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The Development of a Dynamic-Interactive-Vehicle Model for Modeling Traffic beyond A Microscopic Level (Paper #08-3059)

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THE DEVELOPMENT OF A DYNAMIC-INTERACTIVE-VEHICLE MODEL FOR MODELING TRAFFIC BEYOND A MICROSCOPIC LEVEL

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ABSTRACT

This research effort is geared towards better representation of vehicle dynamics in traffic simulation. The result will not only lead to improved understanding of traffic dynamics but also provide insight into highway design, traffic safety, more accurate estimates of vehicle emissions, and transportation forensics. The objective of this paper is to propose a simple and computationally affordable dynamic vehicle model that is capable of representing vehicle movement with high fidelity. The objective is accomplished by mathematical representations of a vehicle's power-train, brake and steering mechanisms while providing a medium through which a driver can interact with these components. The effects of some of the more dominant external forces on vehicle motion as also accounted for in the vehicle model. Validation results demonstrate that this model is successful in representing acceleration, braking and steering performance of three passenger cars.

INTRODUCTION

This research effort is geared towards further improving the representation of vehicle dynamics in traffic simulation. The result will not only lead to improved understanding of traffic dynamics but also provide insight into highway design, traffic safety, more accurate estimates of vehicle emissions, and transportation forensics.

State-of-the-art representation of vehicle dynamics in traffic simulation is lumped into car-following and simplified vehicle dynamics models. Although some of these models are capable of high fidelity representation of vehicle movement and subsequent traffic dynamics, there are a few dimensions of vehicle dynamics that inadequately treated in these models. These dimensions include acceleration / deceleration performance within a vehicle's capabilities, the interactions amongst the vehicle, driver and environment and the representation of lateral movement.

The key in accounting for these dimensions and their corresponding effects on vehicle motion, and on traffic dynamics, is to include particular vehicle specifications which define a vehicle's performance capabilities. While allowing driver inputs such as brake and gas pedal positions along with dialed steering angles to affect the representation of vehicle motion. Upon incorporating these dimensions, more realistic values of acceleration, speed, and position will be obtained for a vehicle's motion. There have been a number of research efforts that have attempted to account for these dimensions but none has incorporated all into a single model for use in traffic simulation.

This paper will propose a Dynamic-Interactive-Vehicle (DIV) model which incorporates all the aforementioned dimensions. Additionally, this model will output a unique two-dimensional representation of vehicle motion. The manner in which the DIV model will account for these dimensions is by formulating mathematical representations of three of the more integral component of a vehicle – the engine / power-train, brake and steering mechanisms – that are primarily responsible for the movement of a vehicle. This will be done while providing a medium through which a driver can interact with these components. The effects on vehicle motion due to some of the more dominant external forces will also be considered in the DIV model. The DIV model will be capable of replicating high-fidelity vehicle movement, while remaining simple and computationally affordable for use in traffic simulation.

The following sections of the paper will present a brief review of the state-of-the-art of vehicle dynamics representation in traffic simulation and afterwards the development of the DIV model. After the development of the model, the paper will present the calibration and validation process of the DIV model as well as the conclusion and future directions of this research effort.

LITERATURE REVIEW

Vehicle motion is primarily represented in traffic simulation via car-following models. There are several car-following models available today, some of which have been implemented in traffic simulators. Many of these

models have a common structure which essentially has the action / *response* of the following vehicle as a function of a *stimulus* and a *sensitivity* factor. The main difference between some of these models with this structure is how they define each term. In general, the *response* refers to the acceleration of the following car as it tries to avoid colliding with the lead car. The *stimulus* is often times a function of the difference in distance and / or speed between the following and lead car. And as for the *sensitivity* factor it, in essence determines the weight of the *response* as a function of the following vehicle's speed and / or its distance behind the lead vehicle. Models with this structure that have been partly or fully implemented in traffic simulators include the Pipes model (Pipes 1953) and the GHR / GM model (Gazis, Herman et al. 1959). What is noteworthy here is that these models do not directly account for the acceleration / deceleration capabilities of the vehicles they are representing. Instead maximum and minimum acceleration rates of the vehicles being represented are specified by users of these models.

As researchers attempt to improve the representation of how a vehicle follows another, models presented in (Gipps 1981), (Benekohal and Treiterer 1988) and (Helbing, Hennecke et al. 2002) incorporated measures designed to capture realistic acceleration performances. Despite the relative success of these models, they did not manage to fully capture the realism that is associated with such performances. To further improve how acceleration performance is treated in traffic simulation several researchers turn their attention to develop more detailed models.

The work presented in (Rakha, Lucic et al. 2001; Rakha and Lucic 2002; Rakha, Snare et al. 2003) highlights the formulation and comparison of a vehicle dynamics model which is capable of successfully predicting maximum acceleration performance. The model takes into account the effect of the vehicle's engine force and other dynamic properties along with aerodynamic, rolling and grade resistances. This model is a very comprehensive model not only taking into consideration the limitations of the vehicle but also most of the significant external forces that will affect a vehicle's motion.

The realism with which vehicle motion is represented in traffic simulation has increased over the years through the use of existing car-following and simplified vehicle dynamics models. Further improvement maybe made to these modeling efforts by extending the dynamic modeling to the braking and steering systems of a vehicle. Additional improvement may also be had by providing a medium through which a driver can interact with the vehicle. Extending past research efforts in the modeling of vehicle dynamics will not only lead to improved understanding of traffic dynamics but also provide insights into highway design, traffic safety, more accurate estimates of vehicle emissions, and transportation forensics.

The following section will serve to detail the development of the DIV model which will be formulated with the aforementioned improvements in mind to extend to use of traffic simulation.

DEVELOPMENT OF THE DIV MODEL

Overview of the DIV Model

The DIV model is an opened ended model that is aimed at faithfully replicating vehicle motion. This DIV model is capable of accepting inputs from a driver and the environment while outputting a vehicle's acceleration, velocity and position in the x-y plane. The DIV model will also take into consideration the dominant forces that act on a vehicle to influence its motion. Such forces include those due to a vehicle's engine and braking system, gravity, rolling resistance and aerodynamic resistance.

The DIV model will be capable of accepting three inputs from a driver of a vehicle – throttle position, brake pedal position and the steering angle. The manner in which the model will treat these input is to relate each to a particular desire that a driver may have and represent these various desires on a scale of 0 – 1 for the throttle and brake positions and -1 to +1 for the steering angle. Each of these parameters will then interact with their corresponding mechanisms to produce motion.

The following sections will present how DIV model treats the various components of a vehicle in order to faithfully capture its motion. These components include the engine, the braking system and the steering mechanism. Details of how the DIV-model will account for effects due to rolling resistance, air resistance and gravity will also be presented in the following sections.

Modeling Longitudinal Movement

Forces in the x-direction that the DIV model is looking to take into account are the forces due to the engine and the braking system, rolling and aerodynamic resistances and the force due to gravity. The equation of motion such a vehicle can be derived by using Newton second law of motion:

$$\sum F = ma = \frac{W}{g}a = F_e - F_b - R_a - R_r - R_g \tag{1}$$

Where:

W	=	weight of the vehicle
g	=	acceleration due to gravity
F_e	=	tractive force produced by the engine (N)
F_b	=	force produce by the brake (N)
R_a	=	aerodynamic Resistance (N)
R_r	=	rolling resistance (N)
R_g	=	grade resistance (N)

Modeling Acceleration Performance

One of the more unique aspects of the DIV model is its engine model. The engine model that will be incorporated in the DIV model was adapted from (Ni 2007). This model manages to successfully represent the complexity of the inner workings of an internal combustion engine in a very simple and straight forward manner. At the base of this model is an analytical approach which looks at interaction at between the gas pedal and the force produced by the engine body.

The essence of this approach is that when a driver applies pressure to the gas pedal, the orientation of the throttle valve changes. The change in orientation determines the amount of air flowing into the inlet manifold. The fuel injector then releases fuel in a quantity that corresponds to the amount of air flowing throw the inlet manifold. This combination of air and fuel is then passed into the cylinders of the engine block, which then combusts. This combustion produces a force on the crankshaft, which is then transmitted through the power-train mechanism of the vehicle, causing the vehicle's wheel to turn.

From the above mentioned approach, the key to replicating how an engine works to produce a force that propels a vehicle forward is the amount of fuel being injected in the engine block. But in order to determine the amount of fuel being injected the amount of air flowing in the inlet valve must first be determined. The mass of air flowing into the engine's cylinders, m_a, is equal to

$$m_a = \rho_2 \cdot V_a \tag{2}$$

where:

ρ_2 - density of the air in the combustion chamber
V_a - Total volume of the air in the combustion chamber of the engine

However ρ_2 and V_a need to be calculated first. The volume of air in all the combustion chambers of the engine, V_a, is essentially the product of the engine's displacement / volume and its speed; resulting in:

$$V_a = V_e \cdot \frac{1}{2} \cdot \frac{\omega}{2\pi} \tag{3}$$

where:

V_e - engine displacement (m^3)
ω - engine speed (rad/s)

As for the density of air is the combustion chamber, ρ_2, it was calculated based on Bernoulli's Principle which, in essence states that an increase in the speed of a fluid results in a decrease in the pressure or gravitational energy experience by that fluid as long as there no work being done on the fluid. In calculating ρ_2 the cross-sectional areas of the inlet manifold and the opened throttle, along with their respective air pressure and densities were used. Therefore after several iterations:

$$\rho_2 = \rho_0 - \frac{\rho_0^{\,2} V_a^{\,2}}{2 A_0^{\,2} p_0}\left(\frac{1}{\theta} + \frac{A_0^{\,2}}{A_2^{\,2}} - 2\right) \tag{4}$$

where:

A_0 - cross-sectional area of the inlet manifold (m^3)
A_2 - cross-sectional area of the inlet valve (m^3)
p_0 - density of air flow before the throttle (kg/ m^3)
p_0 - air pressure before the throttle (N/ m^2)
θ - throttle position (percent of throttle opening)

After computing the mass of air being taken into to the engine block the corresponding amount of fuel maybe estimated using the Stoichiometric air-fuel ratio (λ) for gasoline, which is 6.8%. Having a value for the amount of fuel entering the engine, the amount of power that maybe generated by this quantity of fuel can be determined by using the energy fuel density of gasoline, E_f, which is 46.9 MJ/kg. It is known that the efficiency of an internal combustion engine, η, is not 100% and as a result corresponding calculations have to take this fact into account. The effective power, P_{eff}, the power that will be delivered to the vehicle's power-train mechanism is defined as:

$$P_{eff} = \eta \lambda E_f m_a \tag{5}$$

With the effective power calculated the effective torque, T_{eff}, delivered to the wheels of the vehicle can be determined with the following relationship.

$$T_{eff} = \frac{P_{eff}}{\omega} \tag{6}$$

Using the effective torque being delivered to the wheel, the effective engine force, F_e, produce by the engine to promote vehicle motion can therefore be calculated with the aid of the appropriate final transmission gear ratio, N_{ft}, and wheel radius, r.

$$F_e = \frac{T_{eff} \cdot N_{ft}}{r} \tag{7}$$

Modeling Braking Performance

The brake system will be represented by equating the force being applied to the brake pedal, by the driver, to the corresponding deceleration of the vehicle. This means of representing the braking ability of a vehicle is as a result of the work presented in (Mortimer, Segel et al. 1970). The objective of this study was to define the brake characteristics, within the space bounded by the relationship between brake pedal force and vehicle deceleration, which will lead to acceptable driver-vehicle performance. In essence this study was done to determine ergonomic properties for brake pedals that would give drivers the most effective control (Gillespie 1992). Therefore, using the results from this study, the DIV model will be able to not only account for the braking performance of the vehicle but also the manner in which the driver interacts with the system.

The results of the aforementioned study include several linear relationships which describe the force being applied to the brake pedal and the rate of deceleration of the vehicle. From these relationships, the DIV model will use the proportionality constant which provides optimal pedal force gain. This proportionality constant, 0.021 g/lb, corresponds to maximum deceleration rate through minimal pedal force. Using this proportionality constant, the

following formulation will be used in the DIV model to represent the brake system of a vehicle and the driver's interaction with that system.

$$F_b = d_{brk} \bullet p_f \bullet W \tag{8}$$

where:

F_b	=	Brake Force
D_{brk}	=	driver's desire to brake (0-1)
p_f	=	pedal-force gain coefficient

Aerodynamic drag is another force that serves to retard the motion of a motor vehicle. This force is dependent of on atmospheric conditions, the frontal area of the vehicle, A_{ft}, and the velocity at which the vehicle is traveling relative to the wind, V_r. The equation below further describes aerodynamic drag, (Wong 2001):

$$R_a = \frac{\rho}{2} C_D A_f V_r^{\,2} \tag{9}$$

where:

ρ	=	Mass density of the air (0.07651 lb/ft3- performance test condition)
C_D	=	Coefficient of aerodynamic resistance

The force due to gravity is mainly experienced when the vehicle is on an incline. This force may either retard the vehicle motion or allow the vehicle to accelerate. The effect that this force will have on the vehicle's motion is dependent on whether or not the vehicle is traveling up or down the incline. The force due to gravity that is acting on the vehicle is calculated by, (Gillespie 1992):

$$F_g = \pm W \sin\theta \cong W\theta \tag{10}$$

where:

θ	=	Grade of the incline in radians

Modeling Lateral Movement

The structure used to represent the movement of the DIV model in the X-Y plane was adapted from (Moret 2003) which included the formulation a kinematics and a dynamic framework to model a vehicle's motion in a two-dimensional space. The kinematics framework that was presented in the aforementioned manuscript was chosen for the DIV model for two primary reasons 1) all the pertinent dynamic properties of the vehicle were already accounted by other means in the DIV model and 2) the ease of use with accurate X-Y position representation.

At the base of the kinematics framework for the 2-D representation of vehicle motion is the treatment of the vehicle as a non-homonymic system, which is a system that does not guarantee return to it original position even if is original configuration is reached. Along with the non-homonymic treatment of the vehicle, come non-homonymic constraints which are related to the velocity of the vehicle and are held under the assumption that there is no slippage at the wheel during a turn. The assumption that there is no slippage at the wheels is predominantly applicable to instances of high speed corning as wheel slippage of low speeds is negligible. The general form of the non-holonomic constraint maybe represented as

$$\dot{x}\sin(\varphi) - \dot{y}\cos(\varphi) = 0 \tag{11}$$

Where *x'* and *y'* represent the velocities in the x and the y direction in the vehicle coordinate system and δ is the vehicle orientation with respect to the global X-Y coordinate system. See Figure-1 for an illustration of the coordinate system being used and also the definition of the various variables that will be used in the development of the DIV model.

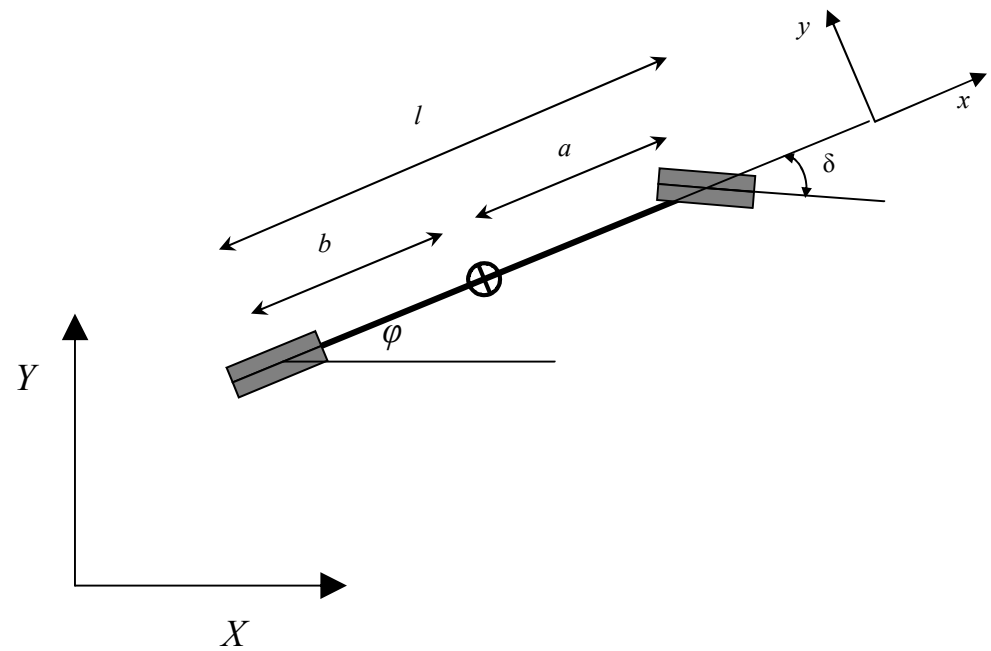

Figure 1 - Description of the coordinate system and other variables used

After a few more iteration of **Eq. 11** the velocity of the center of gravity with respect to the global coordinate system is defined as:

$$\dot{X} = \dot{x}\cos(\varphi) - \dot{y}\sin(\varphi)$$
$$\dot{Y} = \dot{x}\sin(\varphi) + \dot{y}\cos(\varphi) \qquad (12, 13)$$

Having **Eq. 12** and **Eq. 13** the global position of the vehicle can now be determined. However, before these equations can be used lateral velocity, *y'*, has to be first defined. The definition of the Ackerman angle, δ, also has to introduced as this is the parameter that is responsible for change the orientation of the vehicle.

$$\dot{y} = \dot{\varphi} b$$
$$\dot{\varphi} = \frac{\tan(\delta)}{l}\dot{x} \qquad (14, 15)$$

And

$$\delta = d_{turn} \cdot \frac{\pi \cdot lck\_lck}{S_{ratio}} \qquad (16)$$

where:

d_{turn}	-	Driver's desire to turn (-1 to +1)
lck_lck	-	# of steering wheel revolutions from one lock to the next
S_{ratio}	-	Steer ratio (ratio of radians dialed to Ackerman angle)

DIV MODEL CALIBRATION

One for the key design features of the DIV model is that it is meant to be easily and cheaply calibrated. The calibration process of the DIV model will entail the user providing the model with few performance specifications of the vehicle that the user is looking to model. These specifications will be assessable as they are available to the public via car manufactures and various organizations that offer tools to research a myriad vehicles, for example

Cars.com. The vehicle performance specifications that the DIV model requires include: aerodynamic resistance coefficient, engine displacement, gear ratios, steer ratio and the vehicle dimensions.

In addition to these specifications, the model also has a few variables relating to the environment that has an impact on vehicle motion. Such variables include wind speed and gradient of the roadway. Once the values of the vehicle performance specification and the various values describing the surrounding are inputted to the DIV model, it will be able to replicate the motion of the vehicle that these specifications belong to. The calibration process for the DIV model can afford to be this simple largely due to the fact that the DIV model represents the basic processes that contribute to the dynamic properties of an automobile.

DIV MODEL VALIDATION

In validating the DIV model, three standard performance tests were used to determine whether or not the DIV model is capable of successful replicating the movement of the vehicle it is attempting to represent. These tests that are typically conducted on vehicles to determine how well they accelerate, brake and handle. To test how well a vehicle accelerates, the time it takes for a vehicle to go from rest to 60 mph is recorded, as well as the time it takes a vehicle to cover a distance of ¼ mile. As for determining how well a vehicle brakes there are several standards set by the Federal Motor Carrier Safety Administration (FMCSA), dictating maximum allowable stopping distances from various speeds that all vehicle manufacturers must satisfy. And finally, in terms of attaching a measure to how well a vehicle handles, the diameter of the circle scribed by the vehicle's outer front wheel is recorded, after the maximum steering angle has been dialed.

Test Vehicles for Validating the DIV Model

To increase the applicability of the DIV model, passenger cars were chosen as they represent approximately 58% of the registered passenger vehicles in the United States in 2004 (BTS 2004). Therefore, upon successful validation of the DIV model, with the use of passenger cars, the DIV will be capable of representing the majority of vehicles of today's roadways. Three different types of passenger cars were used in this validation process – a sports car - 2006 Porsche Cayman S, a large passenger car - 2006 Ford Fusion Sedan SE, and a small passenger car - 2006 Honda Civic Coupe EX.

Validating Acceleration Performance

As previously mentioned, the tests chosen to verify how well the DIV model replicates the acceleration performance of an automobile are the 0 – 60 mph test and the ¼ mile test. In these tests, the driver of the vehicle opens the throttle body to its maximum position and the time it takes the vehicle to get from rest to 60 mph, as well as the time taken for the vehicle to cover a distance of a ¼ of a mile are recorded. The DIV model simulated these tests and its results were compared to the published results for the test vehicles for similar tests. Published results for the various tests were obtained from (Cars.com 2007). The comparisons of test results are presented in Table-1.

Table 1 - Comparison of Acceleration Performance DIV Model vs. Real Vehicles

Tests	**Porsche Cayman S**			**Ford Fusion**			**Honda Civic**		
	Obs.	*DIV*	*Error*	*Obs.*	*DIV*	*Error*	*Obs.*	*DIV*	*Error*
	(sec)	*(sec)*	*(%)*	*(sec)*	*(sec)*	*(%)*	*(sec)*	*(sec)*	*(%)*
0-60 mph	5.15	4.98	3.30	6.89	6.78	1.60	7.84	7.74	1.28
¼ mile	13.67	13.20	3.44	15.47	15.05	2.71	16.08	15.89	1.18

Evaluation of Test Results

The absolute percentage error between the observed results and that of the DIV model provides a means of quantitatively validating the DIV model. For the purpose of this paper, an absolute percentage no greater than 5% between the observed results and that of the DIV model represents a successful attempt by the DIV model in

replicating the acceleration performance of a *real* vehicle. As seen from the above table the absolute percentage error ranges from 1.60% to 3.44%, signifying that fact that the DIV model is successful at replicating an automobile's acceleration performance.

To further highlight the validity of the DIV model, as it attempts to represent the acceleration performance of an automobile, a diagonal plot was created. The diagonal plot provides a means of qualitatively evaluating how well the DIV model replicated acceleration performances. In essence, for the diagonal plot, if the simulated mechanism, in this case the DIV model, replicated the observed results then the ideal fit will be a 45º line. As seen from Figure-2 this is almost the case, once again proving the validity for the DIV model when replicating acceleration performance. (Ni, Leonard et al. 2004)

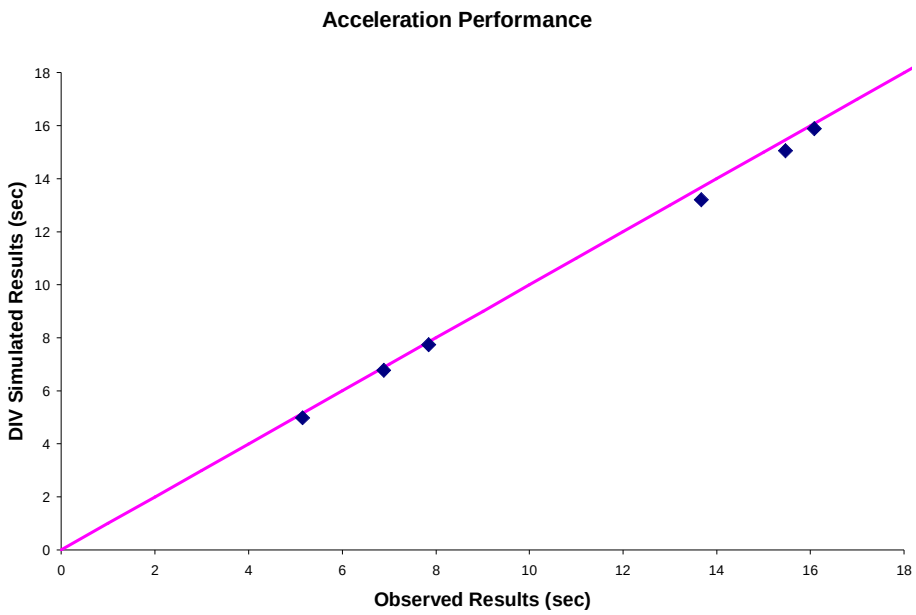

Figure 2 - Diagonal-Plot Observed vs. DIV Simulated Results

Validating Deceleration Performance

Due to the complexity of the braking system and the various ways of measuring braking performance the DIV model's braking capabilities will be validated by comparing its performance to the safety standards set by the Federal Motor Carrier Safety Administration. The braking performance of the DIV model will be compared to Part 571 of the Federal Motor Vehicle Safety Standards – Standard No. 105, which describes the requirements for Hydraulic and Electric Brake Systems. (FMCSA 2005)

There are four effectiveness tests geared toward ensuring safe braking performance.

Effectiveness Test #1 – a small vehicle / passenger car, with pre-burnished brakes, should be able to come to rest from 30 mph and 60 mph within 57 ft and 216 ft respectively.

Effectiveness Test #2 – a passenger car should be able to rest from speeds of 30 mph, 60 mph and 80 mph within 54 ft, 204 ft and 383 ft respectively.

Effectiveness Test #3 – a lightly loaded passenger car should be able to come to rest from 60 mph within 194 feet.

Effectiveness Test #4 – a passenger car should be able to come to rest from speeds of 30 mph, 60 mph, 80 mph and 100 mph within distances of 57 ft, 216 ft, 405 ft, and 673 ft respectively.

Point to note, when referring to brakes being pre-burnished and burnished, the DIV will not account for the difference between the two. This largely due to the complexity in treating the two stages how worn the brakes are and the lack added fidelity that would be gained upon including the effects of pre-burnished and burnished brakes. Another point of clarification is the definition of a lightly loaded vehicle. According the Federal Motor Carrier Safety Administration a lightly loaded vehicle is the unloaded vehicle weight plus 400 lbs.

Evaluation of Tests Results

In carrying out the various effectiveness tests on the DIV model, after the speed, specified by the test, is reached the driver applies maximum brake force. The stopping distance is then calculate based on the distance travel between the time the driver applies the brake to when the car comes to rest. In testing the braking performance of the DIV model, the four effectiveness tests were conducted on the DIV model's representation of the three test vehicles and the results tabulated. See Table 2.

Table 2 - Braking Performance Results and Analysis

Braking Performance	Standard No. 105	DIV	Error	Success
	Stop Dist. (ft)	Stop Dist. (ft)	%	
Porsche Cayman S				
Effectiveness Test #1	57	56.18	-1.4	✓
	216	200.04	-7.4	✓
Effectiveness Test #2	54	56.18	4.0	✓
	204	200.04	-1.9	✓
	383	325	-15.1	✓
Effectiveness Test #3	194	199.87	3.0	✓
Effectiveness Test #4	57	56.18	-1.4	✓
	216	200.04	-7.4	✓
	405	325	-19.8	✓
	673	469.72	-30.2	✓
Ford Fusion				
Effectiveness Test #1	57	61.67	8.2	✗
	216	212.83	-1.5	✓
Effectiveness Test #2	54	61.67	14.2	✗
	204	212.83	4.3	✓
	383	340.23	-11.2	✓
Effectiveness Test #3	194	212.97	9.8	✗
Effectiveness Test #4	57	61.67	8.2	✗
	216	212.83	-1.5	✓
	405	340.23	-16.0	✓
	673	483.35	-28.2	✓
Honda Civic				
Effectiveness Test #1	57	68.06	19.4	✗
	216	237.44	9.9	✗
Effectiveness Test #2	54	68.06	26.0	✗
	204	237.44	16.4	✗
	383	382.5	-0.1	✓
Effectiveness Test #3	194	233.77	20.5	✗
Effectiveness Test #4	57	68.06	19.4	✗
	216	237.44	9.9	✗
	405	382.5	-5.6	✓
	673	514.74	-23.5	✓

Before the test results can be evaluated to determine whether or not the DIV model was successful at replicating the braking performance of *real* vehicles, what the numbers in Standard No. 105 mean and *success* has to first be defined. The numbers of Standard No. 105 represents the maximum allowable distance that a vehicle should come

to rest in, given certain at set of initial conditions. These values are caps set by the Federal Motor Carrier Safety Administration to ensure that car makers produce braking systems that can bring cars to rest from a series of initial condition within a specified distance after the driver has applied the brake. Therefore, success for the DIV model, in terms of replicating braking performance, is had whenever the stopping distance produced by the DIV is less than or no greater than 5% of the stopping distance specified for a particular effectiveness test.

In Table-2, ☑s and ☒s are used to indicate whether or not the DIV model was successful at representing the braking performance of a particular vehicle during a specific effectiveness test. The ☑s represents a successful replication of the braking performance and the ☒s represents a failure. The above evaluation of the test results was a quantitative means of validating the model. For a qualitative evaluation of the model Figure-3 illustrates a diagonal plot of the Standard No. 105 caps versus the braking performance results from the DIV model.

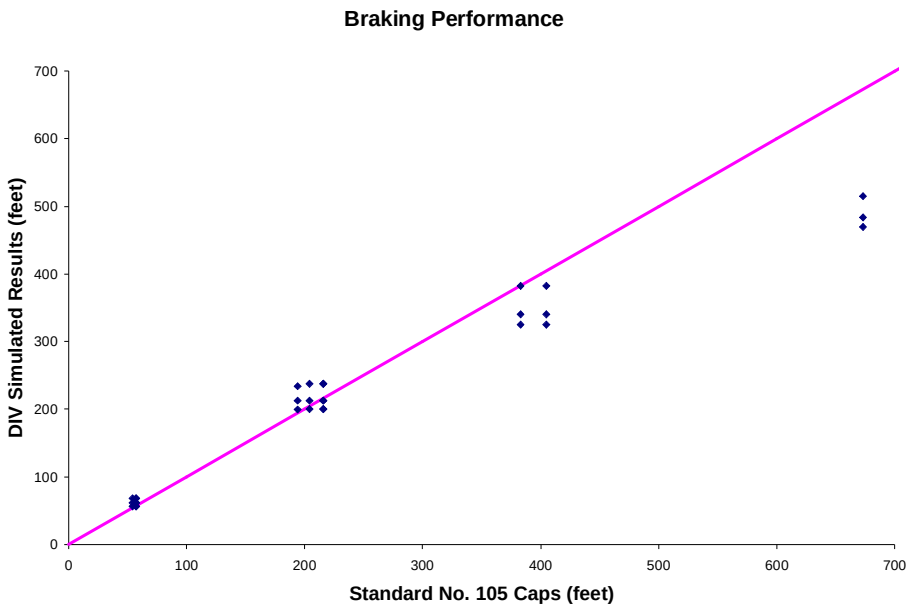

Figure 5 - Diagonal-Plot Observed vs. DIV Simulated Braking Distances

When analyzing the diagonal plot, what is important to note here is that DIV model's success at replicating a vehicle's braking performance is easier to deduce. This is so because the points reasonably above the 45º line represent instances where the DIV model failed to produce stopping distances that are within the standards set by the FMCSA, while points on or below the line represents instances of success.

Examining both the quantitative and qualitative evaluations of the test results of the DIV model's ability to replicate braking performance, one will notice that in light of the several instances where the DIV failed to produce stopping distances under or within reason of the aforementioned caps, the model still manage to perform relatively well. This level of success in the midst of the instances of failure provides motivation for additional research, to correct the problems that may lie in the DIV model.

Validating Lateral Movement

To illustrate this validation process the 2006 Porsche Cayman S lateral movement test is presented. The turning circle as defined by Road and Track Magazine is the circle that is scribed by the outside front tire when the maximum steering lock dialed in. The measurement that is often used to represent the turning circle of a given automobile is the diameter of that circle. According to the manufacture's specifications, the 2006 Porsche Cayman S, with a steering ratio of 15.5:1, scribes a turning circle with an approximate diameter of 36.4 feet.

Using all the specifications of the Porsche Cayman S in the DIV model and dialing the maximum steering lock the diameter of the turning circle is approximately 36.87 feet. The diameter outputted by the DIV model has to be converted from the diameter of the circle scribed by the center of gravity (28.15 ft.) to the diameter of the circle scribed by the outer front wheel of the vehicle. To do this, the distance from the center of gravity to the center of the front axle is calculated, doubled (8.72 ft.) and then added to the diameter of the circle scribed by the center of gravity.

Similar procedures were conducted to determine how well the DIV model is capable of reproducing the turning circles of the other two test vehicles. The DIV model's results of the diameters of the vehicles' turning circles that it attempted to mimic and the corresponding absolute percentage errors are presented in Table-3.

Table 3- Diameter of Turning Circle - DIV Model vs. Real Vehicles

Turning Circle	**Porsche Cayman S**			**Ford Fusion**			**Honda Civic**		
	Obs.	*DIV*	*Error*	*Obs.*	*DIV*	*Error*	*Obs.*	*DIV*	*Error*
	(sec)	*(sec)*	*(%)*	*(sec)*	*(sec)*	*(%)*	*(sec)*	*(sec)*	*(%)*
Diameter	36.4	36.87	1.29	40.0	42.19	5.47	35.4	33.77	4.60

In terms of validating this aspect of the DIV model, a successful representation of the lateral movement of an automobile is had when the absolute percentage error in the DIV model's output of the diameter of a turning circle is no greater 5% of the published corresponding value. Given the percentage error in the above table, it can be deduced that the DIV model demonstrate a rather high level of success when replicating a vehicle's lateral movement.

CONCLUSION

The DIV model is a simple and computationally affordable vehicle dynamics model. The DIV model demonstrated that it is capable of high-fidelity representation of vehicle motion. Upon validating the DIV model, it demonstrated that it is capable of replicating three sets of performance test results for three passenger cars. The three performance tests are meant to measure how well a vehicle accelerates, brakes and turns.

The acceleration test included a measure of how long it takes a vehicle to accelerate from rest to 60 mph and also how long it takes a vehicle to cover a distance of a quarter of a mile. The DIV model output times whose absolute percentage errors are no greater than 3.5% of the observed values of the three test vehicles - well within the 5% range criteria used to represent successful replication of acceleration performance.

As for the brake performance test, the DIV model output the required distances to bring a vehicle to rest from particular speeds. These values of stopping distances were compared to a series of standards set by the Federal Motor Carrier Safety Administration (FMCSA). In several instances the DIV model was able output distances within the limits set by the FMCSA; however there are few instances in which the DIV model's output exceeded the standards. The Porsche Cayman representation was the most successful while the Honda Civic representation was the least successful according to the criteria set forth by this thesis.

The performance test to determine how well the DIV model represents lateral movement, the diameter of the circle scribed by the outer front wheel of the vehicle, after the maximum steering angle was dialed, was measured. The absolute percentage error between the DIV model's results and those observed for the diameters of the various passenger cars were no more the 5.5%.

The DIV model demonstrated that it is able to realistically replicate the two-dimensional movement of an automobile, while allowing both the driver and the environment to influence its motion. What is also noteworthy

here about the DIV model is that it accomplished all the goals with simple mathematical representations of the processes that are responsible for vehicle movement and with very little computational costs.

FUTURE DIRECTIONS

Future directions of this research effort are focused on three specific areas: the mathematical representation of a vehicle's braking system, the comprehensive treatment of the lateral movement of a vehicle and the validation procedure in determine how well the vehicle model is capable of represent vehicle dynamics.

The Braking System: Currently, the DIV model treats the brake system of a vehicle according to the result from a Driver-Vehicle Braking Performance study conducted in 1970 (Mortimer, Segel et al. 1970). Not only is this study due an update – given the advances in the brake technology used on today's vehicles, but also this treatment of the braking system is not vehicle specific. The next step for DIV model in representing the brake system of a vehicle is to utilize simplified mathematical representations of how a driver places his/her foot on the brake pedal and produce a force on the wheels to retard the motion of a vehicle.

Lateral Vehicle Movement: The DIV does a relatively good job of representing the lateral movement of a vehicle but at low speeds. When representing a vehicle's lateral motion the DIV model does not account for slippage at the tire-road interface. At low speeds slippage is negligible and as result does not affect the motion of a vehicle. But at a high speeds, tremendous slippage can occur and as a results greatly influences the motion of a vehicle. Therefore the next step for the DIV model is to account for slippage at the tire-road interface.

Validation Procedure: The current performance tests involved in the validation procedure only test performances with maximum driver input, i.e. maximum gas and brake pedal displacement and maximum steering angle. The next step would be to conduct performance test on both a real vehicle and the DIV model where maximum driver input is rare.

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