

Application of Air Brake Performance Relationships in Accident Reconstruction and Their Correlation to Real Vehicle Performance

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ABSTRACT

This research paper builds onto the wealth of technical information that has been published in the past by engineers such as Flick, Radlinski, and Heusser. For this paper, the pushrod force versus chamber pressure data published by Heusser are supplemented with data taken from brake chamber types not reported on by Heusser in 1991. The utility of Heusser's braking force relationships is explored and discussed. Finally, a straightforward and robust method for calculating truck braking performance, based on the brake stroke measurements and published heavy truck braking test results, is introduced and compared to full-scale vehicle test data.

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INTRODUCTION

Commercial vehicle accident reconstruction often requires that the reconstructionist use the traditional "slide-to-stop" relationship to calculate pre-braking and pre-impact speeds for a heavy commercial vehicle. Many reconstructionists choose to implement Heusser's modified brake sizing equation introduced by Heusser in 1991 [1] as a means to calculate the net drag factor, or deceleration rate, for an air-braked vehicle with brakes that are inoperable or out of adjustment. Mr. Heusser's equation is presented as Equation (1). Although the basis for this methodology is sound, the implementation of the methodology can have pitfalls, in the form of assumptions or estimates that are often made to implement the model (in lieu of measurements). Some primary assumptions or estimates include the coefficient of friction of the interface between the brake linings and drum, as well as the coefficient of friction between the tires and road. Another important assumption is that the tires are carrying the load equally distributed, and the static distribution does not change during the dynamics of the

braking event. Figure 1 shows the measured longitudinal force - versus the longitudinal slip ratio - for a heavy truck steer axle tire tested on dry pavement. The force data in Figure 1 have been normalized by the tire's vertical load. Brake force models based on the modified brake sizing equation must be coupled with a tire model to calculate vehicle deceleration. In the case where brakes are experiencing lockup and/or the antilock brakes are cycling, finding the appropriate operating region for all of the vehicle tires becomes challenging, to say the least. This is a critical basic challenge, as the tire braking force is a direct non-linear function of the longitudinal slip, which is often being controlled by the antilock brake system (ABS) [2, 3]. This paper presents an alternative to methodologies based on Heusser's essential paper, while drawing from the works presented by Heusser, Flick, Radlinski, and others.

$$BForce = \left[\frac{2 * PRForce * SLength * LinF * DRad}{CamRad * TireRad} \right] * 0.6 \quad (1)$$

where:

PRForce = chamber pushrod force (lb)

SLength = slack adjuster length (in)

LinF = brake lining friction (unitless)

DRad = brake drum radius (in)

CamRad = S-Cam effective radius (in)

TireRad = tire rolling radius (in)

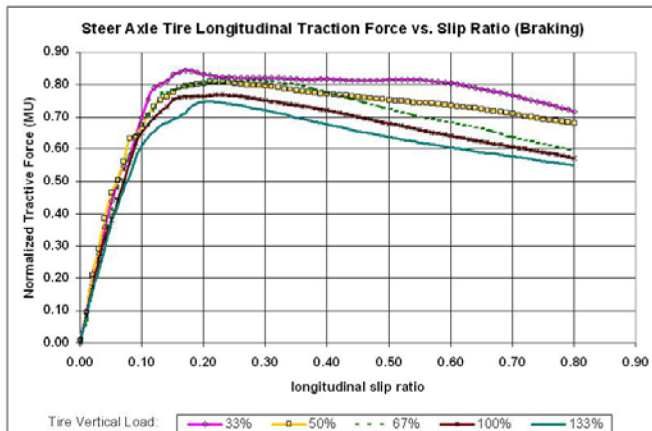


Figure 1. Normalized Longitudinal Braking Force versus Longitudinal Braking Slip Ratio for a Steer Axle Tire

FUNDAMENTALS OF BRAKING AND DECELERATION

Before presentation of the proposed methodology, the fundamental concepts are established using basic physics equations and showing some real-time test data. Equation (2) (Newton's Second Law of Motion) shows a simplified relationship between the sum of the tire braking forces (F_i) and the vehicle's deceleration, or drag factor (a):

$$\sum F_i = ma \quad (2)$$

where:

F_i = longitudinal force for each hub end, i

m = total vehicle mass

a = vehicle acceleration

Equation (2) shows a direct linear relationship between the sum of the tire forces and the vehicle deceleration performance. This is important because the proposed method uses a linear scaling of the air-braked vehicle's performance, based on the condition of the brakes on that vehicle.

Publications including [5, 6] have shown that a commercial vehicle's rate of deceleration is generally constant during a full-application braking event. This relationship could become nonlinear as the brakes begin to experience brake fade. However, significant brake fade will not be evident during most heavy panic stops on wet or dry pavement. The top panel of Figure 2 shows the application pressure (green traces similar to step inputs, labeled "Pc") and equivalent road speed (blue traces, labeled "V") for a series of S-cam brake tests on an inertial dynamometer. The bottom panel of Figure 2 shows the relative consistency of braking torque versus time (for the same series of tests) during the stop. Figure 3 shows tractor speed versus time for a full-application braking event for a heavy truck loaded to a gross combination weight (GCW) of 80,000 pounds (36,287 kg), stopping from 60 miles per hour (96.6 kph), with four foundation brakes (two drive and two semitrailer) disabled. Figure 4 shows the same type of data, but for the same heavy truck stopping from 60 miles per hour (96.6 kph), with two drive axle foundation brakes disabled. The data reinforce the theory that using the constant deceleration rates (drag factors) is appropriate and applicable, even when some brakes on the vehicle are not fully functional, and could therefore be placing an excessive load onto the brakes that are functional.

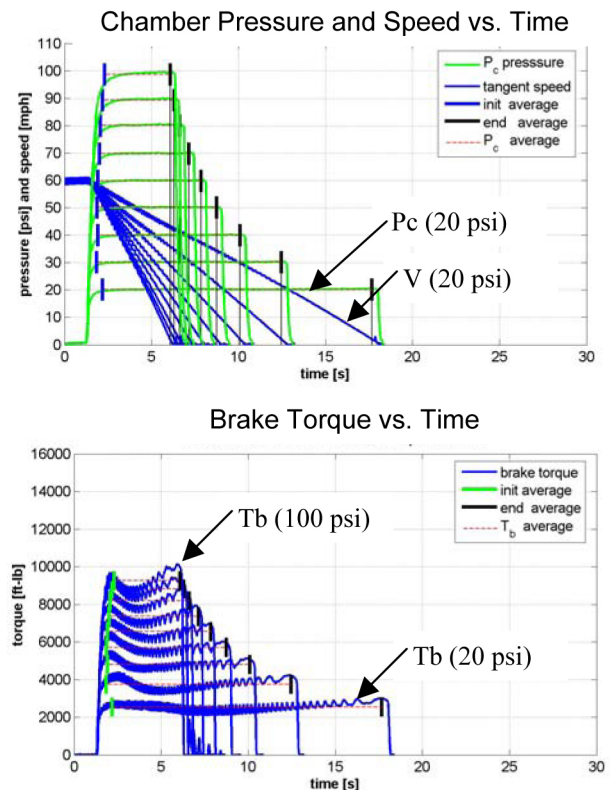


Figure 2. Brake dynamometer data showing a series of brake applications from 20 psi (0.138 MPa) to 100 psi (0.689 MPa), from an initial braking speed of 60 mph (96.6 kph)

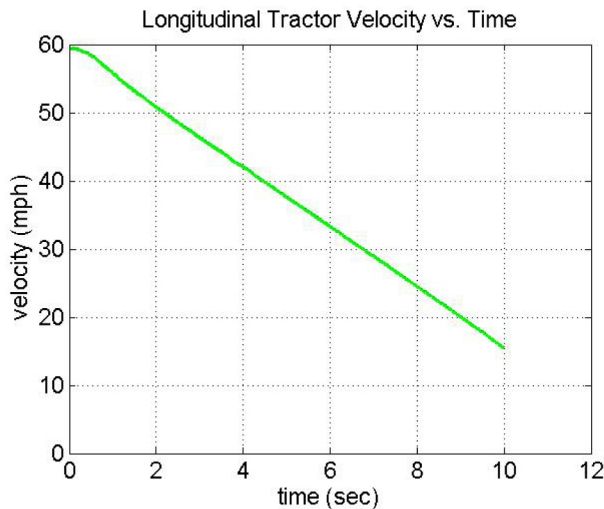


Figure 3. Tractor Speed versus time for a full braking event from 60 mph (96.6 kph) to rest. The data beyond 10 seconds was not recorded for this test. Four foundation brakes were disabled for this test.

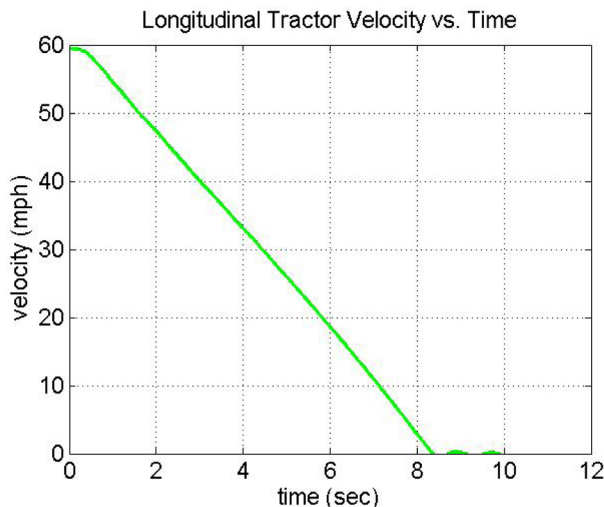


Figure 4. Tractor Speed versus time for a full braking event from 60 mph (96.6 kph) to rest. Two drive axle brakes were disabled.

BRIEF OVERVIEW OF HEUSSER MODEL AND EQUATIONS

Heusser's model was based directly on heavy truck brake dynamometer data reported by Flick in [4]. In his work, Flick fit a fourth-order polynomial to the brake torque data for two different brake configurations. The configurations modeled by Flick included constants for a typical steer axle brake [15 inch (381 mm) diameter by 4 inch (102 mm) wide drum, using a 20 square inch (129 cm²) standard stroke chamber coupled to a 5.5 inch (140 mm) long slack adjuster] and for a typical drive axle brake [16.5 inch (419 mm) diameter by 7

inch (178 mm) wide drum, using a 30 square inch (193.5 cm²) standard stroke chamber coupled to a 6.0 inch (152 mm) long slack adjuster]. Equation (3) below presents the fourth-order equation that Flick related to each of the two brake configurations modeled. The coefficients for the polynomials are given in Table 1. The output for Flick's models is shown in Figure 5, after the outputs in inch-pounds were converted to foot-pounds. These polynomial curve fits are valid from a stroke of 1.5 inches (38 mm) to 2.5 inches (64 mm) and form a foundation for the analyses presented herein.

$$T = a + b*s + c*s^2 + d*s^3 + e*s^4 \quad (3)$$

where:

T = [inch-pounds] brake torque

s = [inches] brake stroke

a,b,c,d,e = coefficients derived for each brake configuration

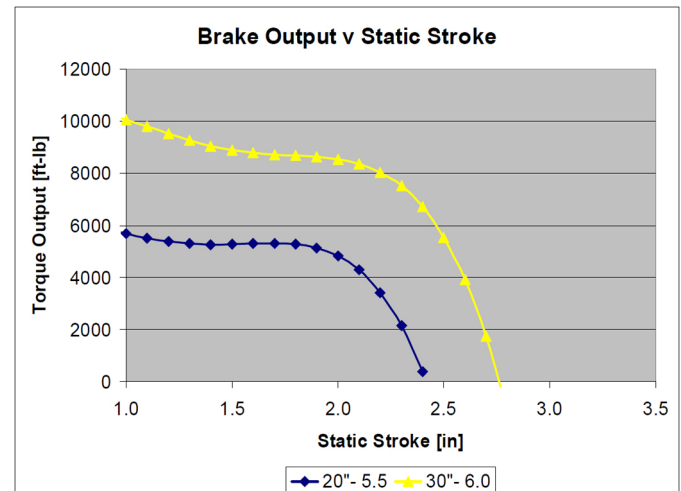


Figure 5. Torque output from Flick's equations, after being converted to foot-pounds.

Heusser continues in his paper [1] to define his model for brake force output, as a function of shoe leverage, slack adjuster length, chamber pushrod force, brake lining-to-drum friction, drum inner radius, S-cam radius, and tire rolling radius (see Equation 5 in [1]). After comparing the output of his new equation (Equation 4 in [1]) to the regression equation (Equation 1 in [1]), and conversion from brake torque to maximum attainable braking force), Heusser chose a correction factor of 0.60 to adjust the magnitude of his equation to that of Flick's 4th-order polynomial (Equation 1 in [1]). Although the Heusser paper does not go into detailed discussion into the root causes of the discrepancies between his theoretical equation and Flick's brake dynamometer data, the origins of the 0.60 correction factor are discussed in an SAE publication authored by John W. Kourik [10]. Discrepancies can result from many causes, the primary

Table 1. Coefficients for 4th-order polynomials as published by Flick [4].

	polynomial coefficients				
chamber – slack length	a	b	c	d	e
20 in ² – 5.5 in	972.3	308,494.1	444,318.8	253,803.0	50,824.2
30 in ² – 6.0 in	3,051.9	439,201.5	539,263.8	263,331.8	45,535.1

being the unknowns of the true lining-drum friction and other parameters and/or effects not modeled in Heusser's equation.

Heusser's work also presents data tables for pushrod force (versus air pressure and stroke) that can be directly utilized in his equation. This basic relationship of chamber pushrod force to stroke holds true (as do the brake torque polynomials in Equation (3)) for scaling overall vehicle brake performance as a function of brake stroke. Figure 6 is a chart that compares the normalized output for the 4th-order polynomial derived from Flick's dynamometer data (30-square inch (193.5 cm²) standard stroke chamber coupled to a 6.0 inch (152 mm) long slack adjuster) and pushrod force for a 30-square inch (193.5 cm²) standard stroke chamber published by Heusser. The outputs were normalized as a means to avoid comparing torques and forces. If all other variables in Heusser's Equation 5 are kept constant (and only pushrod force varied), the net effects are the same from both models. The outputs were normalized to the same basic operating point of 1.75 inches (44.5 mm) of chamber stroke (vertical line on the chart, labeled "Ref. Point." The basis for that choice was the fact that for the chamber size being analyzed, the operating point is sufficiently below the legal stroke limit for that chamber (2.00 inches or 50.8 mm), as required by the Commercial Vehicle Safety Alliance (CVSA) [8]. The lower range of the data regarding the subject chamber was also limited to 1.0 inch (25.4 mm) because of anomalous behavior of the 4th-order polynomial beyond the limits. Some interesting comparisons can be drawn from the data shown in Figure 6. The polynomial output, based on Flick's dynamometer data, swings abruptly approaching 120% of output, at the lower stroke levels. This is counter-intuitive, based on Heusser's chamber pushrod force data and the knowledge that the geometries of the chamber pushrod-slack adjuster angle are not consistent with this behavior. Another artifact is the difference in how the normalized output levels drop off beyond the stroke value of 2.00 inches (50.8 mm). Note that the polynomial model was based on data that ranged from 1.5 inches (38 mm) to 2.5 inches (64 mm) of stroke. The discrepancy in the region between 1.8 and 2.3 inches (45.7 to 58.4 mm) of stroke is probably negligible for

most analyses. However, the region beyond 2.3 inches (45.7 mm) is important, as it can be a significant source of technical disagreement and also have a significant impact on the braking force analyses for the subject vehicle. Although beyond the legal adjustment limit, as well as the intended range of operation for the subject brake chamber, brake strokes for similar chambers can be found in this range when in the field. Furthermore, the contribution of the subject out-of-adjustment brake might be key to competing analyses of the same vehicle. Another trace is added to the data in Figure 6, which re-plots the polynomial fit to Flick's dynamometer data, with the modification being that no output under the normalization value is allowed to be over 100%. This modification was implemented for simplicity, and the fact that it agrees with chamber pushrod force data.

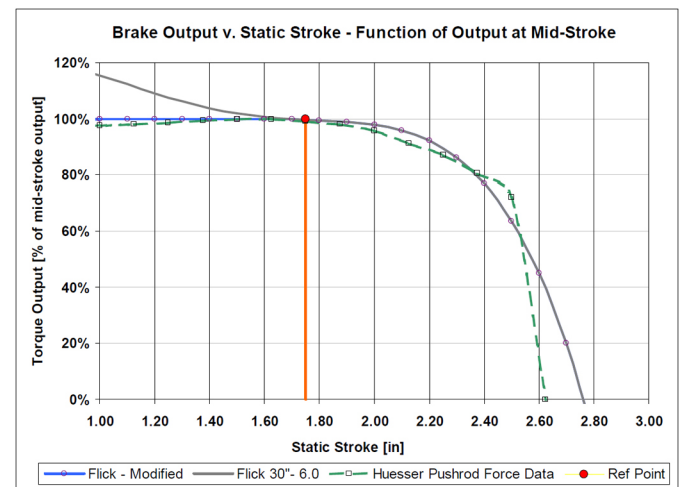


Figure 6. Chart comparing the normalized output from dynamometer data and pushrod force data

Finally, the reader is cautioned about explicitly interpreting very small differences in stroke that occur beyond the critical point where the pushrod force begins to decline sharply. This point occurs in the region of 2.5 to 2.7 inches (63.5 to 68.5 mm) in Figure 6, where the slope of the relationship sharply approaches vertical. Note however, that

the brake torque data presented by Flick in [4] does not drop off vertically as the slack adjustment approaches the limit. Flick's polynomial relationship also reflects this behavior.

METHODOLOGY

Using the data provided by Flick (brake dynamometer data) and Heusser (chamber pushrod data and the polynomial models of Flick's data), an accurate and dependable model can be used to predict heavy vehicle braking performance, based on the brake strokes measured for the subject vehicle in the field. The method presented here uses the polynomial model because it is based on actual dynamometer data and provides for a higher-fidelity interpretation of the brake output. The method is applied to each brake for which a stroke measurement can be obtained, then the sum of the brake force is compared to that calculated for the same vehicle with all brakes in their nominal operating range. From the two calculations, a ratio is reached, which can be applied to a likely range of drag factors for that vehicle. The relationship between force and acceleration demonstrated in Equation (2) prove this to be a valid operation, resulting in a revised drag factor that can be applied to a subject vehicle that has out-of-adjustment or inoperable air brakes.

BASES FOR DRAG FACTORS FOR IN-SERVICE TRUCKS

Choosing a drag factor that is representative of the subject truck is done using published braking data for single trucks and tractor-semitrailers. For a typical 5-axle tractor-semitrailer, the basis data were summarized by Dunn et al in [6]. In that work, the authors published two tables that summarized the available data on like vehicles. Those data are reproduced here as Table 3 and Table 4 in Appendix I. Note that the data presented cover the two loading extremes of unloaded and gross combination weight (GCW). Dunn et al [6] also published data for a combination tractor-semitrailer loaded to 1/2 of its useful payload. In the same work, data are presented for the same truck stopping from 60 mph (96.6 kph) and 30 mph (48.3 kph). Other publications can be referenced that contain testing results for school buses [12], straight trucks [7], and bobtail truck-tractors [14].

Taking an example from Table 3, entries from rows 1 through 6 are considered because they were generated from tests using similar vehicles and initial braking speeds. From those, the average drag factor for the six entries is 0.438. The high and low values are 0.494 and 0.366, respectively, giving a range of 0.128 or 29.2% of the mean. The range of appropriate typical values can be cut down to 14.6% of the mean if the 95% confidence limit is used on the six values chosen. The confidence limits are calculated using Equation (4), and from them the high and low values become 0.406 and 0.470. The same methodology can be used to obtain an appropriate performance range for other vehicles types, initial braking speeds, and/or loading conditions.

$$CL = \pm z_{\alpha/2} \left(\frac{S}{\sqrt{n}} \right) \quad (4)$$

where:

CL = confidence limit

$z_{\alpha/2}$ = area under the normal curve

($z_{\alpha/2} = 1.96$ corresponds to a 95 percent CL)

S = sample standard deviation

n = number of samples.

CALCULATION OF THE PERFORMANCE RATIO

A ratio by which the subject commercial vehicle's braking performance is adjusted is calculated using the polynomial brake equations derived by Flick, and a brake stroke representative of the typical in-service norm (Equation (5)). As discussed above, a "normal" brake stroke of 1.75 inches (44.5 mm) would be used for standard stroke 30 square inch (193.5 cm²) chamber. Using similar logic, a "normal" adjustment value of 1.40 inches (35.6 mm) could be used for a standard stroke 20-square inch (129 cm²) chamber. It is important to realize that the "normal" stroke adjustment value is not critical, as it corresponds to the flat part of the output-versus-stroke curves as presented in Figure 5 and Figure 6. A nominal (or "normal") level of total braking force available to the vehicle is calculated by summing the brake forces for all hub ends, given that every brake is capable of producing 100% of that modeled by the polynomial. The polynomials are conveniently derived for brakes tested with chamber and slack adjuster sizes routinely found on the steer and drive axles of over-the-road combination vehicles.

$$\bar{T}_i = \frac{1}{12} [a + b * \bar{s}_i + c * \bar{s}_i^2 + d * \bar{s}_i^3 + e * \bar{s}_i^4] \quad (5)$$

where:

$\bar{T}_i$ = [ft-lb] nominal brake torque calculated for each ith hub end

$\bar{s}_i$ = [in] nominal (or "normal") in-adjustment brake stroke for each ith hub end

(i.e., 1.40 inches for a standard 20-in² chamber, 1.75 inches for a standard 30-in² chamber)

An appropriate level of total braking force available for the subject vehicle - in the state of brake adjustment and readiness observed in the field - is then calculated using the same polynomial equations applied to the actual stroke measurements taken from the field (see Equation (6)). If a brake was found to be non-functioning prior to the best-effort stop under analysis, then placing a large stroke value into the model (say 3.5 inches or 88.9 mm) will force that output to be "0." The forces for all hub ends are again summed, yielding a total "as measured" braking force sum for the subject vehicle.

Obviously, for those hub ends having brake strokes large enough to compromise the vehicle's stopping performance, the "as measured" level will be lower than the "normal" level calculated above. Also note that limits must be applied to the values calculated using the polynomial relationships, as small stroke measurements will result in unrealistically high brake torque, and large stroke measurements will result in negative brake torques. The limit is applied using the relationship shown in Equation (7).

$$T_i = \frac{1}{12} [a + b * s_i + c * s_i^2 + d * s_i^3 + e * s_i^4] \quad (6)$$

where:

T_i = [ft-lb] actual brake torque calculated for each i^{th} hub end,

s_i = [in] brake stroke measured at each i^{th} hub end

$$T_i = \begin{cases} 0 & \text{if } T_i < 0 \\ \bar{T}_i & \text{if } T_i > \bar{T}_i \\ T_i & \text{otherwise} \end{cases} \quad (7)$$

The numeric output of equations (5) and (6) is only relevant in that they are used to generate a ratio and therefore should not be directly applied to estimate a vehicle's braking performance.

The ratio of the "as measured" force sum and the "normal" force sum will therefore yield a Performance Adjustment Ratio (PAR) less than 1.00 (see Equation (8)), which is applied to the range of appropriate "typical" values for drag factors calculated in the previous sub-section, and a revised drag factor for the subject commercial vehicle is derived (Equation (9)). The strength of this derivation is that the braking performances and amount of appropriate de-rating of the subject vehicle are all based on known, measured quantities, and not theoretical values that depend on many assumptions that are difficult to verify. Another strength in using the ratio method is that when the force sums are divided, the mathematical (and theoretical) effect of the slack adjuster length is nullified. Therefore, since the Flick data - polynomial force values - are only used to form a ratio to de-rate the total braking ability, the effect of slack adjuster length on the basic calculations can be ignored. An exercise in the Appendix II has an illustrative example for a typical 5-axle tractor-semitrailer. The calculations depicted in the example were performed using MathCAD [9], but can also be performed in a spreadsheet. One strength in using MathCAD is that the equations are presented in their entirety, which eases equation checking.

$$PAR = \frac{\sum T_i}{\sum \bar{T}_i} \quad (8)$$

$$DF_{adj} = PAR * DF_{norm} \quad (9)$$

The authors believe that another improvement is that the PAR can be applied to the upper and lower confidence limit (or ranges if one prefers) to construct a vector that covers a plausible range of performance levels for the subject commercial vehicle. This methodology is carried forth in the example presented in Appendix II.

ADDING TO THE EXISTING KNOWLEDGE

Heusser published chamber pushrod force values for many standard stroke chambers that were in use when the work was published in 1991. In the past, the authors have modified the relationships presented above to adapt for force-versus-stroke characteristics of long-stroke brake chambers. This modification can be handled by modifying the brake output calculations to account for the widening of the "near 100% effectiveness" portion of the graphs shown in Figure 5 and Figure 6. This method is justified by existing literature on long-stroke brake chambers [11]. The numeric data for standard and long stroke chambers presented in [11] is co-plotted in Figure 7 with data previously shown. The data from the 30-square inch (193.5 cm²), long stroke chamber is taken from data in the Appendix of this paper and also co-plotted for comparison.

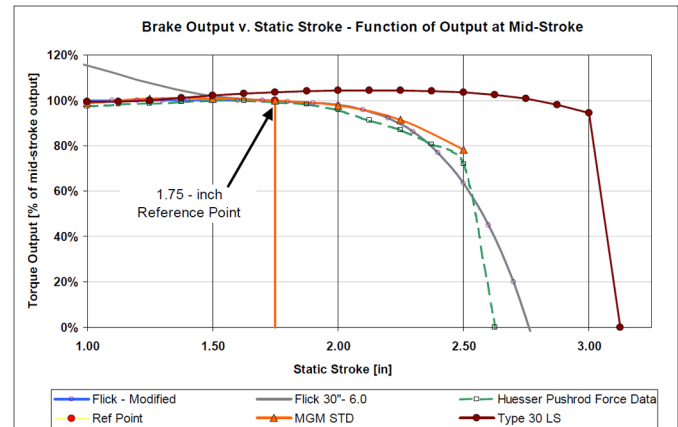


Figure 7. Chart comparing the normalized output from dynamometer data and pushrod force data, with published standard and long-stroke pushrod force data added for reference.

COMPARISON OF NEW MODEL PREDICTION WITH VEHICLE TEST DATA

The model predictions were compared to contemporary full-scale vehicle braking test data. The test methodology and results are discussed in detail in [6]. The testing was performed using a 6×4 truck tractor and a 2-axle, 48-foot (14.63 m) flatbed semitrailer. In addition to the baseline test, selected brakes were disabled for subsequent tests. The tests were run with various foundation brakes disabled (up to four