Individual Analytical Analysis

PHS Capstone Team

Pedal Power Generator System

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Introduction

The pedal power generation system is a student-built, off-grid energy platform designed to supply power Ponderosa High School's small agricultural greenhouse entirely through human-generated electricity. The central generation component is a stationary bicycle generator, in which a rider pedals in place against a mechanical load. The rear wheel of the stationary bicycle makes direct contact with a roller mounted on the shaft of an automotive alternator, transferring mechanical power from the drivetrain into electrical energy. The alternator output will pass through an MPPT charge controller before charging a 24 V battery bank, which powers greenhouse loads including grow lights and a ventilation fan for their resident bunny.

Because the bicycle is stationary, gear selection does not affect travel speed but instead affects the mechanical relationship between the rider's cadence, the force they must apply, and the resulting alternator shaft speed. Selecting too hard of a gear forces the rider to produce high torque at low cadence, which would be unsustainable, while selecting too easy of a gear reduces alternator speed and may fall below the 1000 RPM cut-in threshold. This analysis identifies the gear that maximizes electrical power delivered to the 24 V battery bank. A physiologically grounded Hill muscle force–cadence model is applied to avoid the arbitrary assumptions of linear interpolation used in calculation earlier this semester. Additionally, the parasitic power draw of the interactive display subsystem, which will be used to show real-time energy metrics to students, is quantified and subtracted to determine true net power output.

Assumptions

The following assumptions were used throughout this analysis:

- Rider output is modeled as a single "average" user based on the Hill linearized force–cadence model. Individual fitness variation is not captured.
- Alternator efficiency is treated as constant at 70% across all shaft speeds. In practice, efficiency will vary with speed. Please refer to Kaitlyn Phillips's analysis of the alternator for more details.
- Drivetrain, wheel–roller contact, and MPPT efficiencies are treated as constant loss fractions derived from published system measurements [3].
- The rider performs a natural cadence to each gear based on physical feel: harder gears (smaller sprockets) produce lower cadence, easier gears produce higher cadence.
- No rolling resistance or aerodynamic drag is present, as the bicycle is stationary. All of the rider's mechanical power enters the drivetrain.
- The display subsystem operates continuously and is powered from the same 24 V battery supply through a regulated 3.3 V rail.

Gear Selection Analysis

The stationary bicycle drivetrain connects a 34-tooth chainring to the rear cassette via a standard bicycle chain. The rear wheel then contacts a small roller fixed to the alternator shaft, providing a second mechanical gear-up stage. The combination of cassette gear ratio and wheel-to-roller ratio determines how many alternator revolutions occur per pedal revolution. System constants used in all calculations are provided in Table I. Calculations were performed and verified using a MATLAB script developed alongside this analysis, available in Appendix A.

Table I

System Constants

Symbol	Description	Value
R	Rear wheel radius	0.356 m
r	Crank arm length	0.190 m
r_{alt}	Alternator roller radius	0.025 m
N	Chainring teeth	34
n	Cassette sprocket teeth	11,13,15,18,22,26,34
F_{max}	Max isometric pedal force (Hill model)	400 N
ω_{max}	Unloaded max cadence (Hill model)	120 RPM
η_{alt}	Alternator efficiency	0.70
η_{drive}	Drivetrain efficiency	0.96
$\eta_{contact}$	Wheel–roller contact efficiency	0.96
η_{charge}	MPPT controller efficiency	0.95
ω_{min}	Alternator cut-in speed	1000 RPM

Force–Cadence Model

Rather than assuming a linear force-cadence tradeoff between arbitrary endpoints, this analysis adopts the Hill muscle force-velocity relationship. This is a well-established biomechanical model used throughout cycling science literature [2]. In its linearized form for sub-maximal, sustained pedaling effort, the relationship between sustainable tangential pedal force F and cadence ω is given by Equation 1, where F_{max} is the maximum isometric force at zero cadence and ω_{max} is the theoretical unloaded cadence at which no useful force is produced [2][5].

$$F\,[N] = F_{max} \times \frac{1 - \omega}{\omega_{max}} \tag{1}$$

where:

- F – tangential pedal force (N)
- F_{max} – maximum isometric pedal force, recreational cyclist (Hill Model) (N) [5]
- ω – pedaling cadence (RPM)
- ω_{max} – unloaded maximum cadence, recreational cyclist (Hill Model) (RPM) [2]

A key property of this model is that mechanical power P_{mech} becomes a parabola in cadence, with a theoretical maximum at the optimal cadence given by Equation 2. For the parameters above, this provides a physically meaningful prediction: the gear that places the rider closest to half of ω_{max} should produce the highest power output.

$$\omega_{optimal}\,[\text{RPM}] = \frac{\omega_{max}}{2} \tag{2}$$

Gear Ratio and Alternator Speed

The bicycle cassette gear ratio and wheel-to-roller gear-up are calculated in Equations 3 and 4 respectively. The resulting alternator shaft speed is given by Equation 5 [2].

$$G_{bike} = \frac{N}{n} \tag{3}$$

$$G_{alt} = \frac{R}{r_{alt}} \tag{4}$$

$$\omega_{alt} = \omega G_{bike} G_{alt} \tag{5}$$

where:

G_{bike} – bicycle cassette gear ratio
G_{alt} – wheel–roller gear-up ratio
N – chainring teeth
n – selected cassette sprocket teeth
R – rear wheel radius (m)
r_{alt} – alternator roller radius (m)
ω_{alt} – alternator shaft speed (RPM)

Power and System Efficiency

Rider mechanical power is the product of crank torque and pedaling cadence in rad/s, as given by Equations 6 and 7. The overall system efficiency η_sys (Eqn. 8) combines all conversion stage losses, and the resulting electrical power delivered to the 24 V battery is given by Equation 9 [3].

$$\tau\, [N \times m] = Fr \tag{6}$$

$$P_{mech}\, [W] = \tau\omega \tag{7}$$

$$\eta_{sys} = \eta_{alt}\eta_{drive}\eta_{contact}\eta_{charge} \tag{8}$$

$$P\, [W] = P_{mech}\eta_{sys} \tag{9}$$

where:

τ – crank torque (Nm)
r – crank arm length
η_{alt} – alternator electromechanical efficiency (assumed)
η_{drive} – chain/sprocket drivetrain efficiency [3]
$\eta_{contact}$ – wheel–roller direct contact efficiency [3]
η_{charge} – MPPT charge controller efficiency [3]
η_{sys} – overall system efficiency

Applying these equations across all seven cassette gears yields Figure 1 and Table II. The cadence assigned to each gear reflects the natural tendency of a stationary rider to pedal at lower cadence in harder gears and higher cadence in easier gears. Pedal force at each cadence is derived directly from the Hill model (Eqn. 1) rather than assumed from fixed endpoints.

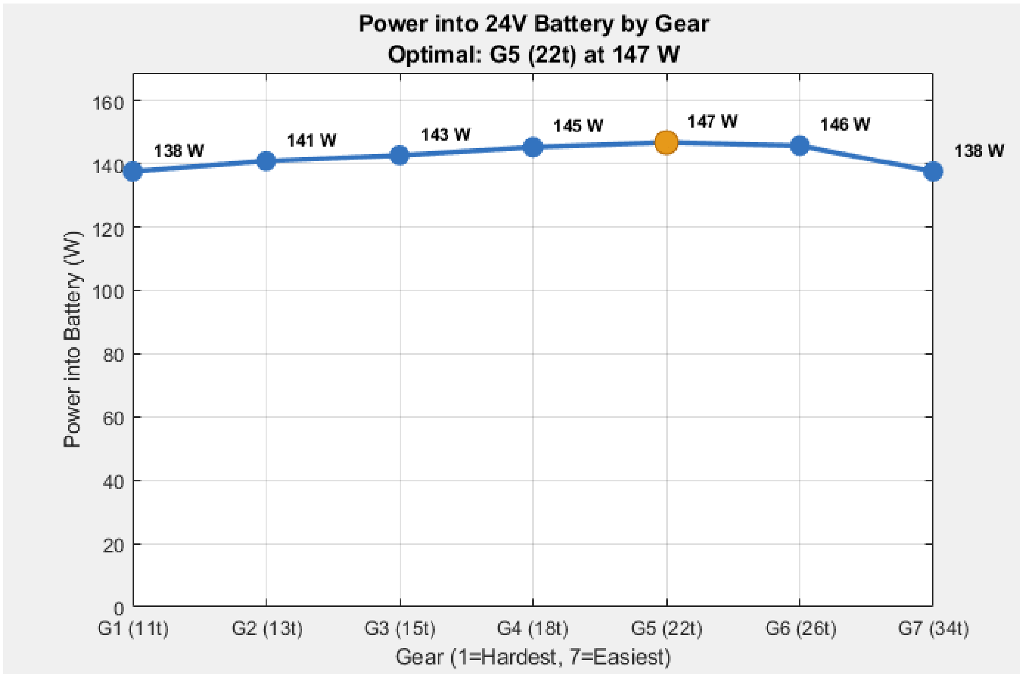

Figure 1. Gear Analysis Results

Table II

Gear Analysis Results

Gear	Teeth	ω (RPM)	F (N)	ω_alt (RPM)	P_mech (W)	P_batt (W)
G1	11	45	250	1981	223.8	137.2
G2	13	48	240	1788	229.2	140.5
G3	15	50	233	1614	232.1	142.3
G4	18	54	220	1452	236.4	144.9
G5	**22**	**60**	**200**	**1320**	**238.8**	**146.6**
G6	26	65	183	1210	237.1	145.3
G7	34	75	150	1068	223.8	137.2

G5 (22t, 60 RPM) produces the highest battery power at 146.6 W, consistent with the theoretical prediction from Equation 2. This result differs from the linear interpolation model used in prior calculations, which predicted a nearly flat power curve peaking at G4. The power curve is parabolic rather than flat, with output declining more noticeably at G1 and G7 than at the central gears. All seven gears remain above the 1000 RPM alternator cut-in threshold, with G7 producing a minimum shaft speed of

1068 RPM. The 146.6 W peak is consistent with the established recreational cyclist sustained output range of 75–150 W [2][5].

Display Subsystem Net Power Output

The PHS bicycle generator includes an interactive display subsystem that shows real-time energy metrics to students. This subsystem draws power from the 24 V battery through a regulated 3.3 V supply, constituting a parasitic load on the generator output. To quantify this overhead and determine the true net power delivered to useful battery storage, the average power draw of the display was modeled as a function of screen refresh interval.

The subsystem consists of three active components: an ESP32 microcontroller, a 7.5" E-Ink display panel, and an INA219 current sensor. Each component operates in either an active draw state during a refresh event or a low-power standby state between refreshes. Because E-Ink technology retains its image without continuous power, most operating time is spent in standby, making average consumption highly sensitive to refresh interval. Component electrical specifications are provided in Table III.

Table III

Display Component Specifications

Component	i_a (mA)	i_s (mA)	t_w (ms)	Source
ESP32	11	0.800	12	[8]
7.5" E-Ink Display	7.9	0.001	200	[9]
INA219 Current Sensor	1.0	0.006	1	[10]
Arduino Uno (baseline)	50	—	—	[11]
16×2 LCD (baseline)	25	—	—	[12]

where:

i_a – active current draw during refresh (mA)
i_s – standby current draw between refreshes (mA)
t_w – wake/active time per component per refresh cycle (ms)

For a given refresh interval *T*, the total active time per cycle is found by summing the wake time of each component, as given by Eq. (10).

$$t_r = \sum t_w \qquad (10)$$

The active and standby current contributions are then time-weighted over the refresh cycle. During the wake period t_w, each component draws its active current i_a. For the remainder of the cycle, each draws its standby current i_s. These are expressed in Eq. (11) and Eq. (12), respectively.

$$I_{active} = (\sum i_a)(\frac{t_r}{T}) \qquad (11)$$

$$I_{standby} = \left(\sum i_s\right)\left(\frac{T - t_r}{T}\right) \quad (12)$$

Average current is the sum of these two contributions, given by Eq. (13).

$$I_{avg} = I_{active} + I_{standby} \quad (13)$$

Average power consumption of the subsystem is then calculated from the standard electrical power relation, Eq. (14), and net power delivered to the battery subtracts this overhead from the generator output, Eq. (15).

$$P_{avg} = I_{avg} \times v_s \quad (14)$$

$$P_{net} = P_{out} - P_{avg} \quad (15)$$

These equations were evaluated across five refresh intervals ranging from 2 s to 60 s. Results are provided in Table 4, along with the baseline power draw of the Arduino Uno and 16×2 LCD combination (247.5 mW constant) for comparison.

Table IV

Net Power Results by Refresh Interval

T (s)	I_avg (mA)	P_avg (mW)	P_avg Baseline (mW)	P_net (W)	Improvement (%)
2	2.840	9.37	247.5	146.591	96.21
5	1.620	5.35	247.5	146.595	97.84
10	1.214	4.01	247.5	146.596	98.38
30	0.943	3.11	247.5	146.597	98.74
60	0.875	2.89	247.5	146.597	98.83

At the minimum permissible refresh interval of 2 s, average subsystem draw peaks at 9.37 mW, yielding a net output of 146.590 W. At 10 s (the selected operating interval), draw falls to 4.00 mW and net output is 146.596 W. Across all tested intervals the ESP32 and E-Ink architecture reduces parasitic consumption by over 96% compared to the Arduino and LCD baseline of 247.5 mW. This reduction is primarily due to the E-Ink panel's near-zero standby current and the ESP32's aggressive low-power mode between refresh cycles. The display subsystem imposes no meaningful penalty on net battery power output.

Discussion

The results of both analyses inform two design decisions. First, G5 (22t) should be designated the recommended operating gear for the stationary generator, as it places the rider at the biomechanically optimal cadence of 60 RPM per the Hill model. G4 (18t) produces nearly identical output (145.0 W) and may suit riders who prefer a slightly lower cadence of 54 RPM. Both gears should be clearly labeled on the physical gear shifter, with instructional materials indicating the recommended range.

The two-bike configuration provides an additional pedagogical opportunity. By designating one bicycle to operate in G3 (15t) or G4 (18t) and the other in a contrasting gear such as G6 (26t) or G7 (34t), students can directly observe how gear selection affects the force-cadence tradeoff and its downstream effect on generator output. G3 and G4 are well-suited for the 'high resistance' demonstration bike, as they sit on the ascending side of the power curve where increasing gear difficulty still yields marginal power gains — making the tradeoff between effort and output tangible and measurable. This hands-on comparison reinforces the central design finding that neither extreme of the cassette is optimal, and that maximum power is achieved by matching rider cadence to the biomechanical sweet spot predicted by the Hill model.

Second, the G7 gear (34t, 75 RPM) remains viable but produces the lowest output at 137.5 W due to the Hill model force reduction at high cadence. G1 (11t, 45 RPM) also outputs 137.4 W, penalized by the lower cadence despite higher force. These extreme gears may still serve a warm-up function but should not be promoted as primary generation gears. All gears satisfy the 1000 RPM alternator cut-in constraint; G7 produces a shaft speed of 1068 RPM, confirming the wheel–roller geometry provides just sufficient mechanical advantage at the easiest setting.

The 61.4% overall system efficiency identifies the alternator (70%) as the largest single loss stage. Upgrading to a higher-efficiency alternator, for example rated at 80%, would raise η_{sys} to 70.2%, increasing peak output from 146.6 W to 167.6 W for the same rider effort: a 14% gain. This should be considered in future component selection.

Physical prototype testing is required to validate the Hill model parameters, as F_{max} and ω_{max} represent population-level averages that will vary among individual student users. Resistance-cadence measurements from the completed prototype will allow the model to be calibrated and gear recommendations refined.

Conclusion

This analysis determined the optimal gear for the PHS stationary bicycle generator and quantified the net electrical power available to the battery bank after accounting for display subsystem overhead. Using a Hill muscle force–cadence model, G5 (22t, 60 RPM) was identified as the optimal gear, delivering 146.6 W to the 24 V battery at an overall system efficiency of 61.4%. All seven gears satisfy the alternator cut-in constraint. The ESP32 and E-Ink display subsystem imposes a maximum parasitic draw of 9.4 mW at the shortest allowable refresh interval, reducing net output by less than 0.01 W, a greater than 96% improvement over the Arduino and LCD baseline. These results provide a quantitative basis for gear labeling, rider instructions, and future component upgrade decisions.

References

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Appendix A: MATLAB Code

```
clear; clc; close all;
%% Constants

% Bike geometry
r_wheel         = 0.3556;        % m, standard 24" wheel radius
chainring       = 34;           % teeth, donated bike chainring

% 7-speed MTB cassette (teeth) (From best donated bike)
cassette        = [11, 13, 15, 18, 22, 26, 34];
n_gears         = length(cassette);
gear_labels     = arrayfun(@(x) sprintf('G%d (%dt)', find(cassette==x), x), cassette,
'UniformOutput', false);

% Alternator contact roller
r_roller        = 0.025;        % m, 25mm radius roller on alternator shaft

% Generic automotive alternator specs
RPM_cutin       = 1000;         % RPM, minimum for useful output
RPM_rated       = 6000;         % RPM, rated speed
V_sys           = 24;           % V, target system voltage
eta_alt         = 0.70;         % alternator efficiency
eta_drive       = 0.96;         % drivetrain efficiency
eta_contact     = 0.96;         % wheel-roller contact efficiency (slip loss)
eta_controller  = 0.95;         % MPPT charge controller efficiency

% Rider inputs
r_crank         = 0.1905;        % m, standard crank arm length
% Note: cadence and force are now gear-dependent (see Force-Cadence model below)

%% Force-cadence tradoff model
% Key insight: gear determines the resistance load on the rider.
% Harder gear (small sprocket) = more torque required = cadence drops.
% Easier gear (large sprocket) = less torque required = cadence rises.
% Using Hill model

% Overall efficiency
eta_total = eta_drive * eta_contact * eta_alt * eta_controller;

% Force range per gear — harder gear allows more force (leverage), easier = less
F_max   = 400;    % N, max isometric force at zero cadence (Hill model)
```

```
w_max   = 120;    % RPM, unloaded max cadence (Hill model)

% Calculate forces
cadence_sustainable = [45, 48, 50, 54, 60, 65, 75]; % Hill model 'natural' riding
cadence
force_sustainable   = F_max * (1 - cadence_sustainable / w_max);

% Preallocate
omega_alt   = zeros(1, n_gears);
P_mech      = zeros(1, n_gears);
P_battery   = zeros(1, n_gears);
usable      = false(1, n_gears);

for g = 1:n_gears
    G_bike = chainring / cassette(g);
    omega_ped = cadence_sustainable(g);                     % sustainable cadence for
this gear
    omega_wheel = omega_ped * G_bike;
    omega_alt(g) = omega_wheel * (r_wheel / r_roller);      % alternator RPM

    % Mechanical power
    omega_rad = omega_ped * (2*pi/60);
    tau = force_sustainable(g) * r_crank;
    P_mech(g) = tau * omega_rad;

    % Power into battery
    if omega_alt(g) >= RPM_cutin
        usable(g) = true;
        P_battery(g) = P_mech(g) * eta_total;
    end
end

%% In-Line Results Table

fprintf('\n============================================================\n');
fprintf('   PEDAL POWER ANALYSIS – 24V System (Force-Cadence Model)\n');
fprintf('==========================================================\n');
fprintf('Overall system efficiency: %.1f%%\n', eta_total*100);
fprintf('Roller gear-up ratio (wheel to alternator): %.1fx\n\n', r_wheel/r_roller);

fprintf('%-16s %10s %10s %10s %12s %14s\n', 'Gear', 'Cadence', 'Force(N)', 'Alt RPM',
'P_mech(W)', 'P_battery(W)');
```

```
fprintf('%s\n', repmat('-',1,75));
for g = 1:n_gears
    if usable(g)
        fprintf('%-16s %9.1f  %9.1f  %9.0f  %10.1f  %12.1f\n', ...
            gear_labels{g}, cadence_sustainable(g), force_sustainable(g), ...
            omega_alt(g), P_mech(g), P_battery(g));
    else
        fprintf('%-16s %9.1f  %9.1f  %9.0f  %10.1f  %12s\n', ...
            gear_labels{g}, cadence_sustainable(g), force_sustainable(g), ...
            omega_alt(g), P_mech(g), '< cut-in');
    end
end
fprintf('\nCut-in RPM threshold: %d RPM\n', RPM_cutin);
[P_max, g_max] = max(P_battery);
fprintf('Optimal gear: %s — %.1fW into battery\n', gear_labels{g_max}, P_max);

%% Plot Data
figure('Name','Power into Battery by Gear','Position',[100 100 700 460]);
x = 1:n_gears;
xlabels = arrayfun(@(i) sprintf('G%d (%dt)', i, cassette(i)), 1:n_gears,
'UniformOutput', false);
P_plot = P_battery;
P_plot(~usable) = NaN;

plot(x, P_plot, '-o', 'Color', [0.2 0.45 0.75], 'LineWidth', 2.5, 'MarkerSize', 8,
'MarkerFaceColor', [0.2 0.45 0.75]);
hold on;

% Highlight optimal gear
plot(g_max, P_max, 'o', 'MarkerSize', 12, 'MarkerFaceColor', [0.9 0.6 0.1],
'MarkerEdgeColor', [0.7 0.4 0]);

% Value labels
for g = 1:n_gears
    if usable(g)
        text(g + 0.15, P_battery(g) + 7, sprintf('%.0f W', P_battery(g)), ...
    'HorizontalAlignment','left','FontSize',9,'FontWeight','bold');
    end
end

xlabel('Gear (1=Hardest, 7=Easiest)');
ylabel('Power into Battery (W)');
```

```
title(sprintf('Power into 24V Battery by Gear\nOptimal: %s at %.0f W',
gear_labels{g_max}, P_max), 'FontSize', 12);
xticks(x); xticklabels(xlabels);
ylim([0, max(P_battery)*1.15]);
grid on; box on;
```