

# WHEN — Q1 2015

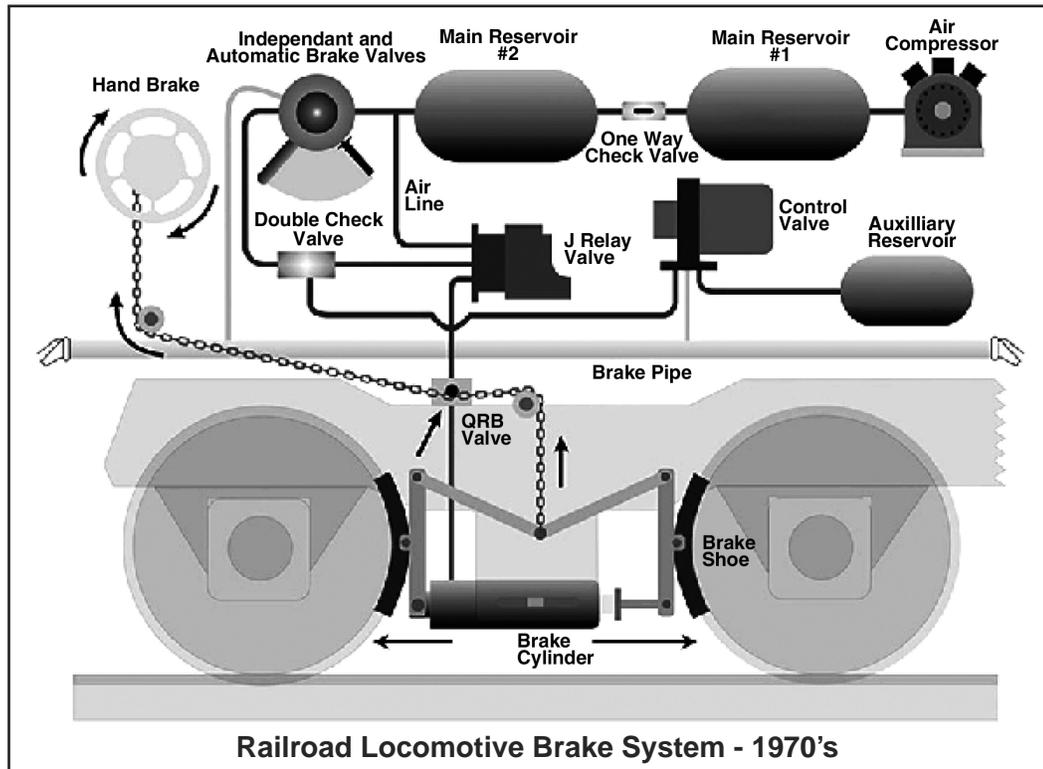
## Federal Safety Standards for Heavy Trucks - Part 3

Update #2349

Attention: Dayton Parts' Distributors and Business Partners.

The Q1 2015 issue of **WHEN** (**WH**eel **E**nd **N**ews)

In Part 1 of this series on the new shorter stopping distance for heavy trucks, we started with the origin of the highway transportation industry and the beginning of Federal Safety Standards focusing mainly on FMVSS-121. In Part 2 we looked at the evolution of the air brake system, from its invention for the railroad industry to the ABS systems we have on heavy trucks today. In Part 3 we're going to focus on the foundation brake which once again has its roots in the railroad industry. Below is the diagram of a cam brake used on railroad locomotives manufactured in the 1970's which has changed very little since its inception in the 1860's.



The design is somewhat different from the s-cam brake used on heavy trucks but the underlying principles are the same. There are fixed brake shoes attached to a lever to gain a mechanical advantage, when the brake force is applied by compressed air, through a brake cylinder (air chamber). The s-cam brake is also a series of simple levers (slack adjuster, s-cam head, and brake shoe) which exchange distance in order to multiply the force being applied but more on that in a bit.

As always I like to first take a look at the historical background on the main subject, which is the s-cam brake.

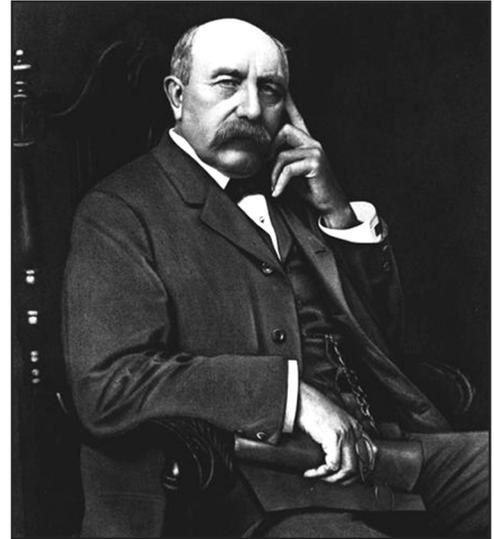
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**Henry Timken** – Henry Timken was born in Bremen, Germany on August 16, 1831, and immigrated as a child with his family to America. In 1855, at the age of 24, he left the family’s farm in Missouri, to open his own wagon making business in St. Louis. In 1877, Mr. Timken received the patent for his Timken Buggy Spring, the first of his 13 patents. His spring design became widely used throughout the country and was produced on a royalty basis by a number of companies. As a result of the spring’s success, Timken became well known across the United States and his carriage business flourished. Around 1895, Mr. Timken started working on ideas for a better wagon axle design. Until then, wagon axles had not changed much since ancient times. Friction bearings, as they were called, were nothing more than oil soaked rags of cotton, called packing, enclosed in a box. Without a constant replenishment of lubrication to this packing, these bearings would fail due to the excessive heat caused by friction, especially when the wagon load was heavy. In 1898, he obtained a patent for the tapered roller bearing and in 1899, he founded the Timken Roller Bearing Axle Company in St. Louis. Mr. Timken’s cup and cone design, that we’re all familiar with, was able to significantly reduce the friction by using roller bearings (as apposed to ball bearings) which had much more contact surface to distribute the weight load from the wagon axle to the wheel. Tapered roller bearings are one of those inventions, that on the surface, we really don’t realize just how much it changed everything. It’s not an overstatement to say that without this innovation there would not be a “modern” transportation industry. In 1901, as the burgeoning automobile industry began to gradually overtake the horse carriage industry, Mr. Timken and his two sons decided to move the company to Canton, Ohio where the corporate offices still reside today. Canton was in close proximity to the car manufacturing plants and steel mills located in Detroit, Cleveland and Pittsburgh, commonly called the “Iron Belt”. As a means of delivering his new invention to the market, Mr. Timken founded the Timken Detroit Axle Company on Clark Street in Detroit, Michigan in 1909.



**Henry Timken**



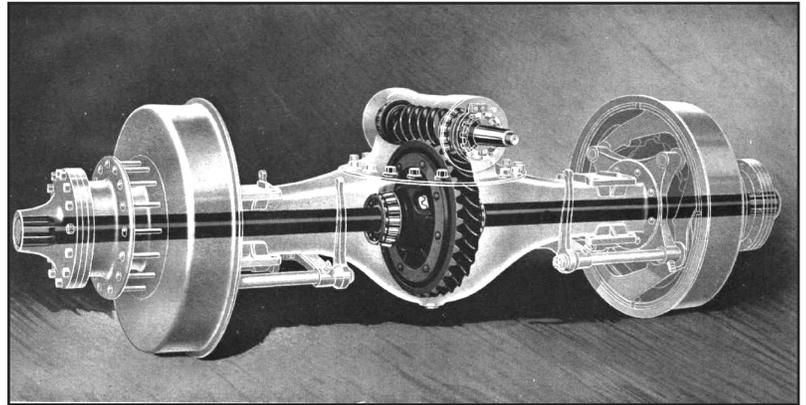
**Timken Detroit Axle Company**

By 1913, the area that could be served daily by a truck was six times larger than what could be done by a horse drawn wagon and trucks could carry nearly four times more weight. The US involvement in WWI bought about a massive deployment of trucks which created an increased demand for steel, affecting its supply and price in the market. This also caused some investors to jump into manufacturing steel at better prices in order to buy some market share, which made the quality of steel available vary quite a bit from one supplier to another. *Does that scenario sound familiar at all?* This became such an important issue, that in 1917 the Timken Roller Bearing Company began its own steel operations in Canton to better maintain control over the steel used in their bearings. With a quality steel product readily available, Timken Detroit Axle set about to take their newly designed drive axle to market.

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Once again as far as the basic design, not much has really changed. Looking at the brake assembly you can see the brake shoes and levers carried over from the railroad industry. You can also see the beginning of what will become the slack adjuster, support bracket and camshaft. One of the key reasons the s-cam brake has hung around for so long is the simplicity of its design (*again more about that in a bit*). Very soon Henry Timken would meet a fellow businessman who would take control of the Timken Detroit Axle Company and leave an indelible mark on the heavy truck industry.



1918 Timken Detroit Drive Axle

His name is Willard Frederick Rockwell.

**Willard F. Rockwell, Sr.** — Born in Boston, Massachusetts in 1888, Mr. Rockwell earned his degree from the Mechanic Arts School at MIT in 1908. After college he went to work for the Torbensen Gear & Axle Company in Cleveland, Ohio eventually becoming the factory manager and then, in 1918, vice-president in charge of engineering and manufacturing. During WWI he was commissioned a major in the Quartermaster Corps and helped to develop standard mobile equipment for the Army, particularly military truck axles and five-ton rear axle drives. In 1919, he purchased the bankrupt Hayes Machine Company in Oshkosh, Wisconsin, renaming it the Wisconsin Parts Company, where he developed worm drive axles, as truck production in the United States was beginning to boom. During the 1920's he spent some of his time and money developing experimental truck and tank drive axles for the Army Ordnance Corps, since Congress had declined to do so. One of the issues facing him was the machining capabilities available at that time. The machines they had could not do a final grind fine enough for the worm gears to work smoothly. He ended up having to import better machining equipment from England to resolve that problem. Once they were able to machine the parts sufficiently, then the steel wouldn't hold up. This led Mr. Rockwell to experiment with different heat treat methods that were also being developed at this time (The Rockwell Hardness scale and tester were invented by Stanley P. Rockwell who was of no relation). He was also using a new bearing he had developed which was a ball design. This design didn't hold up well as the bearing load increased, so he decided to try some tapered roller bearings from the Timken Roller Bearing Company (*didn't see that one coming, eh?*).



Willard F. Rockwell

The success of these new design drive axles led him to merge his Wisconsin Parts Company with the Timken Detroit Axle Company in 1928 to form Timken Detroit Axle and Wisconsin Axle. Wisconsin Axle was eventually absorbed by the Timken Detroit Axle Company which made him Chairman of the Board in 1940, after he helped guide the companies through the Great Depression. During WWII the Timken Detroit Axle Company produced 80% of the axles used in heavy duty Army vehicles. The 2-1/2 ton general purpose truck with four wheel drive, which he helped to design, was later described as one of the six secret weapons which had won the war. He was appointed assistant to the Chief of the Motor Transport Division of the Army and later Director of Production and Procurement in the U.S. Maritime Commission. By 1951, Timken Detroit Axle Company had manufacturing facilities in Detroit and Jackson, MI, Oshkosh, WI, Utica, NY, Ashtabula and Kenton, OH and New Castle, PA. Timken Detroit Axle became the standard drive axle in the heavy truck industry.

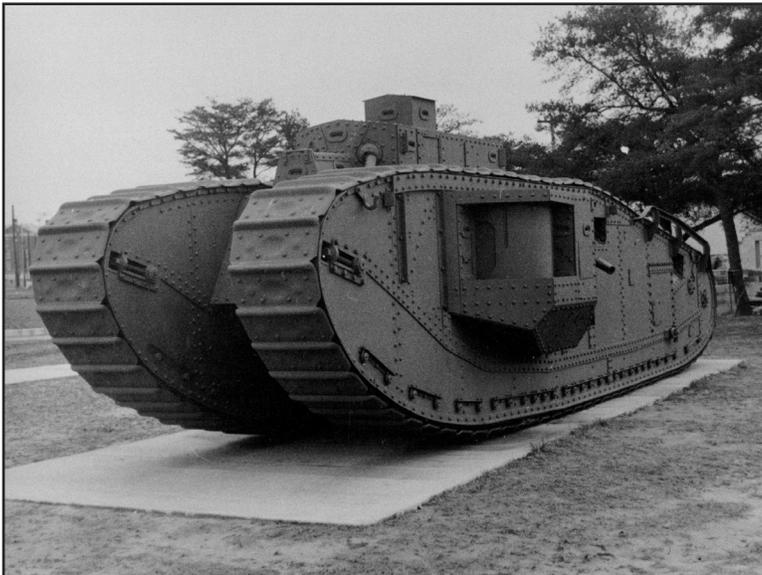


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In 1953, Willard Rockwell decides to merge Timken Detroit Axle with Standard Steel and Spring to form the Rockwell Spring and Axle Company. After various mergers with other automotive suppliers, the company becomes comprised of about 20 factories across the Upper Midwestern US and southern Ontario and in 1958 Rockwell Spring and Axle becomes the Rockwell Standard Corporation. Rockwell Standard then acquired Los Angeles based North American Aviation to form North American Rockwell in September of 1967. It then purchased Miehle-Goss-Dexter, the largest supplier of printing presses, and Collins Radio, a major avionics supplier. Finally in 1973, Rockwell Standard merged with Rockwell Manufacturing, which was run by his son Willard Rockwell, Jr., to form Rockwell International. Mr. Rockwell, Sr. served as the official Chairman of the Board until his death in 1978, at the age of 90. His legacy in the heavy truck industry goes without saying.

Next we'll look an engineer, that you've probably never heard of, who worked for many years under Mr. Timken and then Mr. Rockwell. He was another inventor that was way ahead of the curve or as we say the "father" of the s-cam brake, L. Ray Buckendale.

**L. Ray Buckendale** — Lawrence Ray Buckendale was born in Detroit, Michigan in 1892. While an undergraduate at the University of Michigan he worked as a draftsman for University Motor Car Company and the Dermot Car Company also a machinist for J.N. Smith. Upon graduating in 1916, he went to work as a draftsman at the Timken Detroit Axle Company for Mr. Henry Timken. The young Mr. Buckendale already had an exceptional record in engineering design. When WWI broke out, he took a leave of absence and served as a Captain in the engineering division of the Army's Ordnance Dept. as a technical officer on tanks. He was sent to London to follow through on the design details for development of the Mark VIII tank and then returned to the US to oversee its production at the Locomobile Company (an automobile factory) in Bridgeport, CT. This tank was considered an engineering marvel in its day and rightly so.



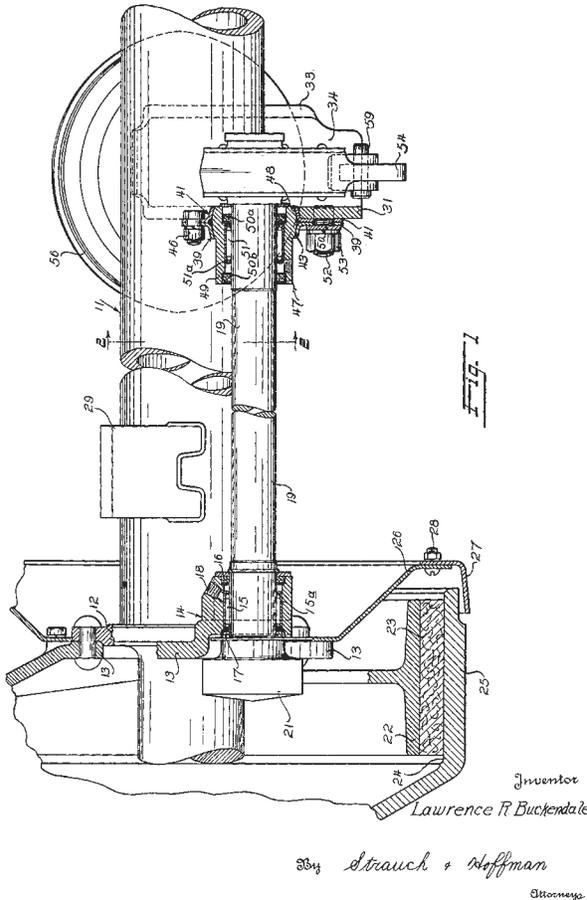
**Mark VIII Tank**

In 1919, he joined a new technical society founded in 1905 by Henry Ford and Andrew Riker (who worked at the Locomobile Company) called the Society of Automotive Engineers or SAE which had such members as Thomas Edison, Glenn Curtiss (inventor of the first aeronautical engine), Glenn L. Martin (founder of what is now called Lockheed Martin) and Orville Wright. Throughout the years between WWI and WWII, Mr. Buckendale worked very closely with the Army Ordnance engineers as a member of the SAE War Engineering Board. He contributed greatly to the development of many heavy duty trucks and tank transport vehicles that helped win WWII and in 1946 he was elected President of SAE.

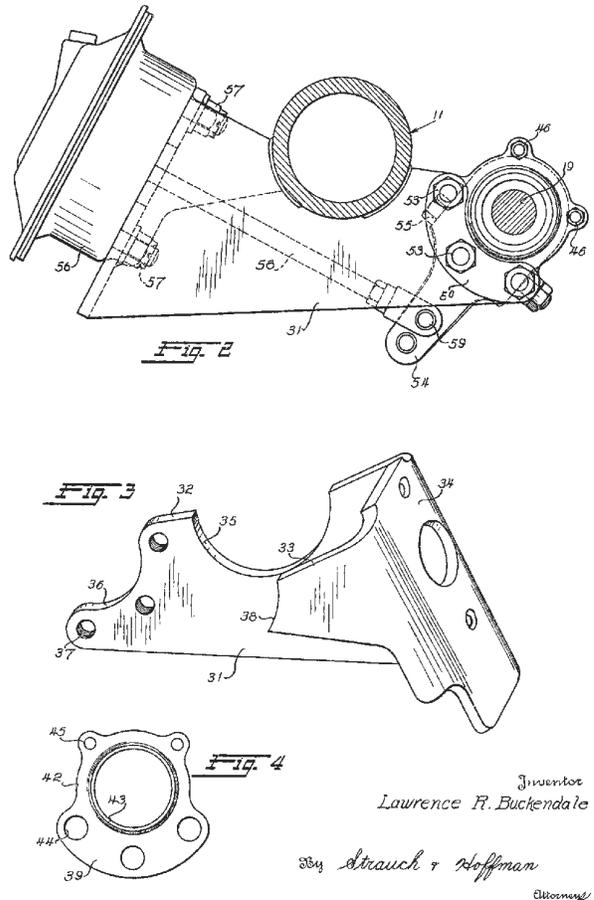
Mr. Buckendale's greatest desire was to develop potential in young people so for this reason the **L. Ray Buckendale Lecture** was created in 1953 to honor his memory. This lecture provides an opportunity for young engineers and students to reap the benefits of sound, practical information on topics within the commercial vehicle industry from the voices of the most esteemed and involved members of the industry. On the following page, are the images from his original patent for the s-cam brake filed in 1941. (*Looks pretty familiar?*)

(continued from page 4)

Oct. 12, 1943. L. R. BUCKENDALE 2,331,652  
 BRAKE MECHANISM  
 Filed April 30, 1941 2 Sheets-Sheet 1



Oct. 12, 1943. L. R. BUCKENDALE 2,331,652  
 BRAKE MECHANISM  
 Filed April 30, 1941 2 Sheets-Sheet 2

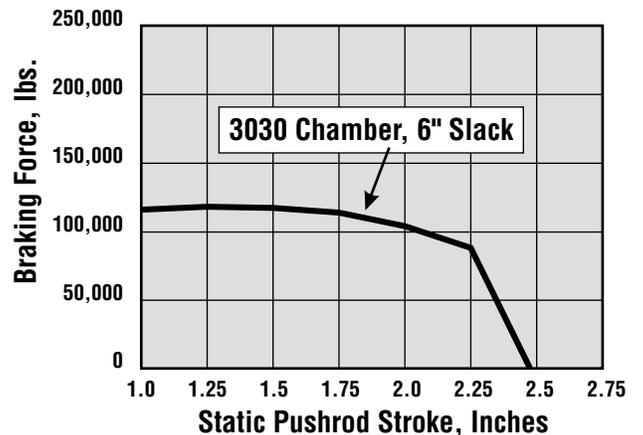


**The S-Cam Brake** — As stated earlier, the s-cam brake is a series of simple levers that takes the pushrod stroke and reduces that movement or distance traveled in order to multiply the force being applied. Here are the three levers used and their values for the common 16.5" drum brake:

1. Slack Adjuster – usually a 5.5" or 6.0" slack arm drilling
2. S-Cam Head – a total lift of 1/2" for a 16.5" s-cam head
3. Brake Shoe – 2 to 1 mechanical advantage

**Slack Adjuster** — A standard 3030 air chamber has 2.5" of pushrod stroke but only 80% of that can be used or 2.0" leaving the additional 0.5" of stroke as a safety margin.

See what happens in this graph for a 3030 chamber with a 6.0" slack when the stroke goes past 2.25".



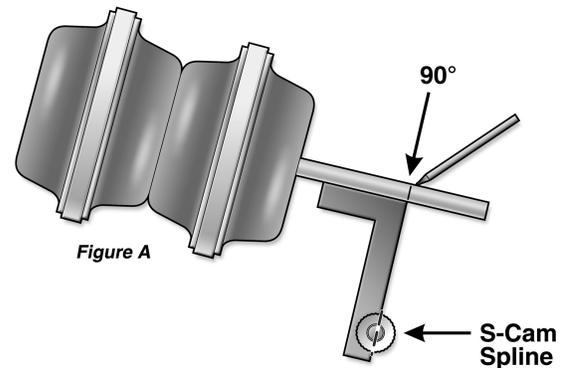
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The amount of braking force begins to taper down slightly at 1.75" of stroke but it falls like a rock after 2.25" of stroke. Why is that? Because if this 16.5" brake assembly is set up correctly it should not take more than 2.0" of stroke to completely apply the brakes. After 2.25" of stroke the slack adjuster has gone well past 90° to the air chamber pushrod and therefore we just lost the mechanical advantage the slack adjuster was providing. By not cutting the air chamber pushrod correctly (which is the main reason for this happening) the effectiveness of the slack adjuster is eliminated basically removing the first of the three levers in the brake assembly.

### Cutting a Type 30 air chamber pushrod for a manual slack adjuster clevis.

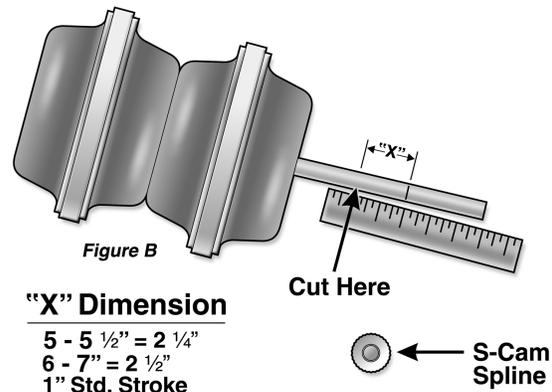
#### Figure A:

First, mount the air chamber in the axle bracket and connect the emergency air line to the spring brake inlet port. Next, charge the spring brake with air so the pushrod is in the released position. Make sure the pushrod is centered in the air chamber and not cocked to one side. Using a square, mark the pushrod at the 90 degree position with the short leg of the square flush along the pushrod and the long leg centered in the end of the s-cam as shown in the diagram to the right.



#### Figure B:

Next, measure from the 90 degree mark back towards the air chamber the "X" distance (see the chart in the second diagram to the right). Mark the pushrod at the "X" dimension and then cut the pushrod at this mark. The clevis for your manual slack adjuster is now ready to be installed on the pushrod.



**For automatic slack adjusters, please refer to the slack manufacturers installation instructions.**

Cutting the pushrod correctly is imperative with auto slacks. All of the common brands of auto slacks in the market today use a double pin clevis assembly except for Haldex. If the pushrod is cut correctly both the large and small clevis pins should install easily by hand. If the pushrod is too long the small clevis pin won't install without the adjusting rod on the auto slack being pried up. If the pushrod is too long when a brake application is made the slack adjuster will go well beyond 90° to the pushrod and pull up on the adjusting rod farther than it was designed to. Eventually the adjusting rod will break where it attaches to the adjusting mechanism inside the auto slack. After that the auto slack will not function and cannot hold adjustment. Also a slack adjuster does just what its name says; it adjusts to take the slack out of the brake assembly as the components wear. An auto slack will always adjust back to its original installed position. If the auto slack is installed wrong it will stay wrong.

**S-Cam Head** — Our second lever is the s-cam head. We'll look at the profile for the original standard s-cam head used with 4515 shoes and the enhanced s-cam head used with 4707 shoes. Based on the equation  $c = \pi \times d$  where "c" is the circumference, " $\pi$ " is 3.14 and "d" is the diameter, we can calculate the circumference of both circles to determine the amount of lift at the brake shoe roller. For both cam head profiles the circumference of the circle at the slack arm clevis pin is as follows — 6" slack arm drilling (the radius) x 2 = 12" diameter or  $3.14 \times 12 = 37.68$  circumference. Now we'll calculate the circumference of the circle at the point where the brake roller normally sets when the brakes are released.

continued on page 7

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### S-Cam Head — continued

As a side note here, I get a fair amount of phone calls about brake rollers that don't sit all the way down in the "pocket" of the cam head. Customers tell me they can see a "little daylight" between the bottom of the brake roller and the pocket of the s-cam head. Actually the pocket is there to allow the brake shoes to be retracted far enough to slip the brake drum on. That's why the depth of the pocket on the standard 4515 head is 0.79" and the enhanced 4707 head is 1.09" or a difference of just over 1/4". 4707 shoes have thicker friction material but more about that in a bit. If the foundation brake is in good shape with new brake shoes and a new brake drum, when you're finished adjusting the brakes, the brake roller will normally sit right at the beginning of the incline on the s-cam head, not down in the pocket.

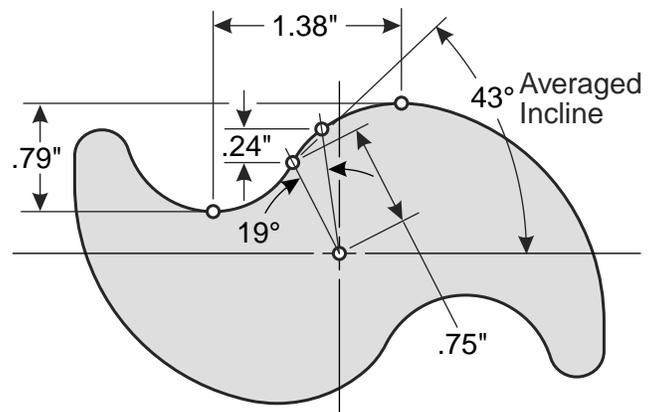
Now back to calculating the circumference of the circle where the brake roller sits when the brakes are released. Let's look at the standard 4515 s-cam head first.

For the standard 4515 s-cam head the radius is 0.75" so the diameter would be 1.50" ( $0.75" \times 2 = 1.50"$ ). Using our  $c = \pi \times d$  formula we get  $3.14 \times 1.50 = 4.71$  and  $4.71 \div 37.68 = 0.125$  so our ratio from the circumference of one circle to other is 0.125. Therefore 2.0" of pushrod stroke times our ratio of 0.125 we'll get us 0.25" of rotation at the s-cam head from where the brake roller sits ( $2.0" \times 0.125 = 0.25"$ ). That 0.25" of cam head rotation on the incline of the cam head will get us 0.24" of lift. Naturally the brake shoes don't sit 1/4" away from the drum when the brakes are released but what if the foundation brake is *not* in good shape. We'll come back to that in the next edition.

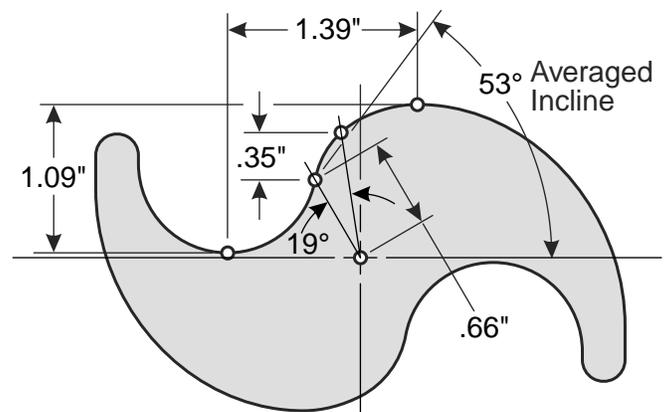
Now let's take a look at the enhanced 4707 s-cam head.

For the enhanced 4707 s-cam head the radius for where the brake roller sits is slightly lower at 0.66" due to the deeper pocket so the diameter would be 1.32" ( $0.66" \times 2$ ). Again using our  $c = \pi \times d$  formula we have  $1.32" \times 3.14 = 4.145"$  or 4.15" and  $4.15" \div 37.68 = 0.11"$  so our ratio this time 0.11". Therefore 2.0" of pushrod stroke times our ratio of 0.11" we'll get us 0.22" of rotation at the s-cam head from where the brake roller sits ( $2.0" \times 0.11" = 0.22"$ ). However this time that 0.22" of s-cam head rotation gets us 0.35" of lift. How's that? The incline of the 4707 s-cam head is steeper. The averaged incline of the 4515 s-cam head is 43° while the averaged incline for the 4707 s-cam head is 53°. A 10° increase in the incline of the s-cam head doesn't sound like much but look at the difference in the amount of lift, 0.24" compared to 0.35". That's an increase in lift of  $((0.24"/0.35") - 1) = .314$  or 31.4% which in the brake world is huge! The enhanced 4707 s-cam brake comes on much faster and harder than the 4515 s-cam brake (which is the real reason the change was made). Now let's take a look at the last of our three simple levers, the brake shoes.

Standard 4515 Head



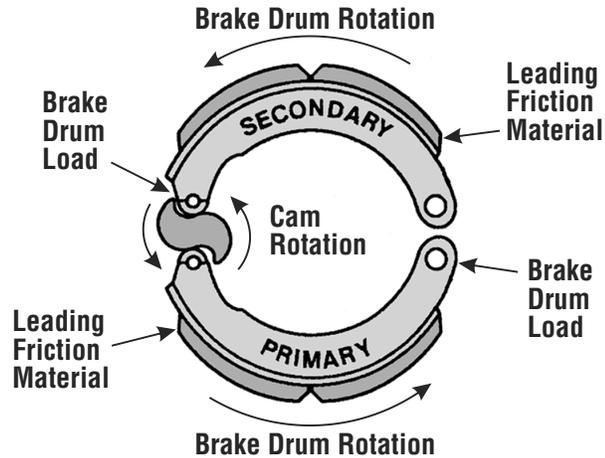
Enhanced 4707 Head



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**Brake Shoes** — S-cam brake shoes are a second class lever with one end fixed called the anchor end and the other end where the force is applied called the roller end. The output is between the two ends covered with friction material. The most common set-up is the cam forward of the axle on the horizontal center line with the cam head rotating in the same direction as the drum (as shown in the diagram below). This will also place the air chambers above the axle with the pushrods pointing forward.

When the brakes are applied the s-cam head rotates as we discussed a moment ago, lifting both brake shoes into the brake drum. The force being applied by the s-cam head is at the roller end of each shoe but when the friction material comes into contact with the drum a load will also be exerted on the shoes in the direction the drum is rotating. This load will impact the “trailing end” of each brake shoe which in this set-up is the anchor end on the primary shoe and the roller end on the secondary shoe.



**Primary or Bottom Shoe** — When the s-cam head rotates the primary shoe is pushed “out and in” which is why the primary or bottom shoe is rarely stretched. The leading friction material on this shoe is the cam block which is also the roller end of the shoe where the force from the s-cam head is being applied. This makes the primary or bottom shoe wear faster because it has both the force of the s-cam head and the load from the brake drum on it simultaneously. The anchor pin bushing for this shoe will also wear faster as it takes the load from the brake drum rotation.

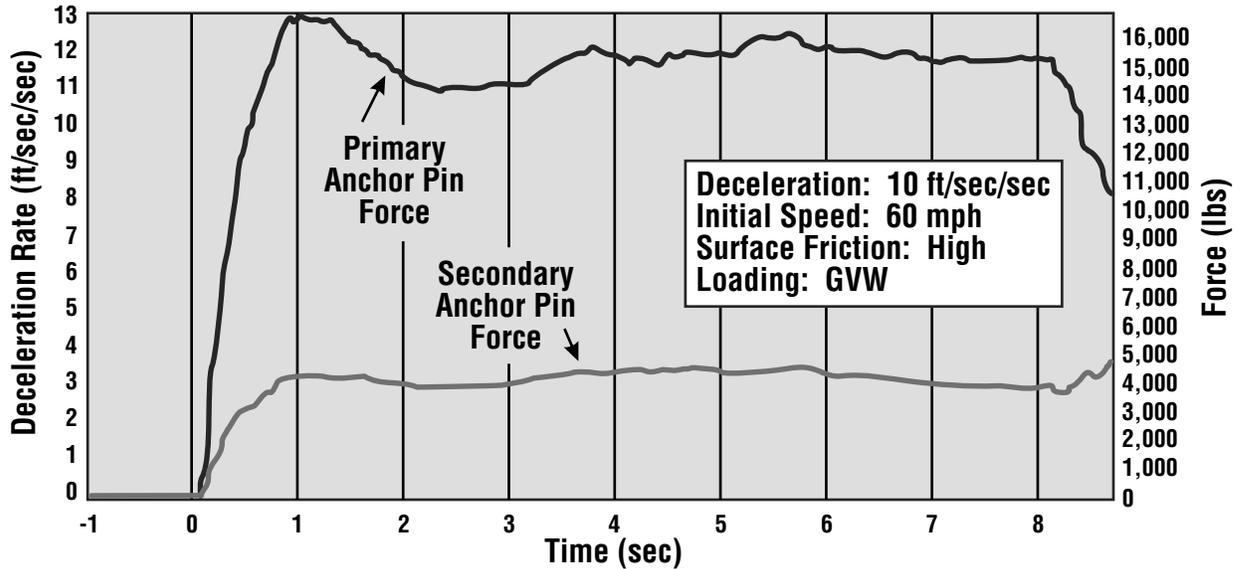
**Secondary or Top Shoe** — When the s-cam head rotates the secondary shoe is pushed “out and out” which is why the secondary or top shoe is usually stretched (this is why brake shoe reliners have “coining machines” because about half of the cores they get back are stretched). The leading friction material on this shoe is the anchor block which makes the anchor end want to “kick up” when the friction material contacts the brake drum. This is why the retainer springs on 4709 shoes should always be installed with the coils up to absorb the anchor end of the shoe wanting to lift up off the anchor pin. The secondary or top shoe will not wear as fast because the force from the s-cam head is being applied to the trailing end of the shoe. This will however exert the load from the brake drum rotation onto the roller end of the shoe and therefore the s-cam head which does two things.

1. The spider bushing will wear on the bottom side of the camshaft. The s-cam head is what centers the brake shoes in the drum on the roller end so as the spider bushing wears the s-cam head will begin to drop. As the s-cam head drops the primary or bottom shoe will get closer to the brake drum and will increasingly contact the brake drum before the secondary or top shoe does which makes the primary or bottom shoe wear even faster.
2. Sliding brake rollers will usually be found on this shoe because the roller end takes the load from the brake drum rotation. If the brake assembly is overworked the roller end can even “mushroom out” in the brake roller opening.

In June of 2008 FMCSA (Federal Motor Carrier Safety Administration) did a study on brake performance monitoring. Part of the study was measuring the load put on the anchor pins when the brakes were applied. They found the load on the anchor pin for the primary shoe was three times that of the load on the anchor pin for the secondary shoe as shown in the graph shown on page 9.

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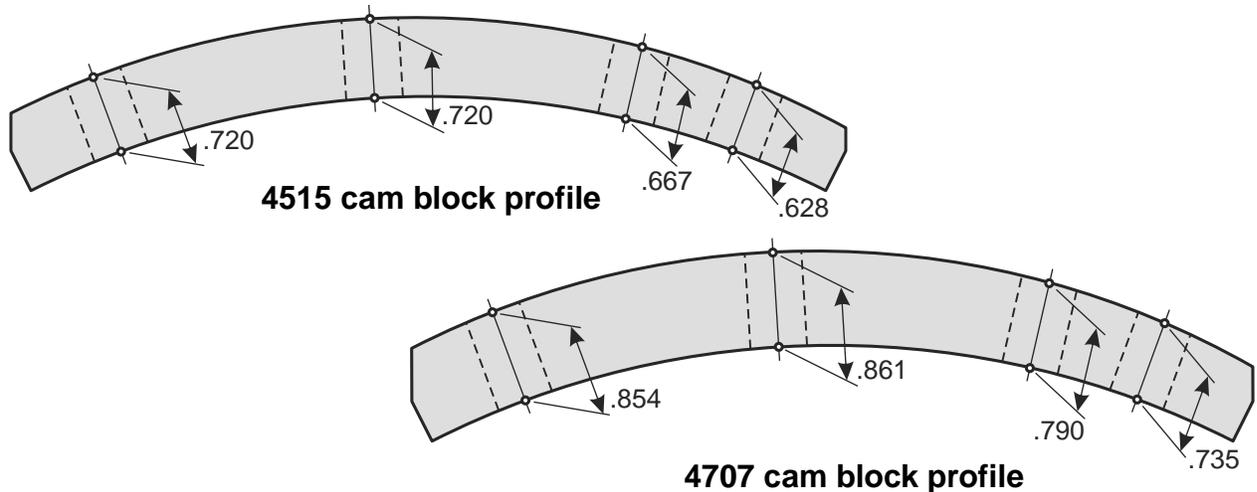
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If you take this data and flip it for the roller end then the load on the brake roller for the secondary shoe would be three times that of the load on the brake roller for the primary shoe.

Now let's take a look at the friction material and the differences between the 4515 and 4707 cam blocks.

**Friction Material** — When the s-cam heads were changed in the late 80's/early 90's, the brake shoes also changed. The 16.5" x 7" 4515 brake shoe which had been the main stay for decades was replaced with the 4707. The 4707 shoe had a slightly different brake block drill pattern so as not to confuse it with the very similar looking 4515 shoe (still lots of brake shoe reliners back then). Also the 4707 friction material was thicker as these new "enhanced" s-cam brake assemblies were going to run so many more miles. Let's do some measuring and see what we get. Here are profiles for 4515 and 4707 cam blocks with the thickness measured at the four rivet holes.



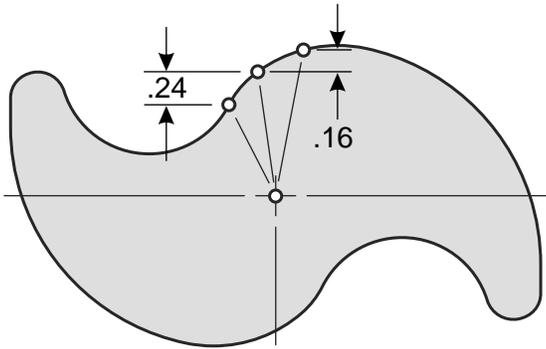
First we need to deduct 0.25" from each measurement for the amount of block that should be left on the shoe when it's replaced. Next we'll calculate the difference in thickness between the 4515 and 4707 cam blocks.

<b>4515</b> — .470"	<b>4707</b> — .604"	$(.604" - .470") / .604" = .222$ or 22.2%
.470"	.611"	$(.611" - .470") / .611" = .231$ or 23.1%
.417"	.540"	$(.540" - .417") / .540" = .228$ or 22.8%
.378"	.485"	$(.485" - .378") / .485" = .221$ or 22.1%

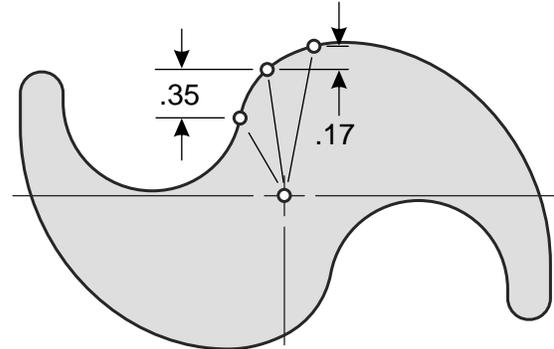
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The results tell us that the 4707 cam block has 22% more useable friction material than the 4515 cam block did. Both s-cam heads have a total lift of 0.50" built into their design to compensate for wear which is mostly in the friction material. As the friction material and other components wear the auto slack will adjust and rotate the s-cam head to keep the brake shoes running at a 0.015" to 0.020" clearance from the brake drum. Remember, when we looked at the first half of s-cam head rotation the lift for the 4707 head was quite a bit more than the 4515 head, 31.4%. Now let's look at the second half of s-cam head rotation for both heads.

Standard 4515 Head



Enhanced 4707 Head



Here again we see the lift for the first half of s-cam head rotation at 0.24" and 0.35" respectively. Now look at the second half of s-cam head rotation, 0.16" of lift for the 4515 head and 0.17" of lift for the 4707 head. For all practical purposes they are identical. What's up with that? The steeper incline in the first half of s-cam head rotation for the 4707 brake will wear away a lot of that "extra" friction material. In trying to improve the original 4515 s-cam head they actually took the smooth progressiveness out of the design because you can't improve on its simplicity. The s-cam brake does have some inherent wear issues in its design but it's a 70 year workhorse that has stood the test of time. Here's L Ray Buckendale's drawing for the original s-cam brake with cast brake shoes filed with the US patent office in 1947. As you can see very little in the basic design has changed.

April 15, 1952

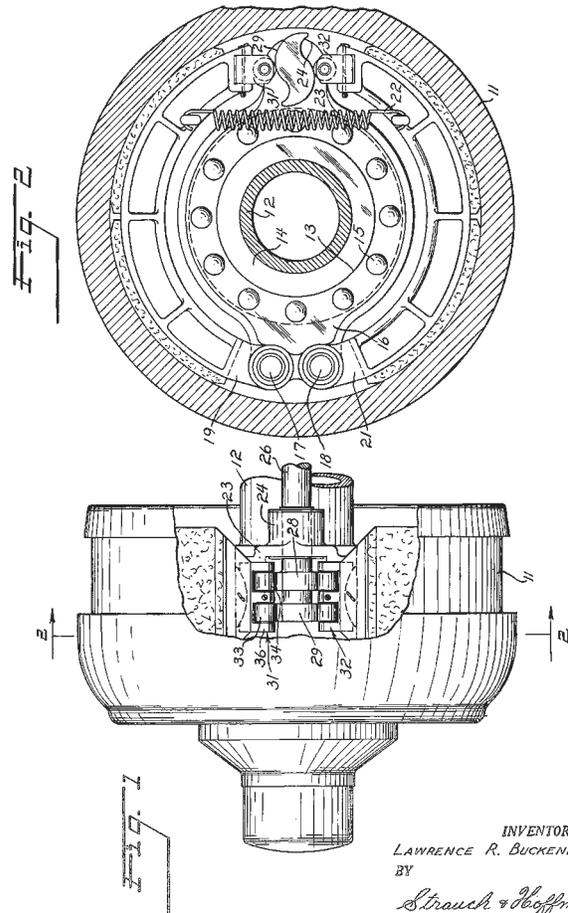
L. R. BUCKENDALE

2,592,536

COMPENSATING BRAKE CAM FOLLOWER

Filed Aug. 7, 1947

3 Sheets-Sheet 1



As always, I hope you found this edition of WHEN informative.

Regards,

*Steven S. Wolf*

Steven S. Wolf  
Axle Group Product Manager  
Dayton Parts, LLC

In the next edition of WHEN we'll look at the air disc brake system and what lies ahead.

INVENTOR.  
LAWRENCE R. BUCKENDALE  
BY  
*Strauch & Hoffmann*  
ATTORNEYS