Crawler Power Tests and Analysis

- Where does the crawler power go?
- How can we choose an ideal motor speed and transmission ratio for efficiency?
- How can we measure force, power, speed to determine what we want to know?

The basic flow of power is shown in Fig. 1 for a crawler going up a hill:

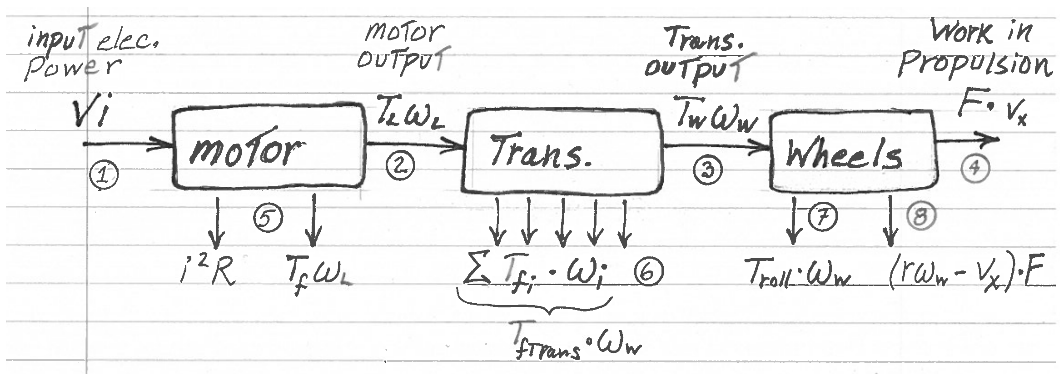


Figure 1: Schematic of power flow in the crawler. **Circled numbers match notes in text**.

1 Power and Loss Terms

1. Input power is just Vi. A power supply and ammeter are set up to measure voltage across the motor and current through the motor. We will be running the crawlers at a maximum of 9 volts.

2. Using the procedure from Assignment 4, we measure i_s and i_{nl} and ω_{nl} at several different voltages and use these measurements and the motor equations to determine the torque constant, k, resistance, R, and friction torque, T_f that describe our motor. Knowing these we can compute the output torque T_L (with T_f subtracted) and velocity ω_L of the motor for any voltage. We can also see if we are near the current and speed (i_η, ω_η) associated with maximum efficiency.

3. The input to the transmission is $T_L \omega_L$ and the output is $T_w \omega_w$ where the total speed ratio of the transmission is $N = \omega_L/\omega_w$. Due to power loss, $T_w < NT_L$.

4. The input to the wheels is the torque and speed applied to the drive axle(s). The "useful work" is the work required to propel the vehicle along the channel. In terms of power, $P_{out} = Fv_x$. In theory, F is the force required for propulsion if there were no rolling resistance.

5. Motor power is lost partly due to the coil resistance and partly due to the friction torque (see motor notes and video).

6. Power is lost at each transmission stage due to gears and bearings (Lego "bearings" are just shafts spinning in holes, so they're not especially efficient). It is convenient to lump the losses together as $T_{ftrans}\omega_w$. Note that the friction forces will tend to be proportional to the total forces and loads. So $T_{ftrans}\omega_w$ will tend to be a constant percentage of $T_w\omega_w$.

7. Rolling resistance is probably significant with high-traction tires. More about this in Section 4 below.

8. A second potential source of lost energy is slippage. Unlike gears, the wheels can slip so $v_x \leq r_w \omega_w$.

9. The system efficiency is $P_{out}/P_{in} = (F v_x)/(V i)$ and, with the various losses in the motor, transmission and wheels, it should not be surprising if the total efficiency is $< 10\%$.

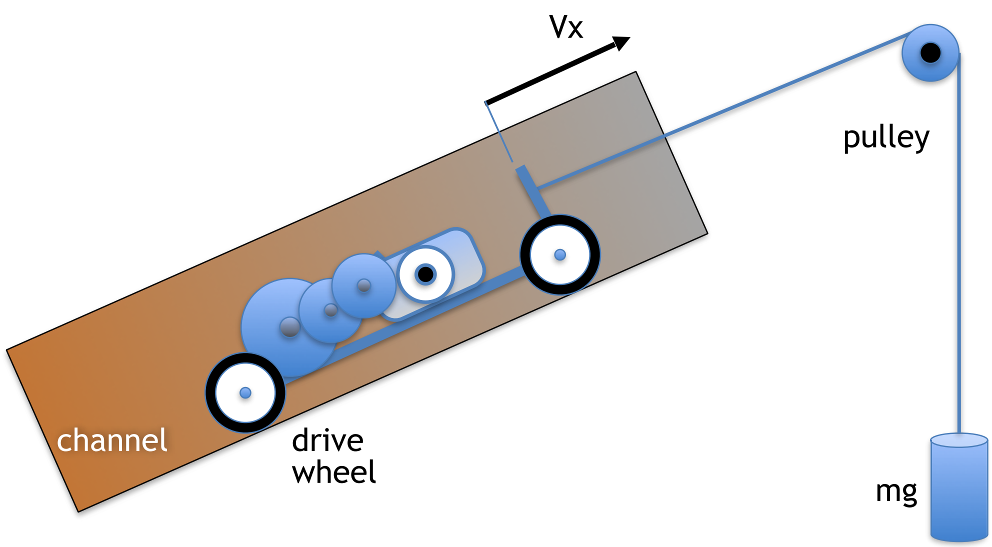


Figure 2: Measuring F and v_x in the channel

2 Testing Tips

1. When doing any tests, watch on the power supply whether the "current limit" light comes on – this means that the power supply cannot deliver as much current as required at the desired voltage. If this happens, you may be close to stalling. (If you measure the actual voltage across the motor terminals, it should still be correct.)

2. You can measure v_x using the stopwatch function on your cell phone to get the time required to travel 1m in the channel. It can also be very helpful to take some high speed video (newish iPhones and some other phones can do 60 frames/second or better). The high speed video makes it possible to detect slippage that is undetectable to the naked eye.

3. Getting an estimate of F can be a bit challenging, but is also possible (Fig. 2). Using a pulley and weights, you can estimate $F_{total} = F + F_{roll}$ where $F_{roll} = T_{roll}/r_w$. Add or remove sand from the container until the string just pulls your vehicle along, without accelerating. Use the digital scale to find out what the weight is. A subtle point, however, is that the force required to *pull* a vehicle through the channel may be slightly different (probably lower) than the force required to *drive* it, as discussed further in Section 4 below.

4. Once you've measured v_x, since you also know ω_L and ω_w from your motor information and speed ratio, you can estimate $v_{slip} = r_w\omega_w - v_x$. For more accuracy concerning ω_w, you may be able to measure the rotational speed of a shaft near the beginning of the transmission (near the motor)[1] by putting a disk with a piece of shiny tape on it and using a tachometer, while following the crawler in its track.

5. You can get a direct estimate of transmission losses by attaching a "drum" (which could be a Lego wheel with the tire removed) to the final axle of your system. Wrap a thin string around the drum and attach a weight below, as in the motor/winch test. The torque on the axle is then $T_w = (m_w g) r_{drum}$ where m_w is the weight and ideally you've chosen it so that the torque T_w is close to what you think it might be while crawling. Hold the machine in air and let the motor raise the weight. Now you know the input power from the motor and the output power (raising the weight) so you can estimate transmission losses – assuming that the gears and shafts are loaded similarly to how they are when crawling.

3 Design Tips

1. Larger wheels will have a proportionately smaller rolling friction and slippage, especially over bumps. (But they are also bigger and heavier...)

2. Precise, well-braced Lego construction will go a long way to increasing efficiency and reliability of the transmission. If shafts are bending, they will also have a lot more friction.

3. A little lubrication helps, but be careful you don't get carried away and make all your Legos unsnap. WD40 is not a good lubricant; it's a solvent and only lasts a short while. Vaseline is OK (but watch for getting dirt in it). Olive oil is good (Brunelleschi would be approve). Graphite (e.g., as used for locks) or dry Teflon is a reasonably good lubricant and less likely to trap dirt. In any case, excess lubricants tend to un-snap Legos.

4. Rubber bands do not make good belt drives. They slip too much and the work done in stretching and unstretching as they go around the small pulley greatly reduces the efficiency. If you really want a belt in a transmission you will need one with teeth.

5. Note that when you change the voltage, you also change the current, i, and speed, ω, at which peak efficiency occurs. So your design variables include the transmission ratio, the operating voltage, and the wheel radius.

More...

Additional tips and notes will be collected and updated as we discover new things for 2016 in the **Crawler News** page on Canvas.

[1] We want to put it near the motor because the tachometers do not work well when speeds drop below several hundred rpm)

4 Rolling Power Loss in Detail

From Fig. 1, the power out of the transmission is $T_w \omega_w$. It goes into 3 activities:

- "useful work": $f_x v_x$, where $f_x = mg\sin\theta$ – This is the work required to go up the slope.
- rolling loss: $T_{roll}\omega_{roll}$ – These are rolling losses (significant with soft tires). We take T_{roll} as the torque about the tire/road contact point required to keep the wheel rolling.
- slippage: $v_{slip}F_{slip}$, where $v_{slip} = r_w\omega_w - v_x$ and v_x is the measured forward velocity.

What are T_{roll} and F_{slip}?

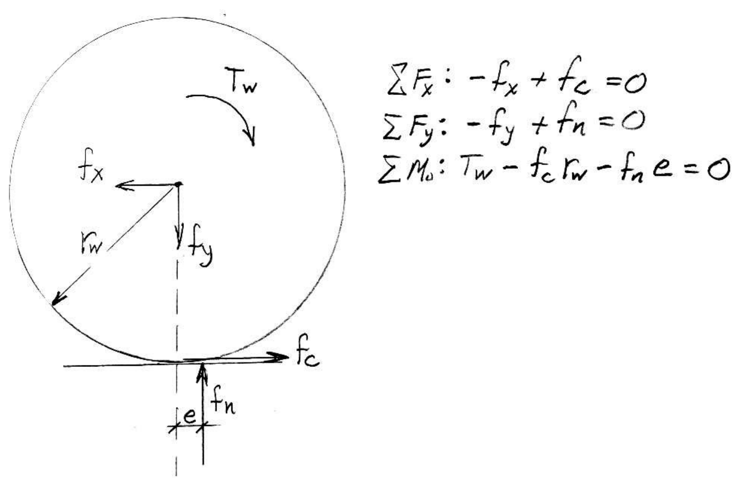
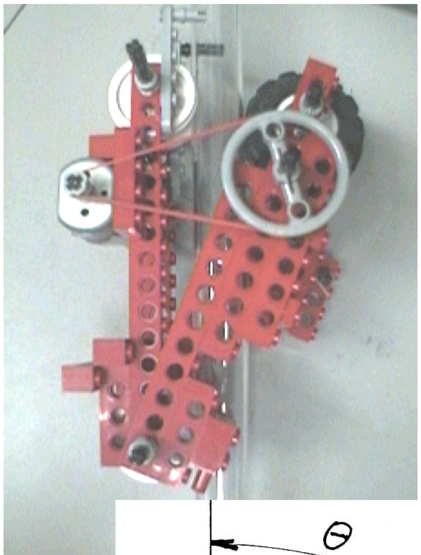


Figure 3: Wheel free body diagram. f_x and f_y are forces on the wheel at the axle; T_w is applied by the axle to the wheel; f_n and f_c are contact forces (where $f_c \leq \mu f_n$). In our example, $f_x = mg\sin\theta$. Moments, $\sum M_o$, are taken about the axle in this example.

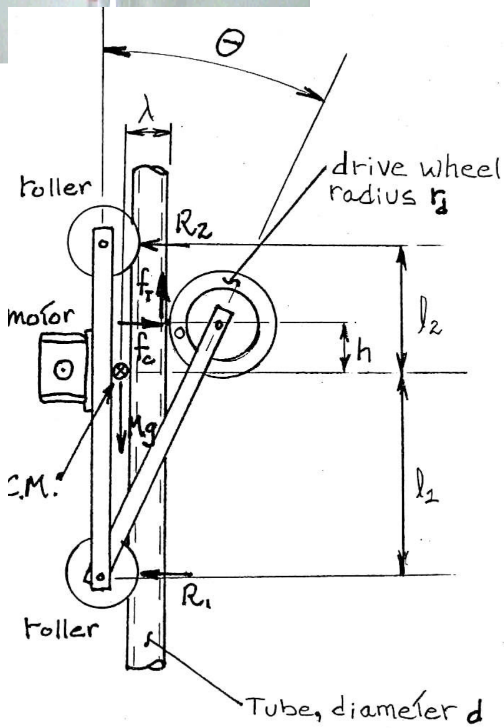
Recall the rolling resistance ($F_{roll}r_w$ or T_{roll}) for a car in Assignment 2. One standard way to define rolling resistance is with the diagram in Fig. 3. Due to rolling resistance, the centroid of the pressure distribution under the wheel is not at the centerline, but is slightly ahead of it, by an amount e (for *eccentricity*). So $T_{roll} = f_n e$. The power loss associated with this rolling resistance $P_{roll} = T_{roll}\omega_w$ because the wheel is instantaneously rotating with angular velocity ω_w about the contact. Now, if we define F_{roll} as the equivalent force needed to pull a cart along a horizontal surface when the only resistance is due to rolling, then by definition: $F_{roll} = T_{roll}/r_w$ and $v_{roll} = r_w\omega_w$.

- Meanwhile, the slippage power loss is given by $f_c v_{slip} = f_x(r_w\omega_w - v_x)$.
- The total losses are therefore: $P_{loss} = T_{roll}\omega_w + f_x(r_w\omega_w - v_x)$.

However, when we do the pulley test (Fig. 2) we define things a bit differently. We say that the "rolling resistance" is the force required to drag the vehicle along the channel (with $mg\sin\theta$ subtracted). The rolling power loss is therefore $P_{roll} = F_{roll}v_x$. Then we say that the slippage loss must be $P_{slip} = (r_w\omega_w - v_x)F_{slip}$ where $F_{slip} = f_x + F_{roll}$. The total power loss in operation is therefore $P_{loss} = F_{roll}v_x + (f_x + F_{roll})(r_w\omega_w - v_x)$ or, $P_{loss} = f_x(r_w\omega_w - v_x) + F_{roll}r_w\omega_w$, which is the same result as above. So we are using a slightly different definition of $F_{roll}v_{roll}$ and compensating with a correspondingly slightly different definition of F_{slip}.



The photo at left shows an early ME112 crawler that was designed to climb a clear plastic pipe. We will use it as an example to show how to do a basic force balance for the crawler as a whole and the drive wheel of a crawler design. (This would be an "exemplary" example of what we're looking for in the 1st design reviews.)



E.T.S. SIDE VIEW & MAJOR DIMENSIONS

EXTERNAL FORCES

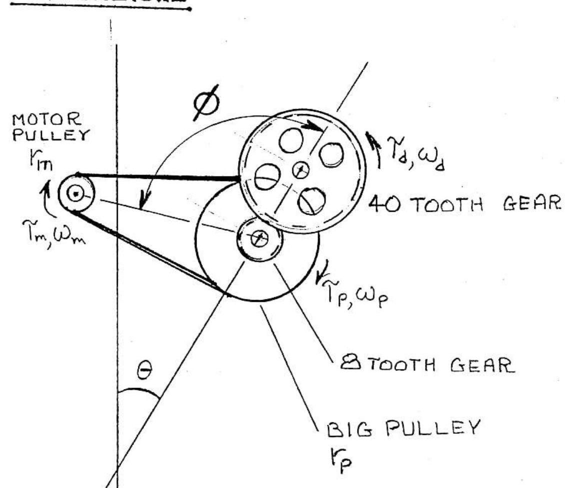
- Assume rollers have negligible friction so only normal forces: R_1, R_2
- If ETS moves @ CONSTANT speed then $f_T = Mg$ $(\Sigma F_y = 0)$
- We also know $f_T \le \mu f_c$
- From equilibrium:

$$\Sigma F_x = 0 = f_c - R_1 - R_2$$

$$\Sigma F_y = 0 = f_T - Mg$$

$$\Sigma M_0 = 0 = R_2(l_2 - h) + Mg\lambda - R_1(l_1 + h)$$

(CONTACT PT. OF DRIVE WHEEL)



DRIVE TRAIN SCHEMATIC

- **Velocity Ratios:**

$$\omega_p = \left(\frac{r_m}{r_p}\right)\omega_m, \quad \omega_d = \left(\frac{8}{40}\right)\omega_p$$

$$V_y = r_d\,\omega_d \quad (V_y = \text{climb speed})$$

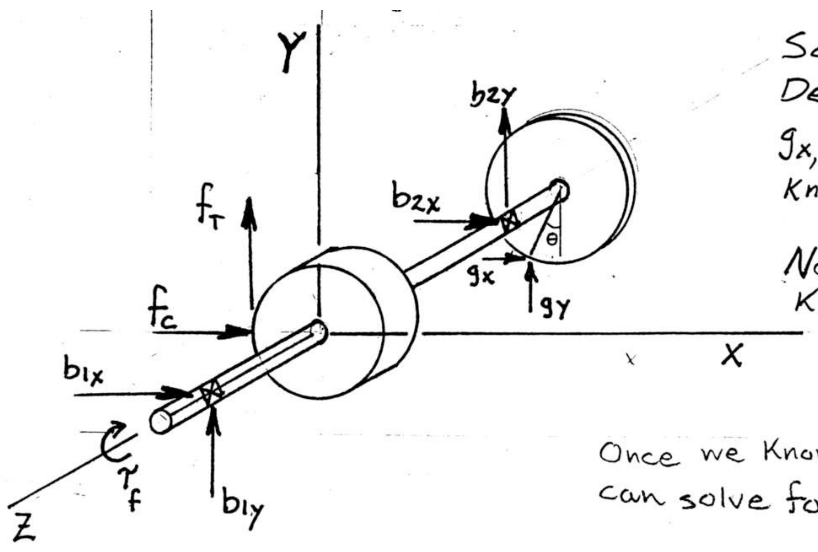
- **TORQUE RATIOS:**

Assume friction torques are approx. equal for p, d shafts; call it τ_f.

$$\tau_m\left(\frac{r_p}{r_m}\right) = \tau_p + \tau_f$$

$$\tau_p\left(\frac{40}{8}\right) = \tau_d + \tau_f \quad \text{so,}$$

$$\tau_d = \tau_m\left(\frac{r_p}{r_m}\right)\left(\frac{40}{8}\right) - 6\tau_f$$

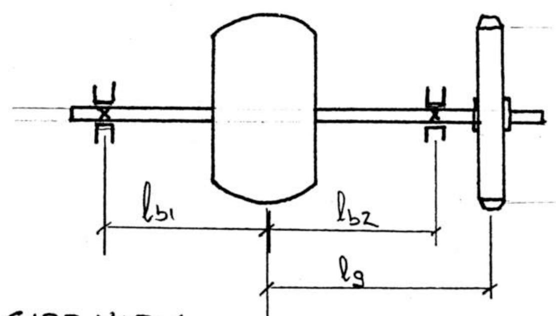


See Gears Detail To get g_x, g_y from Known τ_d.

Now, what is Known? Unknown?

Once we Know g_x, g_Y, f_c, f_T we can solve for b_1, b_2 loads

PERSPECTIVE - FINAL DRIVE SHAFT



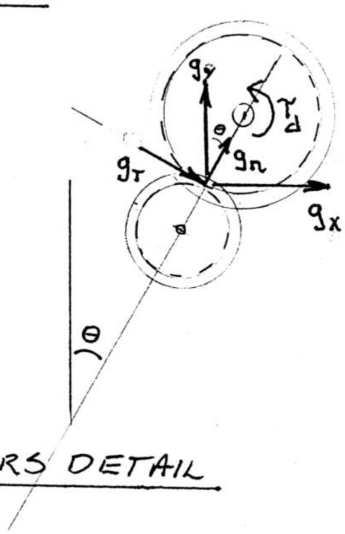
SIDE VIEW

$$\Sigma F_x = 0 = b_{1x} + b_{2x} + g_x + f_c$$

$$\Sigma M_y = 0 = b_{1x}\cdot l_{b1} - b_{2x}\cdot l_{b2} - g_x\cdot l_g$$

$$\Sigma F_y = 0 = b_{1y} + b_{2y} + f_T + g_y$$

$$\Sigma M_x = 0 = -b_{1y}\cdot l_{b1} + b_{2y}\cdot l_{b2} + g_y\cdot l_g$$



GEARS DETAIL

g_T is the Tangential force that produces τ_d: $\tau_d = g_T \cdot r_{40}$ where r_{40} = Pitch radius of 40 Tooth gear.

g_n is a side-effect. For most gears, $g_n = g_T \cdot \tan 20^\circ$ (20° pressure angle). Convert To XY coords:

gives: $g_x = g_T \cos\theta + g_n \sin\theta$, $g_y = g_n \cos\theta - g_T \sin\theta$

Once you have g_t, you can use Lewis gear stress formulas to see whether the tooth bending stresses are OK.