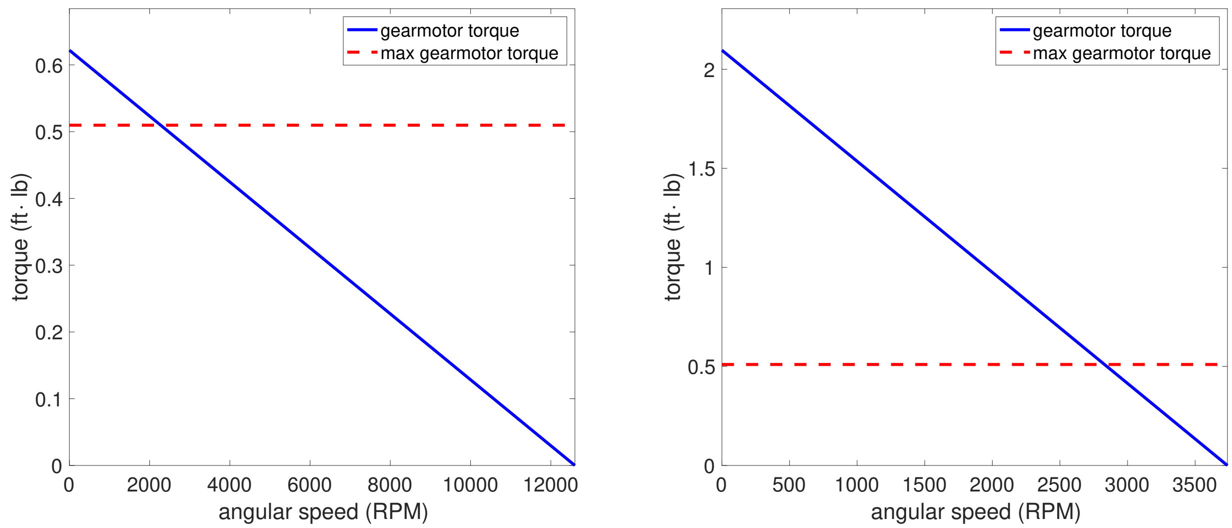

Figure 4: Plot of gearmotor torque-speed curve of the Banebots RS775 24V motor with no gearbox (left) and with Banebots P61 Standard, 3:1 gearbox (right).

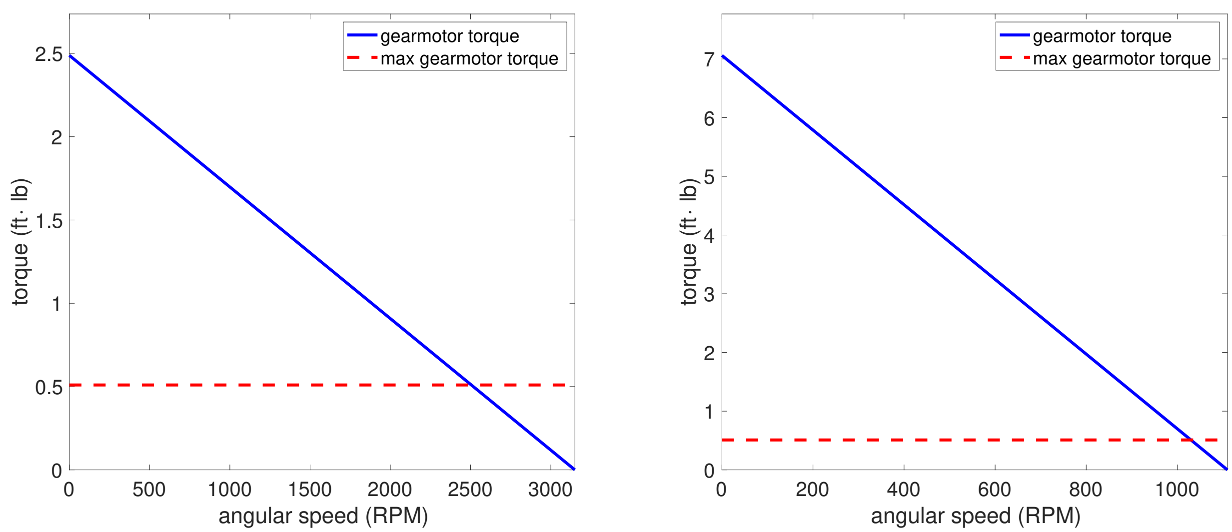

Figure 5: Plot of gearmotor torque-speed curve of the Banebots RS775 24V motor with Banebots P61 Standard, 4:1 gearbox (left) and with Banebots P61 Standard, 11:1 gearbox (right).

All proposed gearmotors met all of the state variables given the current coefficient of friction predictions. The time to complete the task, computed in step 4, is much higher than expected.

Therefore, it is likely that the current wheel-surface coefficient of static friction is a low estimate. If the wheel static friction is changed from $\mu_{ws} = 0.68$ to a much higher value of $\mu_{ws} = 0.90$, the maximum operating gearmotor torque increases to $T_{gm} = 0.67$ ft·lb, and the maximum angular velocity of the gearmotor at T_{gm} increases to 1189 RPM.

In such a case, the motor would not be able to provide enough torque on its own to exceed the torque of the gearmotor, eliminating the possibility of not having a gearbox. With a higher T_{gm}, the maximum angular velocity at T_{gm} will also decrease as the red-dashed line climbs up the blue gearmotor torque line in Figure 4 and Figure 5. The maximum angular speed of the 11:1 gearbox will further decrease, failing to provide 1189 RPM at T_{gm}. Both the 3:1 and 4:1 gearboxes still meet all of the state variables. Table 3 details the qualities of each gearbox compared to the constraints on the state variables, using a wheel-surface static coefficient of friction of $\mu = 0.90$ because it is more selective. Data from step 1 through step 5 were recalculated with the new static coefficient of

Table 3: Progression of choosing a gearmotor through a list of state variables.

Gearbox	R	T_{stall} (lb· ft)	ω_{max} at T_{gm} (RPM)	Cost for 2 ($)
CONSTRAINT	–	> 0.6733	> 1189	< 150
no gearbox	1	0.6220	N/A	0
P61 Standard, 3:1	3.37	2.096	2530	101.66
P61 Standard, 4:1	4	2.488	2292	94.20
P61 Standard, 11:1	11.35	7.060	1002	123.26

friction value by means of a `MATLAB` code. The complete `MATLAB` code for the optimized gearmotor is detailed in Section 6 of the Appendix.

Because both the 3:1 and 4:1 gearboxes satisfied all of the state variable constraints, the 4:1 gearbox was chosen because it will allow the motor to operate at a higher RPM and lower torque, lengthening its lifetime and reducing that amount of current pull from the batteries.

References

[1] Ehsani, Mehrdad, et al. *Modern electric, hybrid electric, and fuel cell vehicles*. CRC press, 2018.

[2] *RS775 Motor - 24V*, www.banebots.com/product/M7-RS775-24.html.

Appendix

1. Trade Study

Design variables:
- *Design Variable 1*: Transmission driving a drive wheel.

Constraints on design variables:
- Transmission must be compatible with the motor and wheel (set by team).

State variables:
- *State Variable 1*: Output torque.
- *State Variable 2*: Output angular velocity.
- *State Variable 3*: Cost of transmission.

Constraints on state variables:
- Transmission must allow for the wheels to reach the slip condition without forcing the motor to operate above its maximum torque (set by team).
- Transmission must enable the the maximum angular speed at T_{gm} to be greater than the maximum angular velocity of the wheel.
- Both transmissions can have a sum cost no more than $150 (set by team).

Prioritization of state variables:

The State Variables have been prioritized in the order above in which they appear.

2. Friction coefficient values on different surfaces.

Weight on drive wheels: 4.11 lbs
Weight of payload: 29 lbs

Table 4: Force required to move the wheel or payload from rest (static) and to keep the payload sliding at a constant velocity (kinetic).

surface	wheel static	payload static	payload kinetic
classroom tile	1.67 lb	9.45 lb	7.40 lb
carpet	2.40 lb	11.0 lb	10.1 lb
marble stairwell	2.80 lb	14.7 lb	11.6 lb

Associated MATLAB Code:

```
% Friction coefficient parameters
% w - wheel     p - payload     s - static     r - rolling     k - kinetic
% [1 - classroom tile     2 - carpet     3 - marble stairwell]
driveWheelWeight = 4.11;                % [lb]
payloadWeight = 29;                     % [lb]
wsforce = [1.67 2.40 2.80];             % [lb]
psforce = [9.45 11.0 (12.3+17.1)/2];    % [lb]
pkforce = [7.40 10.1 11.6];             % [lb]

% Friction coefficient calculations
wsfriction = wsforce./driveWheelWeight;     % wheel static friction
psfriction = psforce./payloadWeight;        % payload static friction
pkfriction = pkforce./payloadWeight;        % payload kinetic friction
```

3. Maximum tractive force.

A force analysis of the vehicle in Figure 2, knowing that the vehicle will not accelerate in the y-direction, gives,

$$\begin{aligned}\sum F_y &= P_r + P_f - M_v \cdot g = 0 \\ P_f &= M_v \cdot g - P_r\end{aligned}$$

Assume that L is the total longitudinal distance between the front and back wheels, that is, $L = L_a + L_b$. A moment analysis of the vehicle in Figure 2, knowing that the vehicle will not accelerate around the center of gravity, gives,

$$\begin{aligned}\sum M_{COG} &= P_f \cdot L_a - P_r \cdot L_b = 0 \\ P_r \cdot (L - L_a) &= P_f \cdot L_a \\ P_r \cdot (L - L_a) &= (M_v \cdot g - P_r) \cdot L_a \\ P_r \cdot L \cancel{- P_r \cdot L_a} &= M_v \cdot g \cdot L_a \cancel{- P_r \cdot L_a} \\ P_r &= M_v \cdot g \cdot \frac{L_a}{L}\end{aligned}$$

Applying equation 1 to the tractive force F_t on the vehicle in Figure 2, gives,

$$F_{tmax} = P_r \cdot \mu_{ws} = M_v \cdot g \cdot \frac{L_a}{L} \cdot \mu_{ws}$$

where M_v is the mass of the vehicle, g is the acceleration due to gravity, and μ_{ws} is the static coefficient of friction of the wheels.

Associated MATLAB Code:

```
% Maximum tractive force parameters
Mv = 21/32.2;                 % mass of vehicle [slug]
g = 32.2;                     % acceleration due to gravity [ft/s/s]
L = 6.25/12;                  % length between front and back wheel [ft]
La = L*0.9;                   % length between COG and front wheel [ft]

% Maximum tractive force calculations
mu_ws = max(wsfriction);                     % wheel static friction
Ftmax = Mv*g*La/L*mu_ws;                     % [lb]
```

4. Maximum operating torque of the gearmotor.

A moment analysis of Figure 3, where F_{tmax} is the maximum tractive force of the drive wheels, r_w is the radius of the wheel, and T_{gmax} is the maximum slip condition torque of the gearmotor, gives

$$\sum M_{shaft} = \frac{F_{tmax}}{2} \cdot r_w - T_{gmax} = 0.$$

The team desires for the wheels to operate within 95% of the slip condition torque, so the maximum operating torque of the gearmotor is given by

$$T_{gm} = 0.95 \cdot T_{gmax} = 0.95 \cdot \frac{F_{tmax}}{2} \cdot r_w.$$

Associated MATLAB Code:

```
% Gearmotor parameters
slip_prct = 0.95;                    % target percentage of slip
rw = 1/12;                           % wheel radius [ft]

% Gearmotor calculations
Tgm = slip_prct*Ftmax/2*rw;          % max gearmotor torque [lb*ft]
```

5. Maximum acceleration, maximum deceleration, and switching position.

Figure 2 details a free-body diagram of the vehicle and payload. Newton's second law in the x-direction is given by

$$\begin{aligned}
\sum F_x &= M_{total} \cdot a_x \\
F_t - F_p &= (M_v + M_p) \cdot a_x \\
0.95 \cdot F_{tmax} - M_p \cdot g \cdot \mu_{pk} &= (M_v + M_p) \cdot a_{max} \\
a_{max} &= \frac{0.95 \cdot F_{tmax} - M_p \cdot g \cdot \mu_{pk}}{M_v + M_p}
\end{aligned}$$

Assume now that the vehicle and payload attempt to decelerate as quickly as possible. The payload will exert the same kinetic frictional force F_p, but the drive wheels on the vehicle will instead act against the direction of motion so F_t will change direction. The maximum deceleration of the vehicle-payload combination will be limited by whichever individual part will take more time to decelerate. Therefore, first a force analysis on the payload reveals how quickly the payload would decelerate on its own.

$$\begin{aligned}
\sum F_x &= M \cdot d_x \\
-F_p &= M_p \cdot d_p \\
-M_p \cdot g \cdot \mu_{pk} &= M_p \cdot d_p \\
d_p &= -g \cdot \mu_{pk}
\end{aligned}$$

Then, a force analysis on the vehicle reveals how quickly the vehicle would decelerate on its own.

$$\begin{aligned}
\sum F_x &= M \cdot d_x \\
-F_t &= M_v \cdot d_v \\
-0.95 \cdot Mv \cdot g \cdot \frac{La}{L} \cdot \mu_{ws} &= M_v \cdot d_v \\
d_v &= -0.95 \cdot g \cdot \frac{La}{L} \cdot \mu_{ws}
\end{aligned}$$

The maximum deceleration d_{max} will be whichever deceleration is smaller in magnitude. The determination of the switching position x_{switch} is found through constant acceleration equations. Assume that the desired stopping distance is at $x_{stop} = 22.5$ ft from the starting point.

$$\begin{aligned}
\text{Constant acceleration equation}: v^2 - v_0^2 &= 2 \cdot a \cdot \Delta x \\
\text{Acceleration portion of travel}: v_{switch}^2 - 0^2 &= 2 \cdot a_{max} \cdot (x_{switch} - 0) \\
\text{Deceleration portion of travel}: 0^2 - v_{switch}^2 &= 2 \cdot d_{max} \cdot (x_{stop} - x_{switch}) \\
v_{switch}^2 &= -2 \cdot d_{max} \cdot (x_{stop} - x_{switch}) \\
\text{Substitute}: \cancel{2} \cdot a_{max} \cdot (x_{switch} - 0) &= -\cancel{2} \cdot d_{max} \cdot (x_{stop} - x_{switch}) \\
a_{max} \cdot x_{switch} &= -d_{max} \cdot x_{stop} + d_{max} \cdot x_{switch} \\
x_{switch} = \frac{d_{max} \cdot x_{stop}}{d_{max} - a_{max}} &= \frac{x_{stop}}{1 - \frac{a_{max}}{d_{max}}}
\end{aligned}$$

The switch distance x_{switch} will not be greater than 20 ft, as there is no reason to continue to accelerate past 20 ft, even if the vehicle-payload combination could continue to accelerate and still be able to stop before 25 ft. Using the derivation direction above, the maximum velocity of the vehicle-payload, which is the same as the velocity at the switching position, is found to be

$$\begin{aligned} v_{switch}^2 - 0^2 &= 2 \cdot a_{max} \cdot (x_{switch} - 0) \\ v_{max} = v_{switch} &= \sqrt{2 \cdot a_{max} \cdot x_{switch}}. \end{aligned}$$

It is desirable to express the maximum vehicle velocity in terms of the maximum angular velocity of the gearmotor. Now that the speed and acceleration profile of the vehicle is known, it is necessary to calculate the amount of time it will take the vehicle to reach 20 ft, the entire purpose of the project.

$$\begin{aligned} \text{Constant acceleration equation} : x &= \frac{1}{2} \cdot a \cdot t^2 + v_0 \cdot t + x_0 \\ \text{Acceleration portion of travel} : x_{switch} &= \frac{1}{2} \cdot a_{max} \cdot t_1^2 + 0 + 0 \\ t_1 &= \sqrt{\frac{2 \cdot x_{switch}}{a_{max}}} \\ \text{Deceleration portion of travel} : 20\text{ ft} &= \frac{1}{2} \cdot d_{max} \cdot t_2^2 + v_{max} \cdot t + x_{switch} \\ t_2 &= \frac{-v_{max} + \sqrt{v_{max}^2 - 2 \cdot d_{max} \cdot (x_{switch} - 20\text{ ft})}}{d_{max}} \\ \text{Combine} : t &= t_1 + t_2 \end{aligned}$$

Associated MATLAB Code:

```
% Acceleration parameters
Mp = 29/32.2;                        % mass of payload [slug]
x_stop = 22.5;                       % stopping distance 20 < x < 25 [ft]

% Acceleration calculations
mu_pk = max(pkfriction);                               % payload kinetic friction
a_max = (slip_prct*Ftmax-Mp*g*mu_pk)/(Mv+Mp);   % max acceleration [ft/s/s]
d_p = -g*mu_pk;                 % max deceleration of the payload [ft/s/s]
d_v = -slip_prct*g*La/L*mu_ws;  % max deceleration of the vehicle [ft/s/s]
d_max = min(max(d_p,d_v),0);    % max deceleration [ft/s/s]
x_switch = min(x_stop/(1-a_max/d_max),20);% transition from acc to dec [ft]
v_max = sqrt(2*a_max*x_switch);         % max velocity of vehicle [ft/s]
w_max = v_max/(2*pi*rw)*60;             % max RPM of gearmotor [RPM]
t1 = sqrt(2*x_switch/a_max);            % acceleration portion time [s]
t2 = (-v_max+sqrt(v_max^2-2*d_max*(x_switch-20)))/d_max;    % dec time [s]
t = t1+t2;                              % total time
```

6. MATLAB Code

```
% Chris Kreienkamp
% Tue Oct 8 2019
% gearbox.m

clear
clc
clf



%% INITIALIZE PARAMETERS
% Friction coefficient parameters
% w - wheel     p - payload     s - static     r - rolling     k - kinetic
% [1 - classroom tile     2 - carpet     3 - marble stairwell]
driveWheelWeight = 4.11;                % [lb]
payloadWeight = 29;                     % [lb]
wsforce = [1.67 2.40 2.80];             % [lb]
psforce = [9.45 11.0 (12.3+17.1)/2];    % [lb]
pkforce = [7.40 10.1 11.6];             % [lb]

% Maximum tactive force parameters
Mv = 21/32.2;                % mass of vehicle [slug]
g = 32.2;                    % acceleration due to gravity [ft/s/s]
L = 6.25/12;                 % length between front and back wheel [ft]
La = L*0.9;                  % length between COG and front wheel [ft]

% Gearmotor parameters
slip_prct = 0.95;                    % target percentage of slip
rw = 1/12;                           % wheel radius [ft]

% Torque-speed parameters
R = 4;                               % INPUT GEAR RATIO HERE
Tstall = 119.43/12/16;               % stall torque [lb*ft]
wnoload = 12600;                     % no load RPM [RPM]

% Torque-current parameters
Inoload = 0.85;                      % no load current [A]
Istall = 59;                         % stall current [A]

% Acceleration parameters
Mp = 29/32.2;                        % mass of payload [slug]
x_stop = 22.5;                       % stopping distance 20 < x < 25 [ft]
```

```
%% CALCULATIONS
% Friction coefficient calculations
wsfriction = wsforce./driveWheelWeight;     % wheel static friction
psfriction = psforce./payloadWeight;        % payload static friction
pkfriction = pkforce./payloadWeight;        % payload kinetic friction

% Maximum tractive force calculations
%mu_ws = max(wsfriction);                    % wheel static friction
mu_ws = 0.9;                                % wheel static friction
Ftmax = Mv*g*La/L*mu_ws;                    % [lb]

% Gearmotor calculations
Tgm = slip_prct*Ftmax/2*rw;     % max gearmotor operating torque [lb*ft]

% Acceleration calculations
mu_pk = max(pkfriction);                        % payload kinetic friction
a_max = (slip_prct*Ftmax-Mp*g*mu_pk)/(Mv+Mp);   % max acceleration [ft/s/s]
d_p = -g*mu_pk;                 % max deceleration of the payload [ft/s/s]
d_v = -slip_prct*g*La/L*mu_ws;  % max deceleration of the vehicle [ft/s/s]
d_max = min(max(d_p,d_v),0);    % max deceleration [ft/s/s]
x_switch = min(x_stop/(1-a_max/d_max),20);% transition from acc to dec [ft]
v_max = sqrt(2*a_max*x_switch);         % max velocity of vehicle [ft/s]
w_max = v_max/(2*pi*rw)*60;             % max RPM of gearmotor [RPM]
t1 = sqrt(2*x_switch/a_max);            % acceleration portion time [s]
t2 = (-v_max+sqrt(v_max^2-2*d_max*(x_switch-20)))/d_max;    % dec time [s]
t = t1+t2;                              % total time



%% PLOT
% Torque-speed plot
figure(1)
w = linspace(0,wnoload,1000)/R;
T1 = Tstall*R-Tstall*R^2/wnoload*w;
plot(w,T1,'b-','LineWidth',3); hold on
plot(w,Tgm*ones(1000,1),'r--','LineWidth',3); hold off
xlabel('angular speed (RPM)'); ylabel('torque (ft\cdot lb)')
legend('gearmotor torque','max gearmotor torque'); set(gca,'FontSize',20)
axis([0 max(w) 0 max(T1)+max(T1)*0.1])
```