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Modeling of Bradley Tracked Vehicle Steering and Fuel Consumption with a Detailed Kinematic Model of the HMPT500-3 Transmission and Simplified Vehicle Dynamics

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ABSTRACT

The HMPT500-3 is a split torque path hydrostatic / mechanical CVT used in the Bradley Fighting Vehicle. A previous paper detailed a linear algebraic approach to model forward operation of the HMPT500-3 without steering using a reduced equation set. This model was expanded to the full equation set to allow transmission operation with steering. The equations showed that opposite to a typical automotive "open" differential, the HMPT500-3 enforces a speed difference between the sprockets for steering, but does not have an inherent torque bias. The typical regenerative steering torque from the decelerated inside track must be provided by interfacing with a vehicle model. A simplified 2-D planar dynamics model of the Bradley was developed to explore vehicle performance and fuel consumption with steering. The integrated model showed that fuel consumption during minimum radius turns can double that of straight-ahead operation at the same speed. Commercial vehicle performance codes lack tracked steering models, so test courses are run as simple gradient vs. distance profiles which would result in underestimating fuel consumption. The Munson Fuel Loop course was selected to attempt to quantify the contribution of steering to total fuel consumption. Fuel consumption calculations with steering were found to be 9% to 14% higher than those without steering depending on lap speed, and compared more favorably with Bradley test data.

INTRODUCTION

The HMPT500-3 is a split torque path hydrostatic / mechanical continuously variable transmission used in the Bradley Fighting Vehicle. Power transmission and steering is accomplished through the complex interaction of six planetary gear sets and two variable displacement hydrostatic pump / motor units (HSUs), connected through intermediate shafts and gears. A previous paper (GVSETS 2012, ref [1]) detailed a linear algebraic approach to model forward operation of the HMPT500-3 without steering using a reduced equation set. This model was expanded to the full 22 simultaneous speed / torque equations to allow transmission operation with steering to be modeled. The equation set showed that opposite to a typical automotive "open" differential, the HMPT500-3 as a hydrostatic controlled differential enforces a *speed* difference between the sprockets for steering, but does not have an inherent *torque* bias between the sprockets. The typical regenerative steering torque from the decelerated inside track must be provided by interfacing with a vehicle dynamics model. A simplified 2-D planar dynamics model of the Bradley was developed to allow the transmission model to be fully exercised. This model uses lumped inertias for the vehicle body, running gear and drivetrain. Longitudinal gradient, rolling and aerodynamic resistances are accounted for and steering lateral dynamics are handled by modified Merritt steering equations. The Merritt-model provides a closed form solution for the required steering moment and longitudinal track slip for steady state turns at a given radius and vehicle speed based on simple vehicle geometry and terrain factors. Exercising the integrated transmission & vehicle model showed that fuel consumption during minimum radius turns can reach double that of straight-ahead operation at the same speed. Current commercial codes for vehicle performance and fuel consumption lack tracked steering models, so test courses are run as simple gradient vs. distance profiles. This would result in underestimating fuel consumption, so it was decided to attempt to quantify the contribution of steering to total fuel consumption on a typical test course. The Munson Fuel Loop course was selected as full profile data was available, as well as actual Bradley fuel consumption data. Course gradient and curvature (inverse of radius) were calculated at each step along the course. Pending development of a PID driver sophisticated enough to reliably follow the course, the vehicle model was used to generate fuel burn rate lookup tables over a wide range of speeds, grades and turn radii from infinity (straight road) to the minimum radius capability. The course speed at each location was then determined as the minimum of the target lap speed, speed on grade, speed in turn or acceleration limited speed recovery capability. This allowed calculation of the time to complete each distance step, and fuel burn rates with and without steering. The resulting integrated fuel consumption and average speed were then compared to vehicle test results on the Munson course over a range of speeds. Fuel consumption calculations with steering were found to be ~ 9 - 14% higher depending on lap-speed than those that ignored the effect of steering, and compared more favorably with the test data.

THE TRANSMISSION / DRIVETRAIN MODEL

Details of the simultaneous speed and torque equations used to model the HMPT500-3 for forward operation are available in a previous paper [1]. The simplified model ignored the steering differential, which uses a double-pinion planetary gear to provide a true averaging of the Left / Right HSU outputs. As before, the planetary gear set is described by one speed and two torque equations. With N denoting the number of gear teeth, S rotational speed and T torque and subscripts 1 thru 3 denoting the sun, carrier and ring gear respectively:

$$N_1 S_1 + (N_3 - N_1) S_2 - N_3 S_3 = 0 \qquad (1)$$

$$\frac{T_1}{N_1} + \frac{T_2}{(N_3 - N_1)} + \frac{2T_3}{N_3} = 0 \qquad (2)$$

$$\frac{T_1}{N_1} + \frac{T_3}{N_3} = 0 \text{ and } N_3 = 2N_1 \qquad (3)$$

The resulting 22 simultaneous equations for speed can be solved for each range of transmission operation to determine left and right output speeds based on input (engine) speed and left / right HSU actuation strokes. The set of 22 simultaneous torque equations is however under-determined, and some additional relationship between left and right output torques is required. This suggests that the HMPT500-3 as a hydrostatic controlled differential has no inherent left-right output torque bias, even while enforcing a speed difference between the two outputs for steering. This behavior is opposite to a typical automotive "open" differential, which allows any output speed difference, while balancing the output torques. The complete steering performance of the transmission cannot be determined in isolation, and the typical regenerative steering torque from the decelerated inside track must be provided by interfacing with a vehicle dynamics model.

The complete drivetrain model is comprised of the engine, transmission and final drives / running gear as shown in figure-1. The constant final drive ratio is included in the overall gear ratios between the engine and left / right outputs. Lumped rotating inertias are used for the engine at the transmission input and the sprocket-track system at the outputs.

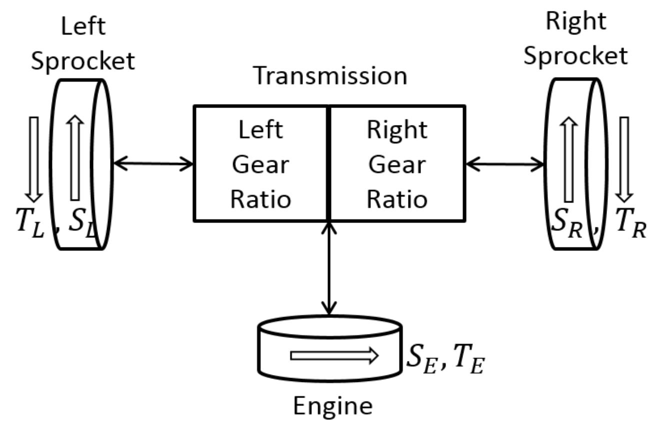

Figure 1.

Neglecting thru-put efficiencies for the moment, the system is described by the following torque balance:

$$\begin{aligned} T_E - G_L T_L - G_R T_R &= (I_E + G_L^2 I_L + G_R^2 I_R) S_E' \\ &+ (G_L G_L' I_L + G_R G_R' I_R) S_E \end{aligned} \qquad (4)$$

$$where\ S_L = G_L S_E ,\ S_R = G_R S_E \qquad (5)$$

The second inertia term is applicable only to CVT's that are varying the gear ratio between the engine and output(s) at a finite rate to control vehicle speed and also in this case to provide steering.

The engine is modeled using lookup tables for torque and fuel consumption vs. speed when operating on the "Command Schedule" provided in [2]. Operation along this engine scheduling curve is enforced by the transmission controller in response to the driver throttle and vehicle speed.

THE VEHICLE MODEL

A simplified 2-D planar dynamics model of the Bradley was developed to allow the drivetrain model to be fully exercised. This model uses lumped translational and yaw inertias for the vehicle body, and incorporates the drivetrain model. Longitudinal gradient, rolling and aerodynamic resistances are accounted for and steering lateral dynamics are handled by modified Merritt steering equations. The forces acting on the vehicle in a turn are shown in figure-2.

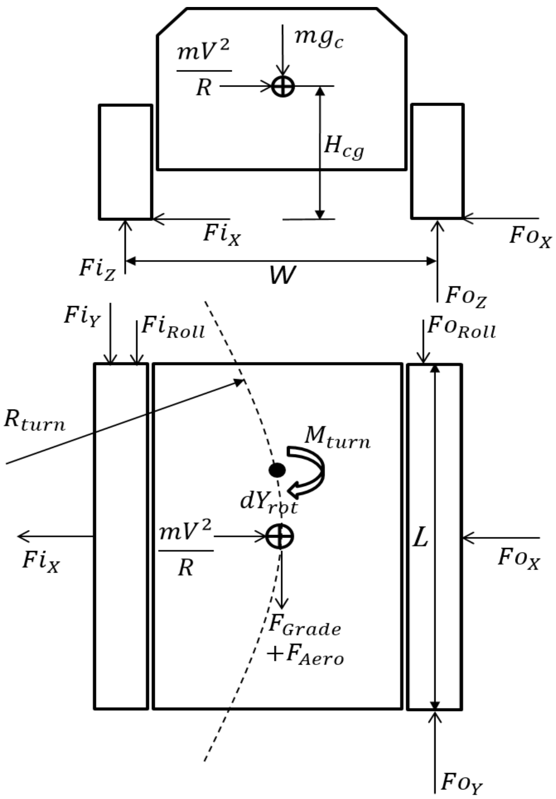

Figure 2.

A speed dependent rolling resistance coefficient C_{RR} is determined from a towing motion resistance test of the M2A2. The simultaneous equations for balancing longitudinal and lateral forces and roll and yaw moments will not be given here, as they are thoroughly treated in the classic text [3].

While there are many more complex tracked-steering models in existence, the Merritt steering model was selected as it provides a reasonably accurate closed-form solution for the required steering moment and longitudinal track slip for steady state turns at a given radius and vehicle speed based on simple vehicle geometry and terrain factors. The Merritt equations use empirical steering constants K_0 and K_1, an empirical track slip factor β and friction coefficient μ. K_0 and β are functions of the vehicle geometry factor L/W, and K_1 is a function of turn radius [4]. Longitudinal track slip is used to determine the actual radius of turn based on inside and outside (relative to the turn) sprocket speeds:

$$R_{turn} = \left(\frac{W + \beta L}{2}\right)\left(\frac{S_o + S_i}{S_o - S_i}\right) \qquad (6)$$

In the Merritt model the equal and opposite inside and outside track slips cancel and thus do not change the vehicle speed calculated from the average sprocket speed. While not used in the original Merritt model, the forward shift of the vehicle center of rotation to balance centripetal forces shown in figure 2 is generally accepted [3]:

$$dY_{rot} = \frac{LV^2}{2g_c\mu R} \qquad (7)$$

This offset, as well as the different vertical loads on the inside and outside track due to the roll moment result in a modified Merritt turning resistance moment:

$$M_{turn} = mg_c\mu\left\{K_0K_1\frac{W}{2} + \frac{dY^2}{L}\right\} + m\frac{V^2}{R}\left\{\frac{C_{RR}}{100}H_{cg} - dY\right\} \qquad (8)$$

The calculated turning resistance was found to be in reasonable agreement with data collected from an M109 test vehicle over a range of speeds and radii as shown in figure-3 [4]. No equivalent Bradley data has been found.

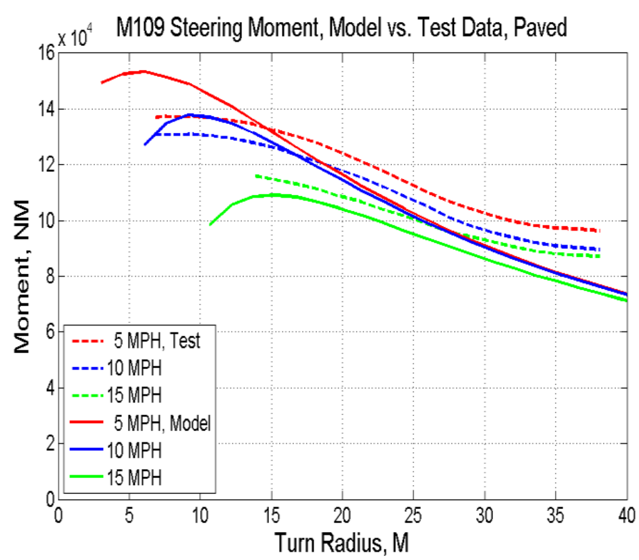

Figure 3.

The integrated Bradley drivetrain & vehicle model linear dynamic equations are solved simultaneously for engine and left / right sprocket angular accelerations and the resulting vehicle body translational and yaw accelerations. An explicit three step Adams-Bashforth integrator is used to calculate velocities and displacements at each time step.

Throttle control is by PID tuned according to the Pessen integral rule. The steer command input rate is a differential HSU actuation stroke scaled by the first derivative of the Logistics Function (“S” function) and terminated when the desired radius of turn is achieved.

MODEL OUTPUTS

In the current configuration the model is able to perform a prescribed steering maneuver by accelerating from rest to a set speed, entering and holding steer to a set radius of turn while maintaining speed, and then exiting the turn to continue straight ahead. Details of all drivetrain controls such as engine throttle, transmission range and left / right HSU actuation are recorded. All forces, accelerations and velocities as well as engine fuel burn rate are also recorded. A set of detailed outputs for steering to a 50 M radius on a level secondary road at 25 kph is provided in figures-7, 8 in the Addendum.

While the main purpose of the model is to investigate steering, validation runs were performed to ensure that straight-line performance also matched available test data. As shown in figure-4, speed on grade performance showed good agreement with test data [5].

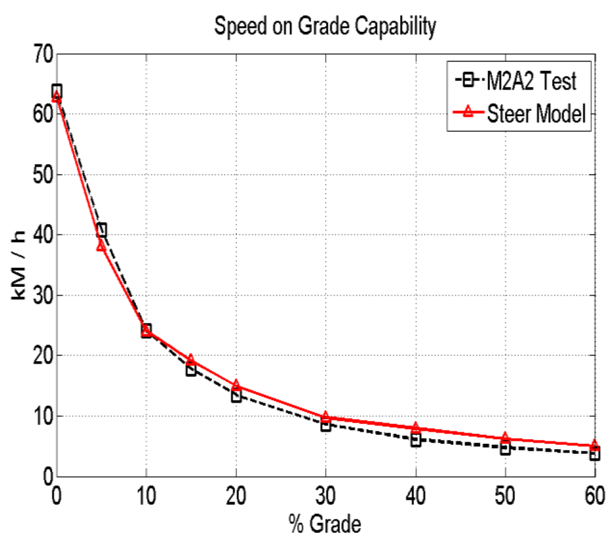

Figure 4.

Unfortunately no non-pivot steer testing data for the Bradley has been found to complete the model validation. A constant speed sweep of turn radii from ∞ (no steer) to the minimum turn radius capability showed that while the effect of steering on fuel consumption for large radii was minimal, fuel consumption during heavy steer maneuvers can double that of straight-ahead operation, as seen in figure-5.

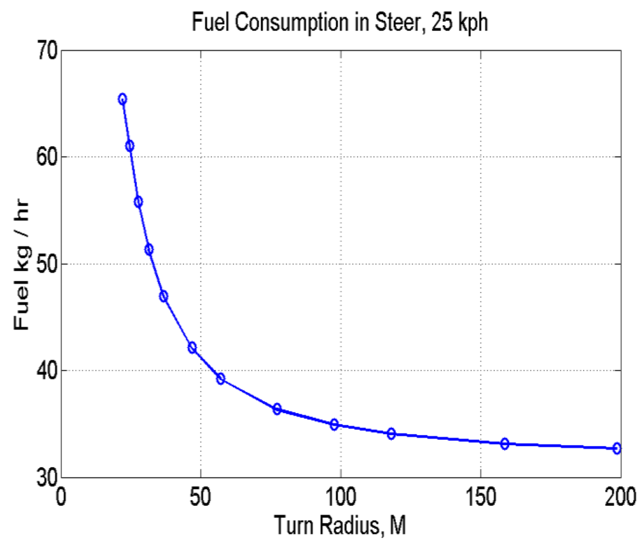

Figure 5.

Commercial vehicle performance codes lack tracked-vehicle steering models, and can only run test courses as simple gradient vs. distance profiles. This would result in underestimating fuel consumption, so an attempt was made to quantify the contribution of steering to total fuel consumption on a typical course. The Munson Loop is one of the Army’s standard fuel consumption test course for wheeled and tracked vehicles. It is used to determine fuel consumption rates that might be expected under field conditions [6]. This course was selected as full profile data was available, as well as actual Bradley fuel consumption data for model validation. Parametric curves were generated for GPS Easting, Northing and elevation vs. distance which allowed course curvature (inverse of radius) and gradient to be calculated at a 1M increment along the course. The course has twelve turns in its 2.43 km length, some of which are small radius switch-backs as seen in figure 6. Steering could thus be expected to have a noticeable effect on fuel consumption on the Munson course.

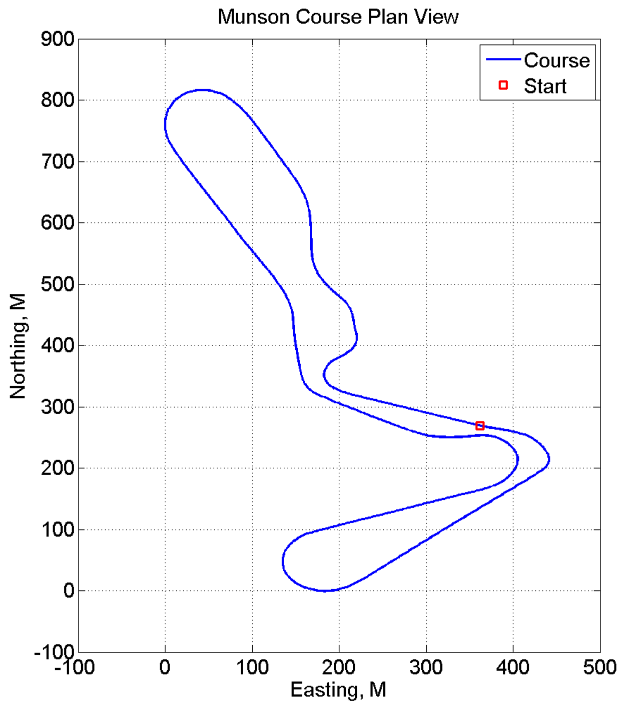

Figure 6.

Pending development of a PID driver sophisticated enough to reliably follow the course, the vehicle model was used to generate fuel burn rate lookup tables over a wide range of speeds, grades and turn radii from ∞ (straight road) to the minimum radius capability. To account for the dynamic speed profile on the course where the vehicle might be forced to slow below the target speed due to grades or turns, speed recovery capability was determined from the full throttle acceleration speed and distance vs. time profiles, corrected for gradient. The course speed at each location was then determined as the minimum of the target lap speed, speed on grade, speed in turn or acceleration limited speed recovery capability. This allowed calculation of the time to complete each distance step and fuel burn rates with or without steering. For the non-steering case the curvature of the entire course was set to zero, but the calculations were the same in all other respects. A set of detailed outputs for traversing the Munson course in the CW direction without and with steering for a 45 kph target speed is provided in figure-9 in the Addendum. The resulting integrated fuel consumption and average speed were then compared to results from testing of the M2A2 Bradley from 15 to 45 kph [5]. As shown in figure-7, model fuel consumption without steering is consistently lower than the test data. Model fuel consumption with steering is much closer to the test data, and duplicates the shape of the curve better.

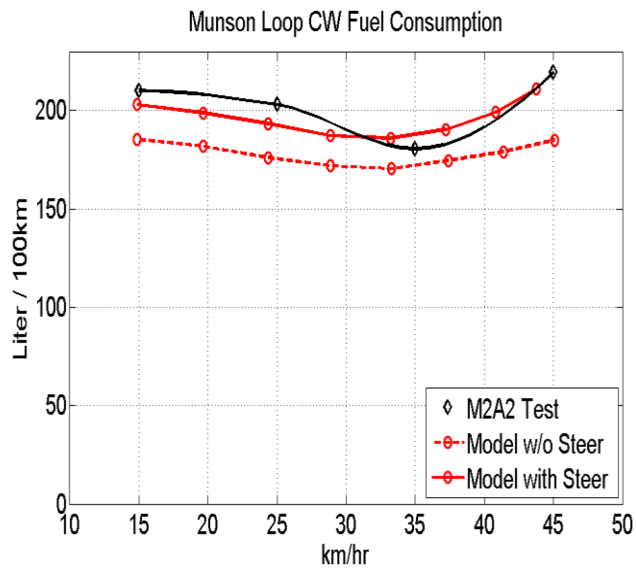

Figure 7.

Comparison of model output without and with steering in table-1 shows that steering accounts for a relatively constant 9% additional fuel consumption until the vehicle must begin slowing below the target speed to make turns, as shown by the relative decrease in average speed for the steering vs. non-steering case. From that point the extra fuel consumed for the steering case increases with speed, reaching 14% for the highest speed run. This is due to the extra fuel burned in accelerating back to the target speed after exiting turns.

Munson CW Model Output					
	W/O Steer		With Steer		
Vtarget	Vavg	Fuel Econ	Vavg	Fuel Econ	Steer Δ
kph	kph	L /100 kM	kph	L /100 kM	%
15	14.9	185.5	14.9	202.9	9.4
20	19.7	181.7	19.7	198.8	9.4
25	24.3	176.1	24.3	193.3	9.8
30	28.9	171.8	28.9	187.1	8.9
35	33.3	170.5	33.2	185.9	9.0
40	37.4	174.5	37.2	190.4	9.1
45	41.4	179.1	40.8	199.1	11.2
50	45.1	184.9	43.8	210.9	14.1

Table 1.

As modeled the vehicle never exceeds the target speed, while it is understood that test drivers will attempt to "make up time" where possible, such as on descents. This may affect the results by increasing the average lap speed without significantly increasing fuel consumed.

CONCLUSIONS

A powertrain using a detailed kinematic model of the HMPT500-3 transmission was successfully integrated with a simplified vehicle dynamics model of the Bradley Fighting Vehicle. This model has the capability to explore the full spectrum of tracked vehicle operation including steering, which commercial vehicle performance codes lack. While not yet fully developed, the model demonstrated its utility in a preliminary investigation of the influence of tracked vehicle steering on fuel consumption on a typical Army test course. Steering was shown to account for ~9 - 14% additional fuel consumption on the Munson Fuel Loop compared to running the gradient profile alone, depending on vehicle speed. The "quasi-dynamic" method used as an interim approach for simulating running of the course showed good agreement with the test data.

DISCLAIMER

Reference herein to any specific commercial company, product, process or service by trade name, trademark, manufacturer or otherwise, does not necessarily constitute or imply its endorsement, recommendation or favoring by the Unites States Government or the Department of the Army. The opinions of the authors expressed herein do not necessarily state or reflect those of the United States Government or the Department of the Army, and shall not be used for advertising or product endorsement services.

REFERENCES

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ADDENDUM

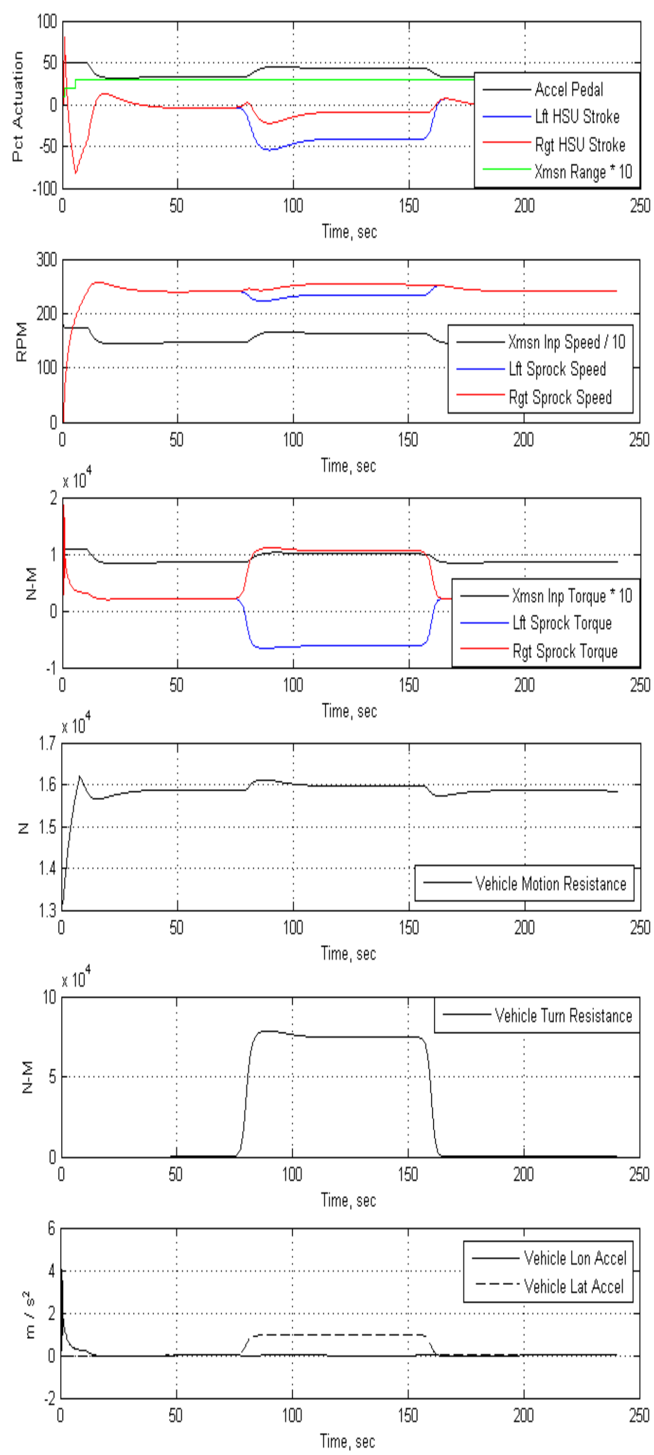

Figure 7.

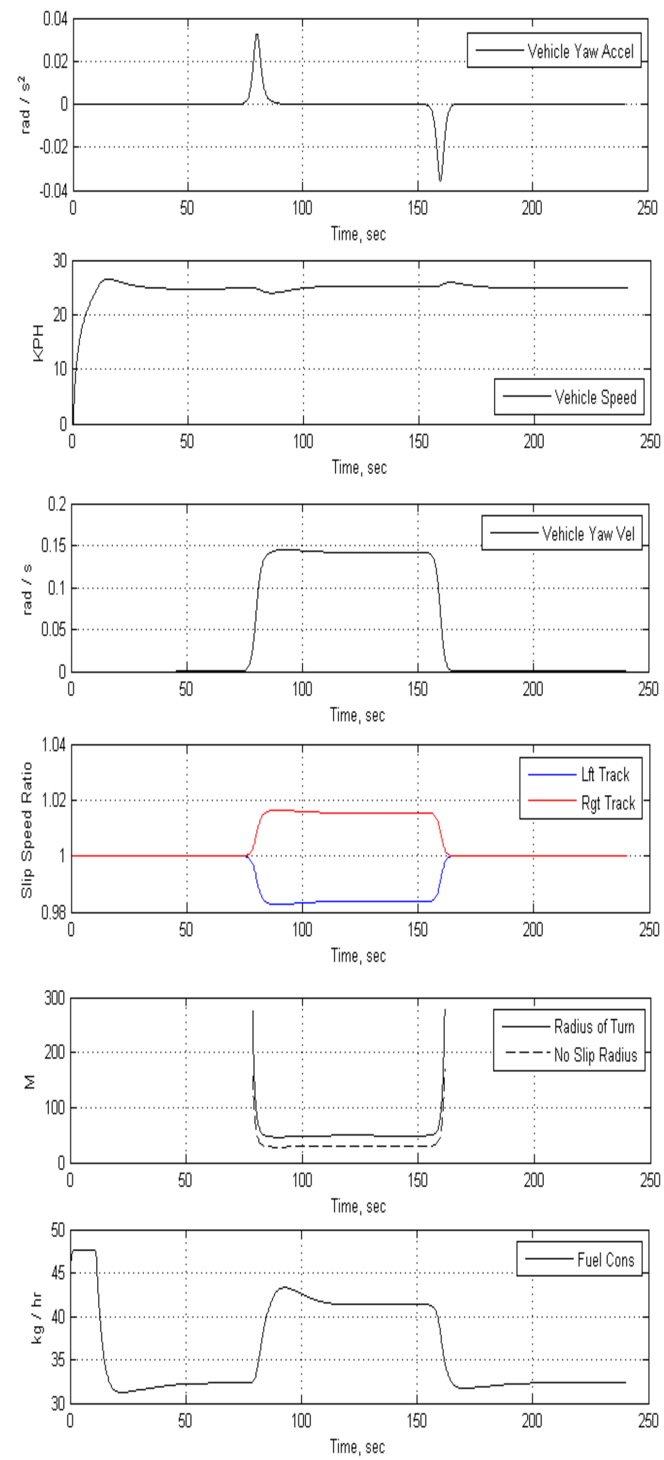

Figure 8.

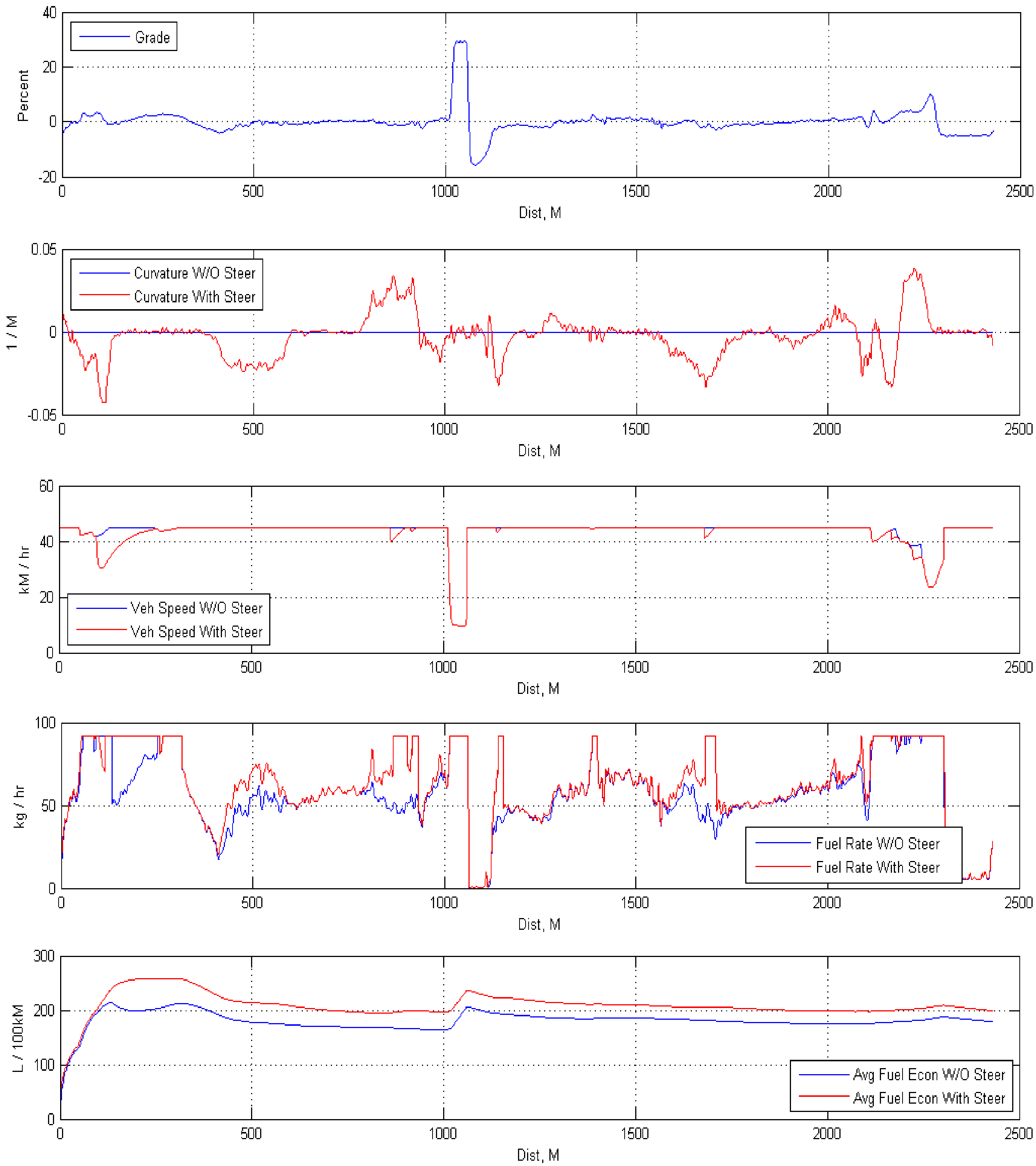
Grade
Percent
Dist, M
Curvature W/O Steer
Curvature With Steer
1 / M
Dist, M
Veh Speed W/O Steer
Veh Speed With Steer
kM / hr
Dist, M
Fuel Rate W/O Steer
Fuel Rate With Steer
kg / hr
Dist, M
Avg Fuel Econ W/O Steer
Avg Fuel Econ With Steer
L / 100kM
Dist, M

Figure 9.