

A Cost-Effective Alternative to Moving Floor Wind Tunnels in Order to Calculate Rolling Resistance and Aerodynamic Drag Coefficients

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ABSTRACT

This study investigates the practicality of vehicle coast down testing as a suitable replacement to moving floor wind tunnel experimentation. The recent implementation of full-scale moving floor wind tunnels is forcing a re-estimation of previous coefficient of drag determinations. Moreover, these wind tunnels are relatively expensive to build and operate and may not capture concepts such as linear and quadratic velocity dependency along with the influence of tire pressure on rolling resistance. As a result, the method elucidated here improves the accuracy of the fundamental vehicle modeling equations while remaining relatively affordable. The trends produced by incorporating on road test data into the model fit the values indicated by laboratory tests. This research chose equipment based on a balance between affordability and accuracy while illustrating that higher resolution frequency equipment would further enhance the model accuracy.

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INTRODUCTION

Many forces act on a vehicle during operation with its total tractive force (F_{TE}) a sum of the linear acceleration (F_{LA}), aerodynamic drag (F_D), rolling resistance (F_{RR}) and road gradation forces (F_{GR}):

$$F_{TE} = F_{LA} + F_D + F_{RR} + F_{GR} \quad (1)$$

where linear acceleration acts to propel the vehicle forward and the other forces work in opposition to vehicle motion. In this [Eqn. 1](#), the two key forces of interest influenced by vehicle design and associated components are the aerodynamic drag and rolling resistance. One method for measuring these forces for a particular automobile is to record vehicle velocity verses time data during a coast down experiment [1, 2, 3, 4, 5].

A coast down test is an experiment in which an operator drives a vehicle at a high speed and then coasts to a near stop while the drive components are disengaged. Since these drive components are uncoupled, it is reasonable to neglect the linear acceleration force. Moreover, researchers often neglect the efficiency of these elements along with losses such as transmission friction and brake drag due to their relatively small magnitudes and measurement difficulty [2]. Furthermore, drive

component drag forces are vehicle dependent and require measurement for each test vehicle. While estimation of these items could occur, the amount of induced error associated with such approximations is difficult to quantify.

In the absence of a perfectly horizontal road, the analysis must consider elevation through the road gradation force. In addition, wind direction and magnitude can have an impact on the coast down experiment through the drag force. While its effect is minimal at lower speeds, it can become a significant factor at higher speeds (e.g., highway driving) due to the quadratic velocity component dependency in the drag force, discussed later. Ideally, measurement of vehicle elevation and wind velocity occurs at the same intervals as the velocity and time of the vehicle under study.

Of importance for this paper, vehicular modeling efforts often neglect concepts such as linear and quadratic velocity dependency, the influence of tire pressure on rolling resistance along with a dynamic coefficient of aerodynamic drag in the force balance [6]. While some researchers do incorporate these effects [2, 4, 5, 7], they often do not include all of previously mentioned concepts at the same time; e.g., enhanced rolling resistance but simple coefficient of drag. Another concern with these models is the lack of validation. While the models may perform well in a laboratory

environment, the lack of true road testing hinders the application of these models for real world scenarios. Achieving the desired “on road” values in a controlled setting requires the use of a large moving floor wind tunnel. For example, Krajnović and Davidson find that the motion of the floor reduces aerodynamic drag by 8% for a full scale automobile [8]. Geropp and Odenthal arrive at similar results using a one-fifth scale two-dimensional body [9] and Fago discovers that road simulation via moving floor wind tunnels reduces the aerodynamic drag coefficient by as much as 40% at various velocities in comparison to a stationary floor [10]. A significant topic of discussion in this field is the effect of rotating wheels on overall aerodynamic drag of the vehicle [11, 12, 13, 14, 15]. While this paper focuses on the vehicle as a whole, the procedure outlined here may prove useful for further experimentation involving changes in the rotating wheel assembly of the vehicle.

Full size moving floor wind tunnels are relatively uncommon and testing is expensive. Moreover, these testing methods can prove to be cost prohibitive for smaller automotive manufacturers, college and experimental car teams such as those that compete in Formula SAE and solar challenge events. Even stationary floor wind tunnel testing can be too expensive for some situations. In comparison, the primary expense of a coast down test is a function of the data acquisition equipment. Unlike the cost of a wind tunnel where companies charge based on an hourly rate, once the researcher purchases the needed data acquisition equipment, they can perform numerous coast down tests for multiple vehicles or testing environments with minimal additional expense. Given the advancement in vehicle data logging technology, Global Positioning Systems (GPS), and the addition of the On-Board Diagnostic (OBD) ports, commercially available data logging and acquisition equipment has become relatively inexpensive and available. The data collection equipment for this analysis was purchased for less than the cost of two hours of local stationary wind tunnel testing (less than 500 USD). It is worth noting that a full-scale wind tunnel is capable of providing aerodynamic data beyond that of a coast-down test and may be required for a more thorough aerodynamic analysis. In the following sections, this paper first describes all pertinent model parameters followed by its calibration via vehicle data logging equipment and a coast down test.

MODEL AND DATA COLLECTION

In order to understand vehicle dynamics, the basic fundamental methods are first considered. Then, the authors improve upon these methods to determine the data set required from testing. The following sections present the proposed model and data collection methods.

Theoretical Vehicle Dynamics Model

The first item in the force balance of Eqn. (1) to consider during a coast down test is the drag force on a moving automobile. A common approach to the fundamental equation to model aerodynamic drag utilizes a constant drag coefficient (C_D) if sufficient experiments do not exist to determine the overall functionality of the parameter [16]:

$$F_D = \frac{1}{2} \rho A_f C_D V^2 \quad (2)$$

where ρ is the density of the medium, A_f is the frontal area and V is the velocity of the vehicle.

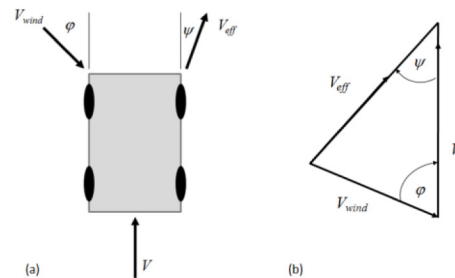


Figure 1. Diagram of influence of wind speed (a) upon relative velocity of the vehicle and (b) the resulting velocity triangle.

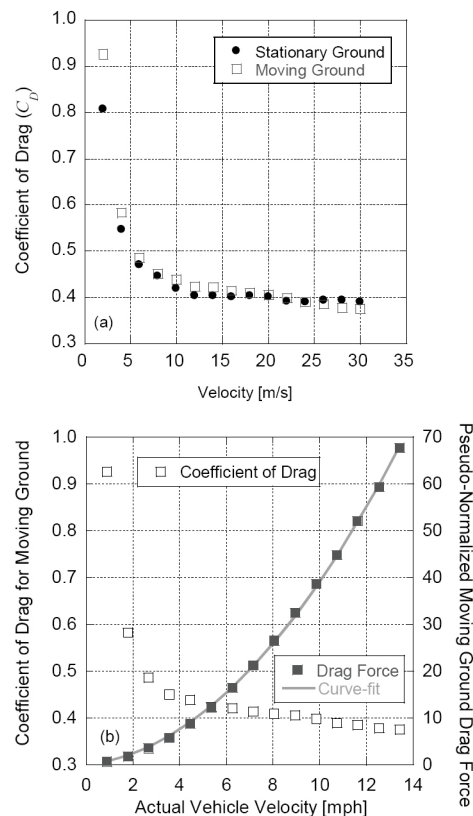


Figure 2. Geropp and Odenthal data of (a) coefficient of drag for stationary and moving grounds [9] and (b) normalization by the authors in order to determine C_D functionality.

While this generally accepted equation works well for a principal understanding of vehicle dynamics, it neglects key aspects necessary for a high-level model. In a real world setting, unlike a wind tunnel that simulates vehicle motion in a stagnant environment, wind and pitch of the vehicle can act to hinder or help the resultant drag force.

As illustrated in [Figure 1](#), the effective velocity of a vehicle (V_{eff}) is a function of its actual velocity (V), the wind speed (V_{wind}), the angle of wind relative to the direction of motion (ϕ) and the yaw (ψ) of the vehicle:

$$V_{eff} \cos \psi = V - V_{wind} \cos \phi \quad (3)$$

Note that the wind speed is positive if assisting and negative if hindering as the figure shows. Proper consideration of the yaw angle must account for the direction of the wind effects. As a result, measurement of the yaw angle is according to the relative angle of the wind in reference to the heading of the vehicle.

In addition, the fundamental coefficient of drag does not necessarily have a singular value. It is a function of the Reynolds, Froude, and Mach numbers describing velocity, free-surface and compressibility effects, respectively [16]. This is evident in the work of Geropp and Odenthal who endeavor to find the coefficient of drag in [Figure 2a](#) as a function of stationary and moving ground [9]. Geropp and Odenthal find that the effect of a moving ground slightly reduces the coefficient of drag. This variance is generally less than five percent but is more evident at lower velocities.

Combining the density and frontal area components (since Geropp and Odenthal omitted these values from their paper), creates a pseudo-normalized drag force that provides further investigation of the coefficient of drag:

$$F'_D = \frac{F_D}{0.5 \rho A_f} = C_D V^2 \quad (4)$$

From this equation, fitting the data of Geropp and Odenthal in [Figure 2b](#) to a second-order polynomial curve-fit, while incorporating the $1/5^{\text{th}}$ scaling of their model finds the following:

$$F'_D = 0.3384V^2 + 0.5299V \quad (5)$$

with an R^2 value of 0.9997 between the curve-fit and data. Since the basic fundamental derivation of the drag force requires that it must have a squared dependency on velocity, this result indicates that the coefficient of drag can include an additional velocity component for increased accuracy.

Another factor influencing vehicle drag is the yaw drag that can have varying effects on the effective coefficient of drag depending on the body design of the vehicle:

$$C_{D,yaw} = k_D (1 - \cos 6\psi) \quad (6)$$

where k_D , the yaw drag coefficient, ranges from 0.02 for well streamlined vehicles to 0.053 for less streamlined sedans with sharp corners and square front ends [17].

As a result, a complete model of the coefficient of drag while incorporating wind effects is:

$$C_D = a_D + \frac{b_D}{V_{eff} \cos \psi} + k_D (1 - \cos 6\psi) \quad (7)$$

and the resultant drag force becomes:

$$F_D = \frac{1}{2} \rho A_f \left[a_D + \frac{b_D}{V_{eff} \cos \psi} + k_D (1 - \cos 6\psi) \right] (V_{eff} \cos \psi)^2 \quad (8)$$

with the traditional (zero yaw angle) value of C_D present in the form of a_D , the increased dependency on velocity included in b_D and the yaw effect of drag captured in the k_D term.

In addition to drag, the body of a vehicle can provide a lift force as a function of the planform area (A_p) of the vehicle:

$$F_L = \frac{1}{2} \rho A_p C_L (V_{eff} \cos \psi)^2 \quad (9)$$

where C_L is the coefficient of lift. However, the planform area is difficult to measure for a vehicle; hence, typical convention is to use the frontal area (A_f) for both the drag and lift forces [18]. The lift of a vehicle can effectively change the dynamic weight of the vehicle as follows:

$$Z = mg \cos(\theta) - F_L \quad (10)$$

This will factor into the rolling resistance and road gradation forces instead of being included directly in the force balance of [Eqn. \(1\)](#). Note that the slope of the road (θ) is included in the effective weight of the vehicle in order to provide the correct normal force to the road.

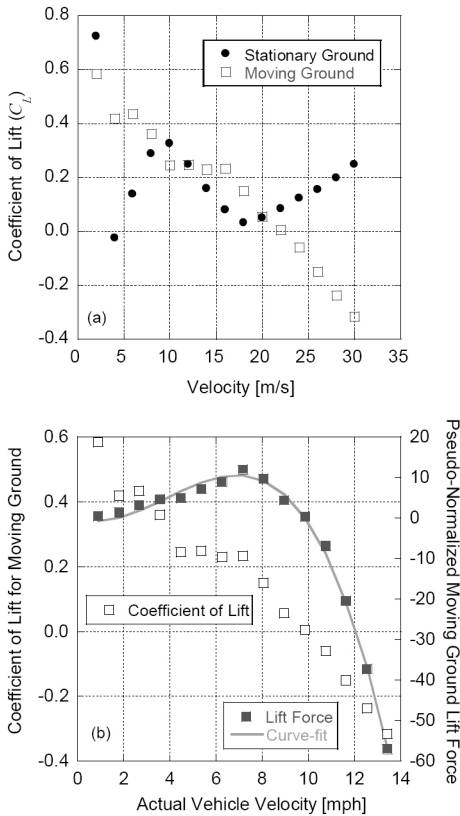


Figure 3. Geropp and Odenthal data of (a) coefficient of lift for stationary and moving grounds [9] and (b) normalization by the authors in order to determine C_L functionality.

In the same work of Geropp and Odenthal [9], they measure the lift force on a scale vehicle as a function of stationary and moving ground in Figure 3a. Unlike drag force where the moving ground has a small effect, for lift, the influence of a moving ground significantly changes the trend with respect to velocity. While not explicitly stated in their paper, the authors infer that this is a result of a large increase in the pressure forces on the underbody of the vehicle. Following the earlier example, a pseudo-normalized lift force provides investigation into the coefficient of lift:

$$F'_L = \frac{F_L}{0.5\rho A_p} = C_L V^2 \quad (11)$$

From this equation, fitting the data of Geropp and Odenthal in Figure 3b to a third-order polynomial curve-fit, while incorporating the $1/5^{\text{th}}$ scaling of their model finds the following:

$$F'_L = -0.1016V_{eff}^3 + 1.1813V_{eff}^2 - 1.7851V_{eff} \quad (12)$$

with an R^2 value of 0.9961 between the curve-fit and data. Since the basic fundamental derivation of the lift force requires a squared dependency on velocity, this result indicates that the coefficient of lift can include two velocity components for increased accuracy.

Another factor on the lift force of the vehicle is the increase in lift forces due to the yaw moment:

$$C_{L,yaw} = k_L \sin^2 3\psi \quad (13)$$

where k_L ranges from 0.25 for well streamlined vehicles to 0.52 for non-streamlined sedans [17].

As a result, a complete model of the coefficient of lift while incorporating wind influence equals:

$$C_L = a_L (V_{eff} \cos \psi) + b_L + \frac{c_L}{V_{eff} \cos \psi} + k_L \sin^2 3\psi \quad (14)$$

and the resultant lift force becomes:

$$F_L = \frac{1}{2} \rho A_p \left[a_L (V_{eff} \cos \psi) + b_L + \frac{c_L}{V_{eff} \cos \psi} + k_L \sin^2 3\psi \right] (V_{eff} \cos \psi)^2 \quad (15)$$

with the traditional value of C_L present in the form of b_L .

Modeling the interaction between vehicle tires and driving surfaces requires consideration of the rolling resistance force. Often, research represents this force through a constant coefficient of rolling resistance (μ_{RR}) multiplied by the vehicle weight:

$$F_{RR} = \mu_{RR} mg \quad (16)$$

This model again works well for a fundamental understanding of rolling resistance; however, it lacks additional considerations due to velocity and tire pressure. As velocity increases, the contact area of the tire decreases due to centrifugal effects. The tire diameter increases, effectively narrowing the tire and reducing the area of contact between the tire and the road. The same concept holds true for an increase in tire pressure. As the tire pressure increases, the tire diameter increases reducing the contact area and the overall rolling resistance of the vehicle.

Taking into account these influences, Grover proposed a model of rolling resistance that includes load, tire pressure (p_{tire}) and speed. While Grover made no mention of units, the only means for preventing a unit imbalance is to provide a reference pressure (p_{ref}) of one kilopascal and reference vertical load (Z_{ref}) of one newton [7]:

$$F_{RR} = \left(\frac{p_{tire}}{p_{ref}} \right)^\alpha \left(\frac{Z}{Z_{ref}} \right)^\beta (a_{rr} + b_{rr}V + d_{rr}V^2) \quad (17)$$