Fundamentals of Steam Locomotive Tractive Force

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Tractive Force.

An ordinary steam locomotive relies upon rotational force or torque developed at the driving wheels produced by the machinery of the locomotive and friction between the driving wheels and the rails to propel it and any cars attached along the track. The propelling force developed is generally referred to as tractive force or tractive effort.

For the remainder of this article references to a steam locomotive shall be understood to be an ordinary two cylinder, reciprocating double acting, single expansion, non condensing, adhesion or friction traction, side rod driven steam locomotive. Rack drive, geared drive, three or more cylinder, multiple expansion compound or steam turbine drive, etc., etc., steam locomotives are not considered.

Derivation of Tractive Force, Single Cylinder Steam Engine Example.

A steam locomotive may be thought of as comprising two single cylinder reciprocating double acting steam engines. One on the left side and the other on the right side of the locomotive. Each engine imparting rotation to a common crank shaft, that being the main driving axle of the locomotive. Each engine imparting rotation to a common crank shaft. Figure 1 illustrates such a single cylinder steam engine having a rope winding drum coaxially attached to the crank shaft. Given the diameter of the cylinder and piston, and the steam pressure applied in the cylinder, a theoretical force of 50,000 pounds is applied to the crank pin by the main rod. Given the surface of the rope winding drum and the center of the crank pin are the same distance from the center of the crank shaft, for the crank pin position illustrated, theoretically a weight of 50,000 pounds may be raised by the wire rope attached to the winding drum. The effects of inertia and losses due to the angularity of the main rod and friction between moving parts of the engine are ignored.

![Diagram of single cylinder steam engine](image-url)
Formula 1 may be used to ascertain the piston force of the engine in Figure 1.

\[ F_p = P_c \times \left( \frac{C}{2} \right)^2 \times \pi \]  
(Formula 1)

where:

- \( F_p \) = Piston force in pounds.
- \( P_c \) = Steam pressure in cylinder in pounds per square inch gauge.
- \( C \) = Cylinder bore diameter or piston diameter in inches.
- \( \pi \) = 3.14159

Using Figure 1 engine:

\( P_c = 176.35 \) p.s.i.g.
\( C = 19 \) inches.

results in:

\[ F_p = 50,000 \text{ pounds} = 176.35 \times \left( \frac{19}{2} \right)^2 \times 3.14159 \]

In Figure 2 the rope winding drum of the engine has been increased to 48 inches in diameter and better represents the tread diameter of a driving wheel of a steam locomotive.

In Figure 2 the surface of the rope winding drum is 24 inches from the center of the crank shaft and the center of the crank pin is 12 inches from the center of the crank shaft. Because of this dimensional difference the force developed at the rim of the rope winding drum is less than the piston force transmitted to the crank pin by the main rod. The effects of inertia of and losses due to the angularity of the main rod and friction between moving parts of the engine are ignored.
Rule II of Levers.

To find the delivered force, multiply the applied force by the length of the lever between the applied force and the fulcrum, and divide by the length of the lever between the delivered force and the fulcrum.

The foregoing may be applied to the engine illustrated in Figure 2 by assuming the center line of the crank shaft is the fulcrum, the center line of the crank pin is the point of applied force, and the surface of the rope winding drum (tread surface of a locomotive driving wheel) is the point of delivered force.

\[ F_d = F_a \times \frac{L_a}{L_d} \]  (Formula 2)

where:

- \( F_d \) = Delivered force in pounds.
- \( F_a \) = Applied force in pounds or piston force \( F_p \) in pounds from Formula 1.
- \( L_a \) = Length of lever between fulcrum and applied force in inches.
- \( L_d \) = Length of lever between fulcrum and delivered force in inches.

Using Figure 2 engine:

- \( F_a = 50,000 \) pounds.
- \( L_a = 12 \) inches.
- \( L_d = 24 \) inches.

results in:

\[ F_d = 25,000 \text{ pounds} = \frac{50,000 \times 12}{24} \]

If the crank shaft of the engine in Figure 2 is considered to be the main driving axle of a single driving axle locomotive, then the two 48 inch diameter driving wheels attached would share the 25,000 pounds of force illustrated resulting in 12,500 pounds of tractive force at the tread surface of each of the two driving wheels.

Effect of Crank Pin Position.

If the engine in Figure 3 is considered to be rotating clockwise, the maximum torque developed at the crank shaft derived from the force applied to the crank pin by the main rod will occur when the crank pin is near the 90 degree position during the leftward stroke of the piston, and near the 270 degree position during the rightward stroke of the piston. In any other crank pin position the effective crank arm length is reduced, thus for any given cylinder steam pressure the torque developed at the crank shaft will be reduced when compared to the 90 degree or 270 degree crank pin positions.
Cylinder Steam Pressure During Stroke of Piston.

Figure 4 illustrates simplified theoretical cylinder steam pressure from the beginning to the end of the stroke of the piston of a typical engine illustrated in Figure 3. This occurring on the right side of the piston to produce one-half a revolution of the crank shaft (0 degrees to 90 degrees to 180 degrees), then the similar occurring on the left side of the piston to complete the revolution of the crank shaft (180 degrees to 270 degrees to 0 degrees). The valve gear of the engine is taken to be set for maximum cut-off (set for full forward or full backward, typical when starting a steam locomotive), thereby providing maximum cylinder steam pressure during the majority of the stroke of the piston. In this case being 90 percent. In addition the engine is considered to be operating under maximum load with wide open throttle and is rotating at low speed. Under these conditions the maximum cylinder steam pressure will be less than although near the boiler steam pressure.

Fundamentals of Steam Locomotive Tractive Force

Figure 4

THEORETICAL CYLINDER STEAM PRESSURE vs PISTON STROKE

Combined Effect of Crank Pin Position and Cylinder Steam Pressure.

Figure 5 illustrates the theoretical torque developed by the engine in Figure 3 as the crank shaft makes one revolution. The serpentine line has been plotted from data resulting from calculations that take into consideration the varying effective crank arm length, the varying steam pressure (admission pressure) on the piston tending to move the piston in the direction of rotation of the crank shaft, and the varying exhaust steam pressure (back pressure) on the opposite side of the piston tending to retard the movement of the piston. The engine is again considered to be operating under maximum load with wide open throttle and rotating at low speed. The effects of inertia of moving parts of the engine are ignored.

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Figure 5

THEORETICAL TORQUE DEVELOPED RESULTING FROM COMBINED EFFECTS OF VARYING CRANK PIN POSITION AND CYLINDER STEAM PRESSURE
Coupled Driving Wheels.

Figure 6 illustrates the mechanical scheme typically used to couple the driving wheels of a steam locomotive. If the locomotive in Figure 6 is considered to be moving forward and toward the right, then the right side crank pins are referred to as leading the left side crank pins. Whether the locomotive designer specified the right side crank pins to lead the left or the left side crank pins to lead the right is of no consequence. Regardless of the number of coupled driving axles or on what side the crank pins lead the other; the primary design element is the 90 degree angular difference between the crank pins on the left side and right side of the locomotive. Positioning the crank pins in this manner results in the four force impulses produced by the pistons of the left and right sides of the locomotive to occur at approximately equal intervals as the driving axle or axles make each revolution. This also provides for the starting of the locomotive in any rotational position of the driving wheels given that when one side of the locomotive is on either dead center the other side is midway between its dead centers.

![Fundamentals of Steam Locomotive Tractive Force](image)

Torque Diagram for Two Cylinder Steam Locomotive.

Figure 7 illustrates a theoretical torque diagram for a steam locomotive produced by calculations similar to those used for producing the torque diagram of Figure 5. The two black color serpentine lines illustrate the theoretical torque produced by the individual double acting steam engines on the left and right sides of the locomotive. The red color serpentine line illustrates the theoretical combined torque or tractive force produced by the locomotive given the engines of the left and right sides are coupled together by the driving axles as illustrated in Figure 6. The locomotive in this case is considered to be operating at low speed under maximum load, at maximum allowed boiler steam pressure, with wide open throttle and the valve gear set for maximum cut-off. The effects of inertia of moving parts of the engine are ignored.

![Fundamentals of Steam Locomotive Tractive Force](image)
**Tractive Force Formulas.**

Torque diagrams such as Figure 7 clearly illustrate the varying nature of the tractive force produced by a steam locomotive under the theoretical conditions described. The data points calculated in order to produce a torque diagram may also be used to ascertain the mean or average tractive force produced. Of more practical use would be a mathematical formula the result of which would be an estimate of the mean tractive force produced, thus avoiding the necessity of calculating the numerous data points required to construct a torque diagram.

Figure 7 illustrates the minimum combined tractive force of a steam locomotive is produced at points in the rotation of the driving wheels at which the torque produced by the individual engines of either the left or right side of the locomotive is the greatest.

By combining previously described Formulas 1 and 2, Formula 3 is produced the result of which may be considered to be either the estimated maximum torque produced independently by each of the left side or right side engines of the locomotive, or the estimated minimum combined tractive force Tmin produced by the locomotive as a whole.

\[ T_{min} = P_c \times \left( \frac{C}{2} \right)^2 \times \frac{P_i \times L_a}{L_d} \]  
(Formula 3)

where:

- \(T_{min}\) = Tractive force minimum in pounds.
- \(P_c\) = Steam pressure in cylinder in pounds per square inch gauge.
- \(C\) = Cylinder bore diameter or piston diameter in inches.
- \(P_i = 3.14159\)
- \(L_a\) = Length of lever between fulcrum and applied force in inches, or distance from center of crank shaft to center of crank pin in inches, or one-half the stroke of the piston in inches.
- \(L_d\) = Length of lever between fulcrum and delivered force in inches, or distance from center of crank shaft to tread of driving wheel in inches, or one-half the driving wheel diameter in inches.

In Formula 3 the length of the stoke of the piston may be substituted for \(L_a\) if the driving wheel diameter is substituted for \(L_d\), thereby producing Formula 4.

\[ T_{min} = P_c \times \left( \frac{C}{2} \right)^2 \times P_i \times \frac{S}{D} \]  
(Formula 4)

where:

- \(T_{min}\) = Tractive force minimum in pounds.
- \(P_c\) = Steam pressure in cylinder in pounds per square inch gage.
- \(C\) = Cylinder bore diameter or piston diameter in inches.
- \(P_i = 3.14159\)
- \(S\) = Stroke of the piston in inches.
- \(D\) = Driving wheel diameter in inches.

In Formula 4 instead of finding the area of the piston by using \(\left( \frac{C}{2} \right)^2 \times P_i\), the area of the piston may be found by using \(C^2 \times P_i / 4\). \(P_i / 4\) producing constant \(K_1 = 0.785398\), thereby producing Formula 5.

\[ T_{min} = P_c \times C^2 \times K_1 \times \frac{S}{D} \]  
(Formula 5)

where:

- \(T_{min}\) = Tractive force minimum in pounds.
- \(P_c\) = Steam pressure in cylinder in pounds per square inch gauge.
- \(C\) = Cylinder bore diameter or piston diameter in inches.
- \(K_1 = 0.785398\)
- \(S\) = Stroke of the piston in inches.
- \(D\) = Driving wheel diameter in inches.

The reader is reminded that for this article references to a steam locomotive shall be understood to be an ordinary two cylinder, reciprocating double acting, single expansion, non condensing, adhesion or friction traction, side rod driven steam locomotive. Rack drive, geared drive, three or more cylinder, multiple expansion compound or steam turbine drive, etc., etc., steam locomotives are not considered.

The ratio between the minimum and the mean tractive force of Figure 7 may be used to establish the value of constant $K_2$ that when multiplied by the minimum tractive force $T_{min}$ may be used to estimate the mean tractive force. In the case of the torque diagram of Figure 7 the ratio will be discovered to be approximately 1:1.2, therefore constant $K_2$ is assigned the value of 1.2.

In the previously described formulas steam pressure in the cylinder $P_c$ was used in the calculations. In practice the steam pressure in the cylinder is not actually ascertained and instead is estimated to be a pressure less than the maximum boiler steam pressure specified for the locomotive. The amount of the estimated pressure loss establishes the value of constant $K_3$ and is determined by taking into consideration the expected pressure losses in the steam conveying system from the boiler to the cylinders, the cylinder admission steam pressure characteristics as theoretically illustrated by Figure 4 and the exhaust steam back pressure characteristics (not illustrated) tending to retard the motion of the piston. The admission and exhaust pressure characteristics taken to be those resulting from the locomotive valve gear being set for maximum cut-off and the locomotive operating under maximum load with wide open throttle and moving at low speed.

By incorporating into Formula 5 the constants $K_2$ and $K_3$ Formula 6 is produced the result of which is designated as the rated tractive force $T$ of a steam locomotive.

$$T = K_1 \times K_2 \times K_3 \times P \times C^2 \times S / D \quad \text{(Formula 6)}$$

where:

$T =$ Rated tractive force in pounds.
$K_1 = 0.785398$
$K_2 = 1.2$
$K_3 = 0.90$
$P =$ Maximum boiler steam pressure in pounds per square inch gauge.
$C =$ Cylinder bore diameter in inches.
$S =$ Stroke of the piston in inches.
$D =$ Driving wheel diameter in inches.

The constants $K_1$, $K_2$ and $K_3$ of Formula 6 may be multiplied together resulting in one constant $K$ having a value of 0.85, thereby producing Formula 7 that in the United States is the commonly published formula for ascertaining the rated tractive force $T$ of an ordinary steam locomotive.

$$T = K \times P \times C^2 \times S / D \quad \text{(Formula 7)}$$

where:

$T =$ Rated tractive force in pounds.
$K = 0.85$
$P =$ Maximum boiler steam pressure in pounds per square inch gauge.
$C =$ Cylinder bore diameter in inches.
$S =$ Stroke of the piston in inches.
$D =$ Driving wheel diameter in inches.

**Constant $K$.**

In many instances Formula 7 is published using a designated value instead of the variable letter $K$.

$$T = 0.85 \times P \times C^2 \times S / D \quad \text{(Formula 7 alternative)}$$
The constant value of 0.85 in Formula 7 alternative is occasionally referred to as being a value that represents solely the difference between the maximum boiler steam pressure and the expected steam pressure in the cylinders of the locomotive. This explanation is not entirely correct as multiple elements, previously described constants K1, K2 and K3, are in fact involved when establishing the value of constant K.

Rated tractive force data from the 1950-1952 Locomotive Cyclopedia of American Practice provides an example of the value of constant K being adjusted in order to take into consideration more detailed operational circumstances.

### Table 1

**Values of Constant K for an Ordinary Steam Locomotive from the 1950-1952 Locomotive Cyclopedia of American Practice.**

<table>
<thead>
<tr>
<th>Main Valve Maximum Cut-off Percent</th>
<th>Constant K for Main Valve without Auxiliary Ports</th>
<th>Constant K for Main Valve with Auxiliary Ports with 80% Minimum Cut-off</th>
</tr>
</thead>
<tbody>
<tr>
<td>90</td>
<td>0.85</td>
<td>......</td>
</tr>
<tr>
<td>80</td>
<td>0.80</td>
<td>0.80</td>
</tr>
<tr>
<td>70</td>
<td>0.74</td>
<td>0.78</td>
</tr>
<tr>
<td>60</td>
<td>0.68</td>
<td>0.77</td>
</tr>
<tr>
<td>50</td>
<td>0.60</td>
<td>0.75</td>
</tr>
</tbody>
</table>

The 1922 through 1950-1952 editions of the Locomotive Cyclopedia define auxiliary starting port as, “A small opening or port in the valve chamber bushing which is uncovered by the valve before the main ports are opened. Its purpose is to allow steam to enter the cylinders for starting the engine where the steam lap is greater than about 1 inch.” From a review of data and drawings published in the aforementioned editions it may be concluded that only a small minority of steam locomotives constructed in the United States were equipped with auxiliary starting ports. Due to the added complexity required for their construction and their likely minimal operational benefit, auxiliary starting ports are typically not applied on a miniature practice steam locomotive even though its full size practice example was so equipped.

**Rated Tractive Force, Conditions in Effect or Ignored.**

The use of the term rated tractive force indicates the resulting value is an estimate of steam locomotive mean or average tractive force developed at the treads of the driving wheels and is based on a series of conditions considered to be in effect actually or in theory, while others are ignored. Unless specified otherwise, in the United States the following is generally understood to apply in full size and miniature practice:

- Boiler at maximum allowed steam pressure;
- Wide open throttle;
- Valve gear position full forward maximum cut-off typically taken as 90%;
- Valves without auxiliary ports;
- Velocity from zero to not more than 10.0 miles per hour (m.p.h.) in full size practice;
- Velocity from zero to not more than 1.25 m.p.h. in 1:8 scale miniature practice;
- Effect of velocity, ignored;
- Driving wheels not slipping relative to rails, theoretical;
- Driving wheels to rails coefficient of friction or factor of adhesion, ignored;
- Inertia of steam locomotive as a vehicle and its propelling machinery in motion; ignored;
- Resistance of propelling machinery of steam locomotive, ignored;
- Rolling resistance of steam locomotive and tender, ignored.

In the United States formulas used to ascertain rated tractive force generally do not take into consideration elements of resistance inherent in the steam locomotive that must be overcome by a portion of the force produced by the pistons. The actual tractive force developed at the treads of the driving wheels will therefore be less than the rated tractive force and in turn the draw bar force will be an amount that is further reduced. In full size practice the well known formula originally suggested by W. J. Davis, Jr., is generally used to establish the resistance attributable to rolling resistance on straight and level track of a steam locomotive, and, if equipped, the tender. The resistance of the propelling machinery of a steam locomotive caused by main piston ring friction, packing gland friction, cross head friction, crank pin friction, valve gear friction, etc., etc., in full size practice is generally taken to be equal to 1 percent of the total weight on drivers in pounds and is added to the value of resistance derived from the Davis formula.
Weight on Drivers.

Weight on drivers is defined in the 1922 Locomotive Cyclopedia of American Practice as, “The weight of a locomotive in working order that is supported by the coupled driving wheels when it rests on straight and level track.”

Friction Between Driving Wheels and Rails.

The maximum actual tractive force that may be developed at the treads of the driving wheels is limited by the weight on drivers and the condition of friction that exists between the driving wheels and the rails. The condition of friction influenced primarily by whether the treads of the driving wheels and rails are dry, moist, greasy, sanded or not sanded, etc., etc., and the extent that relative motion between the tread surfaces of the driving wheels and rails may or may not be present as when the driving wheels are rotationally slipping relative to the rails. The foregoing circumstance not to be confused with driving wheels that are not rotating and are sliding along the rails due to the application of braking force.

If the maximum actual tractive force that may be developed at the treads of the driving wheels is divided by the weight on drivers, both expressed in pounds, the result is a proportion that numerically represents the condition of friction and is known as the coefficient of friction. A lower coefficient of friction being indicative of more “slippery” conditions when compared with a higher coefficient of friction being indicative of less “slippery” conditions. If the reciprocal of the foregoing calculation is used then the result is known as the factor of adhesion. A factor of adhesion of 4.0 is equivalent to a coefficient of friction of 0.25.

Coefficient of Friction or Factor of Adhesion, Steam Era Understanding.

During the steam powered railway era in the United States it was generally explained that regardless of other conditions that might effect the coefficient of friction, when rotational slipping of the driving wheels commenced a decrease in the coefficient of friction would result when compared to when the driving wheels were considered to be not slipping.

Table 2
Typical Values for Coefficient of Friction and Factor of Adhesion at Assumed Point of Incipient Driving Wheel Slipping for Steel Tired Driving Wheels on Steel Rail Track.
Data from The Steam Locomotive by R. P. Johnson, Edition of 1942.

<table>
<thead>
<tr>
<th>Wheel-Rail Conditions</th>
<th>Coefficient of Friction</th>
<th>Factor of Adhesion</th>
</tr>
</thead>
<tbody>
<tr>
<td>Clean and dry, sanded.</td>
<td>0.30</td>
<td>3.33</td>
</tr>
<tr>
<td>Clean and dry, not sanded.</td>
<td>0.25</td>
<td>4.0</td>
</tr>
<tr>
<td>Moist, not sanded.</td>
<td>0.15</td>
<td>6.66</td>
</tr>
<tr>
<td>Greasy, not sanded.</td>
<td>0.15</td>
<td>6.66</td>
</tr>
</tbody>
</table>

Coefficient of Friction, Steel Rail Track vs. Aluminum Rail Track, 1:8 Scale Miniature Practice.

Anecdotal experience in 1:8 scale miniature practice reveals that when a locomotive with cast iron or steel tread surface driving wheels is developing tractive force and transitions from aluminum rail track to steel rail track the coefficient of friction on the steel rail track will be observed to be less than that on the aluminum rail track. The engineer of such a locomotive will generally take preemptive action by reducing the tractive force developed during the transition in order to avoid the onset of driving wheel slipping on the steel rail track. Detailed studies into the difference of coefficient of friction between steel rail and aluminum rail track in 1:8 scale miniature practice is limited at this time.

Coefficient of Friction or Factor of Adhesion, Diesel-Electric Era Understanding.

After the end of the steam powered railway era in the United States research continued into the condition of friction existing between the driving wheels and rails for diesel-electric locomotives and in particular its relation to driving wheel tractive force and slip control technologies. A notable study on the subject produced by C. F. Logston, Jr. and G. S. Itami, circa 1980, describes in detail the phenomenon known as friction creep or wheel creep where the tread surface of a driving wheel producing tractive force travels a rotational distance an amount that is greater than the distance the driving wheel has moved lineally along the rail. The foregoing typically expressed as percent wheel creep, e.g., 5% wheel creep results when a driving wheel moves 100 inches lineally along the rail and its tread surface has rotated 105 inches. Current theory assumes that for any driving wheel developing tractive force and lineal movement on the rail there is always present an amount of driving wheel friction creep or wheel creep. It was further observed that within a range of percent wheel creep
the coefficient of friction increased instead of decreasing as was typically explained and expected during the steam era. For the circumstances designated in Table 3 the peak coefficient of friction of 0.314 occurs at a percent wheel creep of 7.5.

Table 3

Average Values for Percent Wheel Creep, Percent Friction and Coefficient of Friction for Typical Diesel-Electric Locomotive Steel Driving Wheels on Dry Steel Rail Tangent Track from Graphical Data Presented in Locomotive Friction-Creep Studies by C. F. Logston, Jr. and G. S. Itami, circa 1980.

<table>
<thead>
<tr>
<th>Percent Wheel Creep</th>
<th>Percent Friction</th>
<th>Coefficient of Friction</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.6</td>
<td>20.1</td>
<td>0.201</td>
</tr>
<tr>
<td>2.5</td>
<td>27.9</td>
<td>0.279</td>
</tr>
<tr>
<td>5.0</td>
<td>30.5</td>
<td>0.305</td>
</tr>
<tr>
<td>7.5</td>
<td>31.4</td>
<td>0.314</td>
</tr>
<tr>
<td>10.0</td>
<td>30.9</td>
<td>0.309</td>
</tr>
<tr>
<td>12.5</td>
<td>29.9</td>
<td>0.299</td>
</tr>
<tr>
<td>15.0</td>
<td>28.9</td>
<td>0.289</td>
</tr>
<tr>
<td>17.5</td>
<td>27.9</td>
<td>0.279</td>
</tr>
<tr>
<td>20.0</td>
<td>27.1</td>
<td>0.271</td>
</tr>
<tr>
<td>22.5</td>
<td>26.3</td>
<td>0.263</td>
</tr>
<tr>
<td>25.0</td>
<td>25.6</td>
<td>0.256</td>
</tr>
<tr>
<td>27.5</td>
<td>24.9</td>
<td>0.249</td>
</tr>
<tr>
<td>30.0</td>
<td>24.2</td>
<td>0.242</td>
</tr>
</tbody>
</table>

Friction Creep or Wheel Creep During the Steam Era.

The phenomenon of driving wheel friction creep or wheel creep may now be understood as having existed since the inception of railway locomotive propulsion utilizing adhesion or friction traction. Although the increased coefficient of friction associated with low percentages of wheel creep having gone unnoticed or not commonly reported during the steam era. The characteristic variation in tractive force developed by a steam locomotive at low velocities, as illustrated by Figure 7, contributes to the occurrence of typical driving wheel slipping where a percent wheel creep of 500 or more is attained rapidly from the point where during the steam era it was considered that the driving wheels were not slipping. The foregoing contributing to the difficulty in observing a steam locomotive operating at low percentages of wheel creep and increased coefficients of friction.

The data in Table 4 was compiled by the author from an analysis of a video recording of a former Denver and Rio Grande Western Railroad 3 foot narrow gauge steam locomotive moving backward slowly and shoving a cut of cars. During the video sequence the driving wheels slip intermittently.

Table 4

Frame Counting Analysis of Video Recording of Former Denver and Rio Grande Western Railroad Class K36 2-8-2 Type Steam Locomotive with 44 Inch Diameter Driving Wheels. Slipping Controlled by Engineer to Not More Than 2 Revolutions Duration per Occurrence.

<table>
<thead>
<tr>
<th>Driving Wheels Considered Not Slipping</th>
<th>Driving Wheels Considered Slipping</th>
</tr>
</thead>
<tbody>
<tr>
<td>Driving Wheel Rate of Rotation</td>
<td>8.9 R.P.M.</td>
</tr>
<tr>
<td>Driving Wheel Tread Velocity</td>
<td>102.5 F.P.M.</td>
</tr>
<tr>
<td>Percent Wheel Creep</td>
<td>1.0 Assumed</td>
</tr>
<tr>
<td>Locomotive Track Velocity</td>
<td>101.5 F.P.M. or 1.15 M.P.H.</td>
</tr>
<tr>
<td>Assumed Coefficient of Friction for Clean, Dry and Not Sanded Rails</td>
<td>0.25</td>
</tr>
</tbody>
</table>

R.P.M. = Revolutions per minute.
F.P.M. = Feet per minute.
M.P.H. = Miles per hour.
The Skillful Engineer.

A most prized attribute of a steam locomotive engineer is their ability to properly manage the relationship between the potential tractive force that may be produced by the pistons and the condition of friction that exists at any moment between the tread surfaces of the driving wheels and the rails. An engineer possessing this attribute can ensure that when required by circumstances the maximum tractive force capable of being produced by the locomotive under given wheel-rail surface conditions is transmitted to the draw bar thereby taking full advantage of the design elements of the locomotive to accomplish the maximum work.

Prolonged uncontrolled high speed rotational slipping of the driving wheels of a steam locomotive if allowed to continue may result in damage or destruction to its machinery due to the unbalanced nature of many of the moving parts. In full size practice changes to the metallurgical properties of the rail is likely to occur resulting in what is referred to as an engine burn fracture of the rail. In addition vertically bent rails are likely to result due to the unbalanced nature of the driving wheels.

Fundamental Proportion.

The proportion weight on drivers divided by the rated tractive force is a fundamental element in the design of a steam locomotive. Engineering literature from the period generally does not include a name assigned to the proportion and instead it is usually presented fully thus, “Weight on drivers divided by [rated] tractive force equals value.” For this article the foregoing proportion shall be expressed as $W/T$ and referred to by name as the factor of traction. The factor of traction does not take into account the resistance of the propelling machinery of a steam locomotive. If it was included the resulting value would be increased minimally.

Although the factor of traction appears similar to the factor of adhesion it is not the same as clarified by the following:

Factor of traction = weight on drivers divided by rated tractive force.
Factor of adhesion = weight on drivers divided by the maximum actual tractive force attainable for the given conditions of friction.

A steam locomotive having a factor of traction near 4.0 might be considered optimally proportioned given that during the steam era a factor of adhesion of 4.0 was usually assumed for the condition of friction existing on dry not sanded rails. A steam locomotive having a factor of traction greater than 5.0 might be considered as having been designed with insufficient rated tractive force. A steam locomotive having a factor of traction less than 4.0 might at first thought to have been designed with excessive rated tractive force. Although in full size practice as the velocity of a steam locomotive is increased above approximately 10 m.p.h. the actual tractive force developed begins to decrease. For a steam locomotive having a factor of traction of 3.5, if its velocity is continuously increased after starting, at some velocity greater than approximately 10 m.p.h. the factor of traction will become equal to 4.0 if the factor of traction is based on the actual tractive force developed instead of the rated tractive force.

The engineer of a steam locomotive having a factor of traction lower than 4.0 must keep in mind that when the boiler pressure is near the maximum allowed, the pistons of the locomotive may be made to develop a high potential tractive force at starting that is most likely not usable even under favorable not sanded conditions of friction between the driving wheels and the rails.

An explanation of the phenomenon of decreasing tractive force development as the velocity increases whether in full size or 1:8 scale miniature practice is beyond the scope of this article.

Common Misunderstandings.

The author is aware of two discrepancies that appear infrequently on full size practice locomotive drawings and in literature. First is when the result for the calculation of weight on drivers divided by rated tractive force is incorrectly designated as the factor of adhesion. Second is when the phrase tractive power is incorrectly used in place of tractive force or tractive effort. Work is defined as displacement caused by force and power is defined as the rate of doing work. Common formulas for rated tractive force do not take into consideration the distance the locomotive has traveled nor any interval of time, therefore force or effort are correct descriptions for the result, not power. An explanation of the derivation and use of horse power as it relates to the steam locomotive is beyond the scope of this article.
Rated Tractive Force Examples, Full size Practice vs. 1:8 Scale Miniature Practice.

4-4-2 Type Locomotive, Pennsylvania Railroad Class E6S for Passenger Service.
Data from the 1922 Locomotive Cyclopedia of American Practice page 175, unless noted with an asterisk.
Track gauge = 56.5 inches;
T.F. formula assumed constant $K = 0.85^*$;
Boiler steam pressure = 205 pounds per square inch gauge (p.s.i.g.);
Cylinder diameter = 23.5 inches;
Piston stroke = 26.0 inches;
Driving wheel diameter over tires = 80.0 inches;
Weight on drivers (W) = 136,000 pounds;
Rated tractive force (T) = 31,275 pounds;
W/T or factor of traction = 4.35;
Driving wheel circumference = 20.94 feet*;
Driving wheel revolution rate at locomotive velocity of 10 m.p.h. = 42.0 revolutions per minute (r.p.m.).

4-4-2 Type Locomotive, Deerfield and Roundabout Railway Class E6 for Passenger Service.
Data from the Deerfield and Roundabout Railway.
Track gauge = 7.5 inches;
T.F. formula constant $K = 0.85$;
Boiler steam pressure = 100 p.s.i.g.;
Cylinder diameter = 2.75 inches;
Piston stroke = 3.25 inches;
Driving wheel diameter over tires = 10.5 inches;
Weight on drivers (W) = 682 pounds;
Rated tractive force (T) = 199 pounds;
W/T or factor of traction = 3.43;
Driving wheel circumference = 2.75 feet;
Driving wheel revolution rate at locomotive velocity of 1.25 m.p.h. = 40.0 r.p.m.

W/T or Factor of Traction, 1:8 Scale Miniature Practice.

For any steam locomotive having a designed factor of traction near 4.0 and when the condition of friction between the driving wheels and the rails results in a factor of adhesion near 4.0, if at any time the boiler steam pressure falls below the maximum allowed, the full weight on drivers may not be utilized to develop the maximum tractive force.

In 1:8 scale miniature practice the small volume of a typical steam locomotive boiler combined with the fuel combustion and feedwater induction procedures practiced by the engineer/fireman generally results in the boiler steam pressure varying within a range that is much greater than what may be accomplished in full size practice. In order to provide for a wide range of boiler steam pressure fluctuation within which the full weight on drivers may be utilized to develop tractive force, a 1:8 scale miniature practice steam locomotive may intentionally be designed with a low factor of traction.

Table 5

<table>
<thead>
<tr>
<th>Boiler Steam Pressure in P.S.I.G.</th>
<th>Rated Tractive Force in Lbs. at Designated Boiler Steam Pressure</th>
<th>W/T Factor of Traction</th>
<th>Theoretical Maximum Actual Tractive Force at Starting and Not More Than 1.25 m.p.h.</th>
<th>Theoretical Maximum Actual Traction Force Limited by</th>
</tr>
</thead>
<tbody>
<tr>
<td>100.0</td>
<td>199.0</td>
<td>3.43</td>
<td>169.1</td>
<td>Weight on Drivers</td>
</tr>
<tr>
<td>95.0</td>
<td>189.0</td>
<td>3.61</td>
<td>169.1</td>
<td>Weight on Drivers</td>
</tr>
<tr>
<td>90.0</td>
<td>179.1</td>
<td>3.81</td>
<td>169.1</td>
<td>Weight on Drivers</td>
</tr>
<tr>
<td>85.0</td>
<td>169.1</td>
<td>4.03</td>
<td>169.1</td>
<td>Weight on Drivers and Boiler Steam Pressure</td>
</tr>
<tr>
<td>80.0</td>
<td>159.2</td>
<td>4.28</td>
<td>159.2</td>
<td>Boiler Steam Pressure</td>
</tr>
<tr>
<td>75.0</td>
<td>149.2</td>
<td>4.57</td>
<td>149.2</td>
<td>Boiler Steam Pressure</td>
</tr>
<tr>
<td>70.0</td>
<td>139.3</td>
<td>4.90</td>
<td>139.3</td>
<td>Boiler Steam Pressure</td>
</tr>
</tbody>
</table>
Suggested Reading.

The following publication is part of the Lake Forest Live Steamers Recommended Technical and Historical Reading List and may be reached via U.R.L. provided at WWW.LFLSRM.ORG

1922 Locomotive Cyclopedia of American Practice.

References.

3. Roy V. Wright, Editor, “1922 Locomotive Cyclopedia of American Practice.”

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Additional Information.

Below and on following pages is presented detailed engineering data for various Deerfield and Roundabout Railway steam locomotives and a brief explanation of auxiliary starting ports.

**Deerfield and Roundabout Railway**

*An educational demonstration steam powered 7 and 1/2 inch gauge railway operated by the Lake Forest Live Steamers Railway Museum Incorporated.*

**STEAM LOCOMOTIVES: 4-6-4 Type**

![Image of Deerfield and Roundabout Railway steam locomotive](image-url)

Deerfield & Roundabout 4-6-4 Type Locomotive, Class J1a, for passenger service.
Buker Locomotive Works, Chicago, Builders No. 2, 1960
Patterned after New York Central, Class J1a, circa 1927.
Old Reliable.
Construction completed 1960 by Edward "Bud" Buker.
Image recorded and edited by Brennan M. Caughron, 2008.

**Specifications:**

- **Rated tractive force (k=0.65)**: 228 lb.
- **Weight on drivers divided by R.T.F.**: 4.08
- **Cylinders, diameter and stroke**: 2 3/4 in. x 8 1/2 in.
- **Drivers, diameter**: 9 7/8 in.
- **Weight on drivers**: 930 lb.
- **Weight on front truck**: 92 lb.
- **Weight on trailing truck**: 214 lb.
- **Total weight of engine**: 1,356 lb.
- **Weight of tender**: 975 lb.
- **Total weight of engine and tender**: 2,321 lbs.
- **Driving wheel base**: 21 0 in.
- **Length over draft faces of couplers**: 9 ft. 6 1/4 in.
- **Fuel**: Soft coal
- **Firebox, length and width**: 16 in. x 9 1/2 in.
- **Grate area**: 1.06 sq. ft.
- **Diameter of boiler, outside**: 10 in.
- **Steam pressure**: 100 lb.
- **Track gauge, minimum on tangent unguarded track**: 7 1/2 in.
- **Outside gauge of flanges**: 7 7/16 in.
- **Reverser**: Manual latch lever quadrant
- **Valves**: Inside admission piston
- **Valve & cylinder lubricator**: Mechanical
- **Engine brakes**: Steam
- **Tender brakes**: Manual
Deerfield and Roundabout Railway
An educational demonstration steam powered 7 and 1/2 inch gauge railway operated by the Lake Forest Live Steamers Railway Museum Incorporated.

STEAM LOCOMOTIVES: 2-8-4 Type

Deerfield & Roundabout 2-8-4 Type Locomotive, Class S1, for freight service.
Patterned after New York, Chicago & St. Louis, Class S1, circa 1942.

Pride of the D&R

Image recorded and edited by Brennan M. Coughlan, 2006.

Rated tractive force (K=0.85).......................... 377 lb.
Weight on drivers divided by R.T.F. ....................... 3.35
Cylinders, diameter and stroke.................. 3 in. x 4 1/4 in.
Drivers, diameter........................................... 8 5/8 in.
Weight on drivers........................................... 1,263 lb.
Weight on front truck............................... 158 lb.
Weight on trailing truck............................. 261 lb.
Total weight of engine................................. 1,622 lb.
Weight of tender........................................... 720 lb.
Total weight of engine and tender.................. 2,342 lb.
Driving wheel base...................................... 30 3/8 in.
Length over draft faces of couplers.............. 11 ft. 10 3/8 in.

Fuel......................................................... Soft coal
Firebox, length and width.......................... 16 1/2 in. x 11 3/4 in.
Grate area.................................................. 1.35 sq. ft.
Diameter of boiler, outside.......................... 22 in.
Steam pressure.......................................... 100 lb.
Track gauge, minimum on tangent unguarded track, 7 1/2 in.
Outside gauge of flanges............................ 7 7/16 in.
Reverse..................................................... Manual lever quadrant
Valves..................................................... Inside admission piston
Valve & cylinder lubricator.......................... Mechanical
Engine and tender brakes............................ Vacuum, steam ejector

Deerfield and Roundabout Railway
An educational demonstration steam powered 7 and 1/2 inch gauge railway operated by the Lake Forest Live Steamers Railway Museum Incorporated.

STEAM LOCOMOTIVES: 4-6-4 Type

Deerfield & Roundabout 4-6-4 Type Locomotive, Class L2, for passenger & freight service.
Patterned after Chesapeake & Ohio, Class L2, circa 1942.


Rated tractive force (K=0.85).......................... 231 lb.
Weight on drivers divided by R.T.F. ....................... 3.37
Cylinders, diameter and stroke.................. 2 3/4 in. x 3 1/2 in.
Drivers, diameter........................................... 8 5/8 in.
Weight on drivers........................................... 779 lb.
Weight on front truck............................... 159 lb.
Weight on trailing truck............................. 341 lb.
Total weight of engine................................. 1,282 lb.
Weight of tender........................................... 675 lb.
Total weight of engine and tender.................. 0.98 tons
Driving wheel base...................................... 21.0 in.
Length over draft faces of couplers.............. 13 ft. 4 in.

Fuel......................................................... Soft coal
Firebox, Length and width.......................... 20.0 in. x 10.0 in.
Grate area.................................................. 1.39 sq. ft.
Diameter of boiler, outside.......................... 11 1/2 in.
Steam pressure.......................................... 100 lb.
Track gauge, minimum on tangent unguarded track, 7 1/2 in.
Outside gauge of flanges............................ 7 1/4 in.
Reverse..................................................... Precision manual screw
Valves..................................................... Outside admission unbalanced "U" slide
Valve & cylinder lubricator.......................... Hydrostatic
Engine and tender brakes............................ Straight air
Air compressor.......................................... Twin cylinder duplex
Auxiliary Starting Ports.

Figure 8 shows a cut-away view of a conventional steam locomotive inside admission piston valve cylinder block not equipped with auxiliary starting ports. Figure 9 shows the same with the exception that auxiliary starting ports have been applied and labeled as “F” and “B.” In both figures the valve and piston are illustrated in positions resulting from the reverse lever being centered and the piston at one-half stroke as when the driving wheels are near either top or bottom quarters. Figure 9 is meant to represent in general data and drawings published in the 1941 Locomotive Cyclopedia of American Practice pertaining to a former Pennsylvania Railroad 2-10-0 Type Locomotive, Class I-1-s, equipped with auxiliary starting ports. The published dimensions of the auxiliary starting ports being 1.5 inches in sideways width and 1/8 inches in front to back width. Unlike conventional valve bushing ports that communicate to the cylinder passages and exhaust passages by completely encircling the valve bushing. The auxiliary starting ports exist in only one location at the bottom of each of the respective valve bushings. In 1:8 scale miniature practice the auxiliary starting ports previously described would have the dimensions of 3/16 inches in sideways width and 1/64 inches in front to back width. A detailed description of the effect that auxiliary starting ports have on the steam admission characteristics to the cylinder is beyond the scope of this article.