3

TIRES - Part Two

A Race Engineer's Look at Tire Performance

(It's just not this simple!)

The Bottom Line

It all begins and ends at the four tire contact patches. Everything a race car designer, engineer and driver does should always be focused on producing the highest, the most balanced and most consistent coefficient of friction possible from the four contact patches. To that extent, every word, concept and theory in this book and the four accompanying books is geared towards helping the racer optimize those four little contact patches. *The racer who loses sight of this fact will never consistently run at the front, period*. You may think this is a motherhood and apple pie type of statement which will always be foremost in the race engineer's mind, but the fact is it's easy for all of us to lose sight of the basics when busily running reams of calculations on the newest flavour of the week's magic tweak.

As stated in Chapter One, getting a race car to work at its optimum level is all about understanding how and why the action and reaction forces work, how they are developed and how these forces are transferred from point to point. The loads are constantly and simultaneously being transferred from the road through the tires into the chassis and from the chassis through the tires to the road. This is a fundamental point. These equal and opposite forces are not easily broken down into simple static components let alone their dynamic components. It is not so simple as stating that when you place a known load on the track, the track will exert an equal and opposite force. Race cars are made up of thousands of components and each of these components will affect how the myriad of loads and forces are transferred back and forth from the track through the chassis and back again to the track.

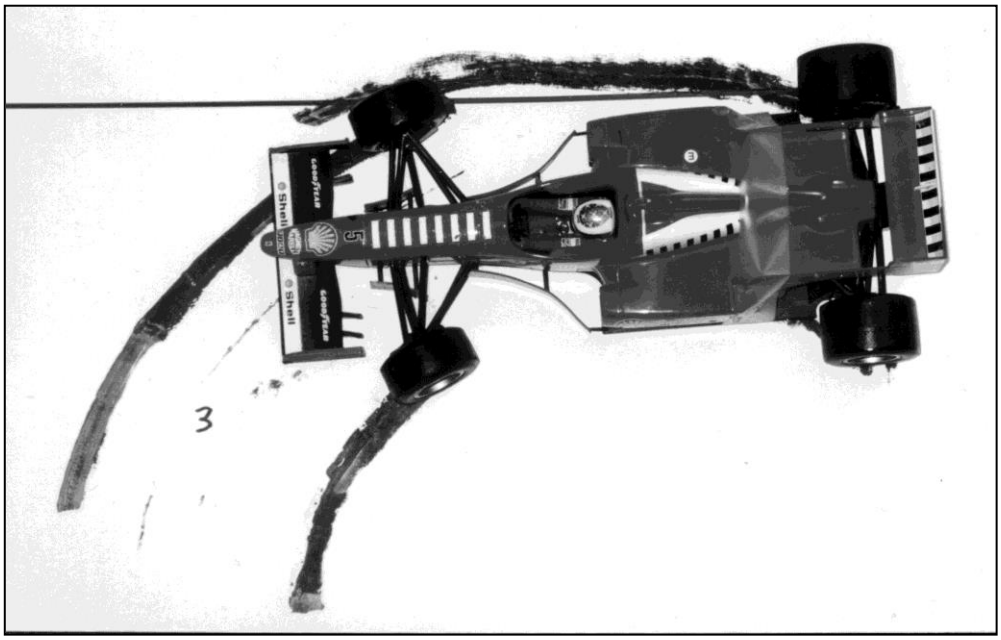


Figure 3-1

Yaw - The First Look

"A race car, like any material object, obeys Newton's First Law and will continue in a straight line at uniform velocity unless acted on by an external force. To turn the car, then, requires the lateral forces developed when the front wheels are steered. These centripetal ("center seeking") forces, transmitted through the suspension bits...etc...are what deflect the vehicle from its straight path. The so-called centrifugal or outward force is actually just the First Law impelling the vehicle and all its parts (driver included) to continue on the straight path." William L. Ball/ Senior Engineer/ General Motors

Action and Reaction Forces - When a tire is turned (steered), it builds a series of lateral forces that cause it to laterally accelerate – Figure 3-2. These centripetal (center seeking) forces are transferred from the steered tire, for the most part, through the suspension control arm bits into the chassis (lateral loading). As a result, the chassis reacts to these forces, and, in turn, produces equal and opposite centrifugal (away from the center) reaction forces which try to push the chassis away from the intended path (radius). These centrifugal forces are transferred, for the most part, back to the tires via the springs, bars and dampers on through to the hub, rim, tire sidewall, and, finally, the contact patch. The phrase, "for the most part," is used because, depending on the suspension kinematics, a percentage, normally the largest percentage of the centrifugal reaction forces transfer via the springs, bars and dampers while the remaining percentage moves through the suspension wishbones/A arms etc. As we will soon discover, if the roll rate is stiffened and/or the heights of the front view instant centres are increased, the percentage of loads and forces which pass through the suspension components back to the contact patch will increase.

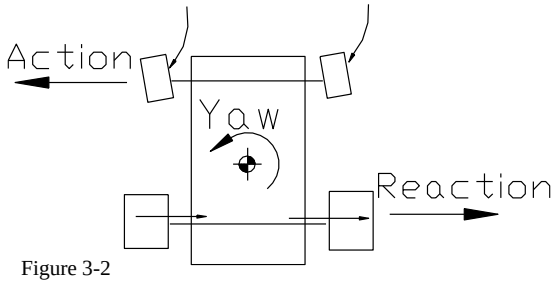


Figure 3-2

The lateral action forces produced as the result of the front tires being steered begin the process of laterally accelerating the front of the chassis while it is primarily the chassis reaction forces at the rear tires which try to fling the rear of the chassis out away from the driver's intended path (the rear of the chassis wants to yaw around the front of the chassis - the car wants to swap ends). Luckily, as the rear of the chassis begins to swing out, shear forces between the rear tires and the track begin to develop their own lateral forces at the rear contact patches, which, in turn, keep the rear of the chassis from swinging out away from the turn (centrifugally accelerating) - Figure 3-3. If the rear tires did not build these reaction lateral forces, the car would spin as if on ball bearings (yaw uncontrollably). It only takes a very slight imbalance between the centripetal and so called centrifugal forces to produce chassis yaw. It is the front to rear balance between these two forces that determine the chassis' handling characteristics. The extent of the action and reaction forces on the front and rear tire contact patches produce either a balanced car, or, in turn, cause the chassis to understeer (push) or oversteer (loose). It is also this balance of action and reaction forces at each individual contact patch, that, to a great extent, determine the point about which the chassis yaws.

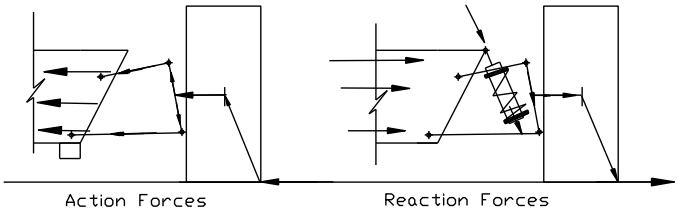


Figure 3-3

If the action and reaction forces at both the front and rear of the chassis are in balance, the chassis will neither laterally accelerate or yaw - Figure 3-4. Figure 3-5 shows the chassis yawing about the front axle. In this case, the front action (centripetal) and reaction (centrifugal) forces are balanced while the sprung mass outward forces at the rear tire are greater than the tire's ability to balance the chassis' mass reaction forces. Having the chassis rotation point move all the way to the front axle centerline would represent an extreme rear out case. Under rear out conditions the yaw point would be somewhere between the center of mass (gravity) and the front axle center line. A slight rear yaw out condition would have the yaw point near the center of mass.

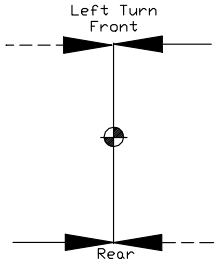


Figure 3-4

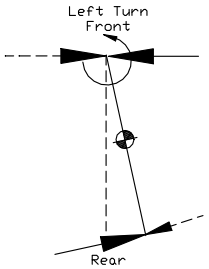


Figure 3-5

Figure 3-6 represents a chassis with a front in yaw condition. In this case, the rear tire forces are in balance while the centripetal forces at the front tire are greater than the centrifugal forces. In this extreme condition, the chassis would yaw about the rear tires. A front yaw in condition would normally see the chassis yawing about a point somewhere between the center of mass and the rear axle centerline. A slight front yaw in condition would have the yaw point near the center of mass. Figure 3-7 shows a chassis in a balanced steady state yaw condition. As discussed further in later chapters, a chassis under race conditions, seldom, if ever, yaws (rotates) about its center of gravity. *It's just not that simple*.

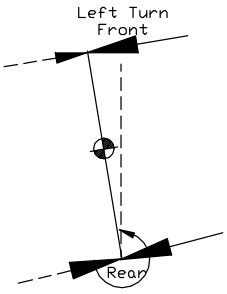


Figure 3-6

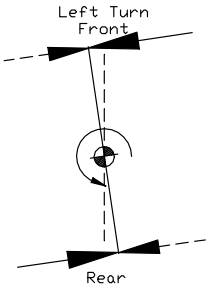
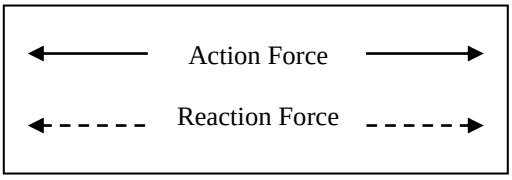


Figure 3-7



Ultimately, a tire's coefficient of friction is the result of the combined ground to tire and tire to ground acceleration forces. As previously stated, the highest percentage of ground to tire lateral force is generally transferred to the chassis through the suspension links while a higher percentage of chassis to ground force is primarily transferred from the chassis to the tire via the springs, bars and dampers. As we will soon see, this last point plays a key role in how a chassis is designed and engineered. Sprung mass reaction forces passing through the suspension components react at the tire contact patches quite differently than those forces passing through the springs, dampers and bars i.e. the suspension forces generally have a higher horizontal force component while the forces passing through the springs, dampers and bars generally have a higher vertical force component. It is both the magnitude and direction of the combined action and reaction forces that produce the endlessly varying contact patch and sidewall configurations. As just discussed, sprung mass reaction forces passing through the suspension components generally are more horizontal in nature, and, as such, are seen at the contact patch as an impact shear force - not a good thing. It only takes a few milli-seconds of impact shear force at the contact patch to upset the chassis balance. The race engineer wants any resultant sprung mass reaction force to contain the highest vertical component possible.

Vertical action forces as the result of bumps or banking generally travel from the contact patch through to the chassis via the springs, bars and dampers.

Rate of Load Transfer

- Load transfer passing through the suspension control arm bits, ignoring the compliance factor, is close to being instantaneous

- The rate of load transfer passing through the elastic bits (springs and bars), can, to some degree, be controlled by the rate versus motion settings of these devices

- The rate of load transfer passing through the suspension dampers, can, to some degree, be controlled by the force versus velocity settings of these devices

- The rate of load transfer passing through the tires (visco–elastic system) is controlled via the tire's design and construction. A tire with a short aspect height will generally respond to a change in steer angle faster than a tire with a tall aspect height. Additionally, when a tire manufacturer decreases a tire's sidewall stiffness rate, race teams will often need to change their set ups to compensate for the tire's increased response time to steer angle changes. A tire with a soft sidewall rate and tall aspect height will often hunt more (tire trying to find a point of balanced equilibrium) than a tire with a stiff sidewall rate and a short aspect height. In this last example, the first tire will generally require a quicker rack and pinion or steering box rate.

Rate of Load Transfer

- Resultant **action** load transfer from a steered front tire primarily moves from the tire to the chassis through the suspension control arms

- Resultant **reaction** load transfer generally moves from the chassis to the rear tires through the suspension's elastic bits

- The front end of the chassis will generally react faster to a change in lateral acceleration than the rear of the chassis due to the fact that the action forces at the front are primarily moving through the suspension control arms while the reaction forces at the rear of the chassis are primarily moving through the suspension elastic bits. The softer a tire's sidewall rate and the taller its aspect height, the more the front and rear reaction times will be balanced out.

Studying how a sidewall flexes will tell you a great deal about the types of loads and forces the tire is experiencing. As in so many instances, high quality, high speed digital cameras are providing the race engineer with a whole new way to look at chassis performance. *Key tuning tool*. The amount of sidewall deflection may have as much to do with lateral impact loading (forces) as it does with the tire's load versus coefficient of friction numbers.

Reaction forces transferred from the chassis through to the tires do not produce inward (centripetal) acceleration but rather centrifugal forces that may produce yaw. No matter how great the shearing forces, chassis reaction forces can not produce inward lateral acceleration although they can certainly affect the efficiency of the lateral acceleration forces.

When studying tire performance parameters, it is always important to remember that shear forces, as the result of steering input, start at the surface of the tire's contact patch and migrate up through the sidewall into the rim while chassis reaction forces migrate from the rim down through the sidewall to the contact patch. Centripetal forces are moving in towards the center of the arc while centrifugal forces are moving outward away from the center of the arc.

- Centrifugal forces are trying to increase the radius of the arc.
- Centripetal forces are trying to decrease the arc radius.

Lateral Acceleration Versus Yaw - Cornering is normally a combination of lateral acceleration and yaw. A car can laterally accelerate without yawing i.e. four wheel drift - Figure 3-8, but it can not negotiate a turn without also having some degree of yaw. The smaller the turn radius, the more the chassis will need to yaw to remain balanced - large turn radiuses require very little yaw in relation to the amount of lateral acceleration. This is one reason why speed corrected steering plots are used during data analysis as discussed in Book Four. If the rate of yaw acceleration is too great, the car will be in an oversteer/loose condition – Figure 3-9. If the rate of yaw acceleration is too little, the car will be in a understeer/push condition. There is a fine line between loose and spinning. If the rate of yaw acceleration is such that it produces an oversteer/loose condition and if that rate of acceleration continues for other than a very short period of time, the car will move from loose to spinning.

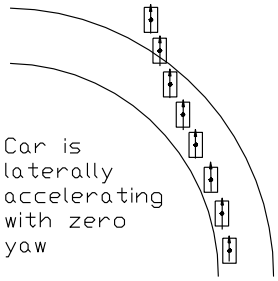


Figure 3-8

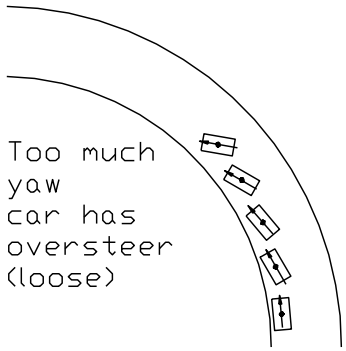


Figure 3-9

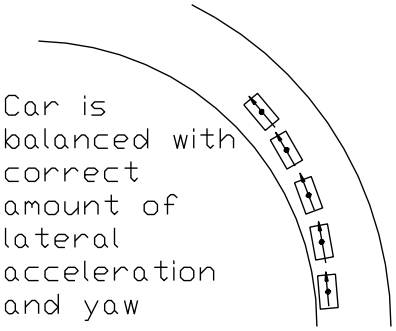


Figure 3-10

Definitions

- Lateral Acceleration - Lateral acceleration equals the rate of lateral (sideways) motion and change in speed.
- Yaw Angle - Yaw angle equals the amount the chassis has rotated.
- Yaw Acceleration - Yaw acceleration equals the rate of rotational change.

A solid rear axle or independent suspension system can also produce steer (lateral acceleration) but only when offset through kinematic motion - Figure 3-11. *Key tuning tool for live axle cars*.

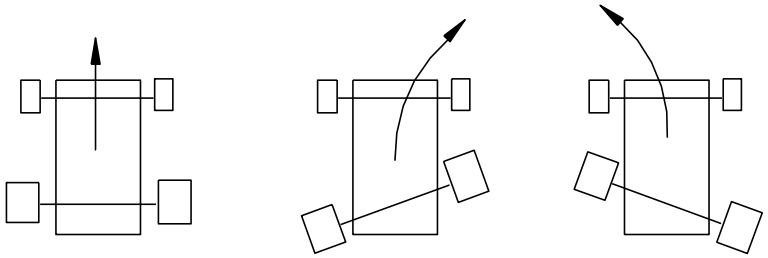


Figure 3-11

Race Tire Parameters

Each type of racing tire has a particular set of parameters which determine its coefficient of friction in relation to the percentage of combined shear angle. Remember that the coefficient of friction determines cornering force, which, in turn, means that cornering force varies with shear angle. For the tire in Figure 3-12, the coefficient of friction increases quickly from zero up to a maximum of eight and a half degrees of shear angle. At this point, the curve drops down slightly, flattens out and quickly begins to decline. A race driver is always trying to hold the car at this peak coefficient of friction. If the shear angle is held at a point where the coefficient of friction begins to drop, the tire will begin to lose grip as the contact patch develops an increasingly higher percentage of slip to the point where the tire begins to slide. The tire can still generate a certain amount of grip through hysteresis and conventional friction, but, as the slip angle continues to increase, the amount of grip drops and the amount of sliding increases. Heat generation, as the result of increases in shear angle, slip and amount of tire sliding is much higher on the right side of the graph for an equal amount of lateral force.

Slope - The slope angle represents the tire's cornering stiffness, which, in turn, determines the tire's sensitivity to the steering input and shear angle – Figure 3-12. A steeper slope equates to a stiffer tire, which in turn, determines the steer angle to force coefficient. A stiffer tire requires a smaller shear angle to produce an equal amount of lateral force and, in turn, generates less heat for an equal amount of force. Tires with smaller aspect ratios generally have steeper slopes.

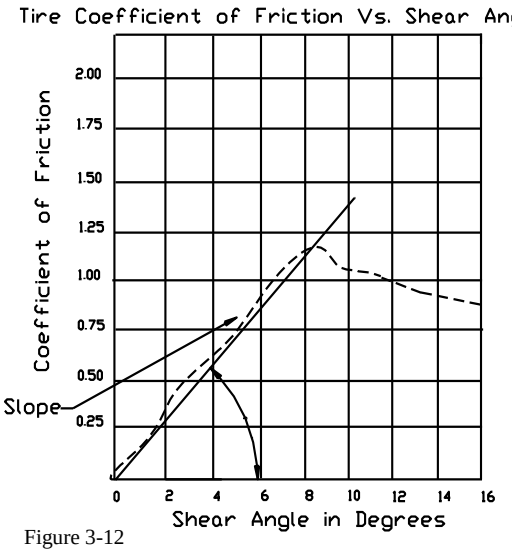


Figure 3-12

Stiffer tires are much less forgiving, move very quickly from the peak force coefficients and are bad news for the less talented driver. Conversely, in the hands of a talented driver, they are capable of producing much higher sustained coefficients with lower heat generation. As many tire designs are geared towards the brilliant driver, it's no wonder that the less talented drivers have trouble adapting to them. Stiffer tires react much faster to steering input.

As was previously explained, once the front tires are turned and the lateral forces begin to build at the leading edge of the contact patch, the resultant centripetal action force at the front of the car and centrifugal reaction force at the rear of the car, if not perfectly balanced, will begin to rotate the car. If the rear tires have no coefficient of friction or grip, the rear would quickly swing away from the center of the turn and spin out. However, as introduced earlier in the chapter, once the car begins to rotate around its center of mass, the rear tires begin to react to the chassis' reaction forces and start to develop their own slip angles and cornering forces. If the front and rear cornering forces balance out, the car's handling will be neutral and the car will neither understeer nor oversteer for a given radius. The tire's tread compound, construction, chassis geometry and chassis weight distribution all affect the lateral force versus slip angle coefficient. A driver's talent level also has a huge effect on the lateral force versus slip angle coefficient.

Camber and air pressure under various conditions may also play a significant role in the tire's lateral forces versus slip angle. Tables 3-1 & 3-2 and their accompanying graphs illustrate the changes in corning forces versus load at 0 and 4 degrees of camber. Tire pressure is set at 24 psi. Camber forces are in lb. units while steer angle is in

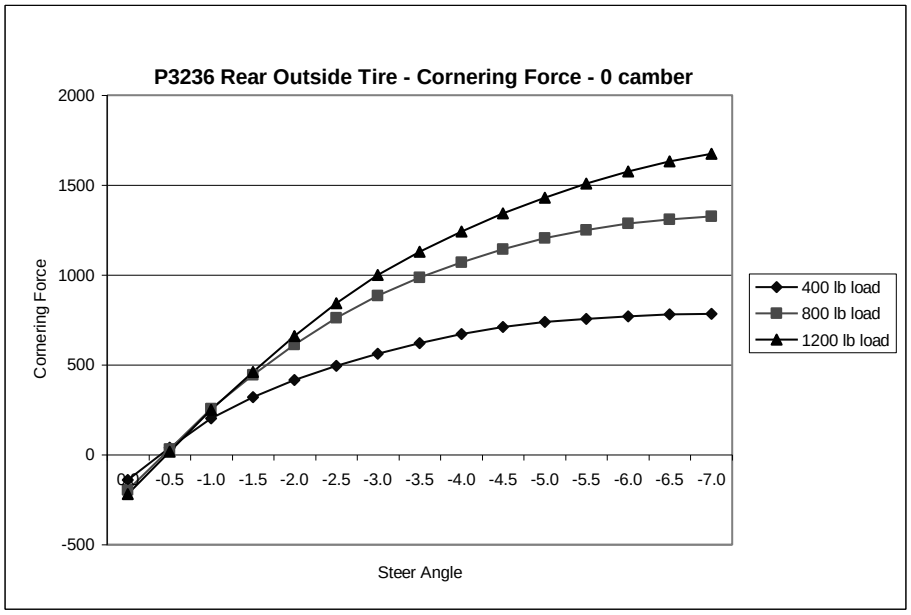
degrees. A negative number under cornering force indicates the tire is trying to steer towards the outside wheel. *Key tuning tool.*

Table 3-1

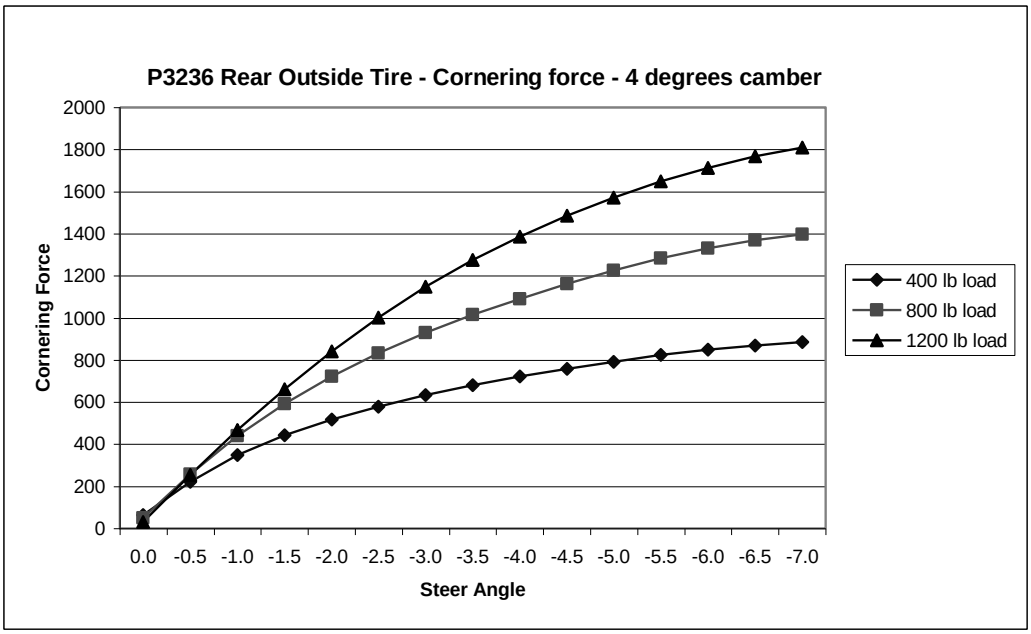
Pressure	24 psi		
LOAD	400 lb.	800 lb.	1200 lb.
CAMBER	0º	0º	0º
Steer Angle	Cornering Force		
0.0	-139.868	-196.903	-218.835
-0.5	37.97935	29.70373	16.31828
-1.0	202.6453	255.2412	249.3232
-1.5	321.1988	444.962	460.3868
-2.0	416.4894	613.1692	660.1371
-2.5	493.5272	759.8389	842.7089
-3.0	560.9679	883.9064	998.99
-3.5	621.3735	985.3144	1128.638
-4.0	671.3503	1071.388	1241.535
-4.5	710.2757	1143.459	1342.278
-5.0	738.8672	1205.69	1430.481
-5.5	756.2787	1250.229	1508.227
-6.0	770.3057	1285.38	1575.796
-6.5	779.7841	1309.869	1631.025
-7.0	782.9925	1325.755	1672.793

Table 3-2

Pressure	24 psi		
LOAD	400 lb.	800 lb.	1200 lb.
CAMBER	4º	4º	4º
Steer Angle	Cornering Force		
0.0	64.11332	50.75638	30.17907
-0.5	219.9855	256.6909	253.3256
-1.0	347.2711	440.223	468.4437
-1.5	441.6585	592.6823	660.3844
-2.0	517.4603	721.0562	839.7856
-2.5	579.2873	833.0596	1000.197
-3.0	633.3377	928.4453	1146.953
-3.5	679.9254	1014.129	1276.168
-4.0	721.9165	1090.53	1385.129
-4.5	759.1258	1161.184	1485.477
-5.0	792.3043	1226.044	1572.217
-5.5	823.1631	1283.968	1647.643
-6.0	848.0517	1331.274	1711.553
-6.5	868.5481	1368.892	1767.104
-7.0	884.4719	1396.618	1809.756



Graph from Table 3-1



Graph from Table 3-2

Vertical versus Normal Load - Normal load is seen as perpendicular to the tire. For a flat track and zero camber angle vertical and normal load represent the same thing, and, for this reason, the terms vertical and normal load are often interchanged.

Vertical or normal load is the total downward force applied to an individual tire. This load or force is applied in a direction or plane perpendicular to the road surface – Figure 3-13. When the car is stationary, this load is equal to the static weight on each corner of the car. In the dynamic state, these forces are constantly being transferred from one tire to another due to continuous lateral (transverse) and longitudinal (fore/aft) load transfer. It is this continuous change that makes the art and science of balancing a car's handling such a challenge. Changes in loads due to banking are discussed in Chapter 14.

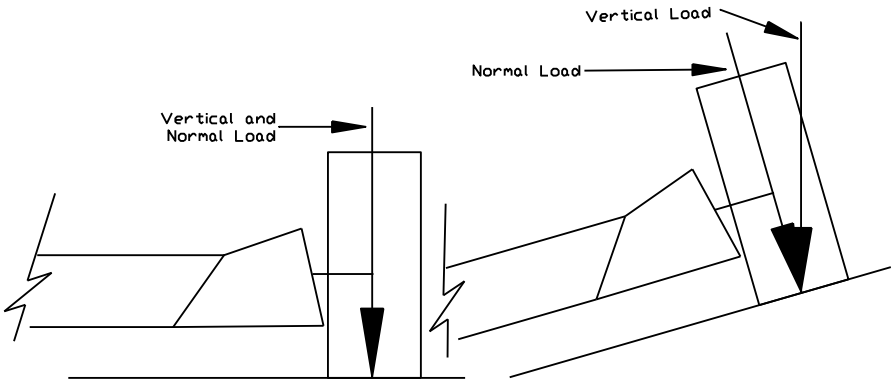


Figure 3-13

Decreasing Coefficient - A tire's coefficient of friction decreases as the vertical load on the tire is increased. On the other hand, up to the tire's ultimate traction capacity, an increase in vertical load will actually increase a tire's traction or grip. As weight on the tire is increased, the traction/tire force also increases, but to a lesser and lesser extent. As more weight is added to the tire, it becomes less and less efficient.

Each type of tire will have its own performance curve and some will be much better than others; however, they will all have a curve that shows a smaller increase in traction as vertical load is added - Figure 3-14. When it comes to lateral load transfer and cornering force, this is not a good thing.

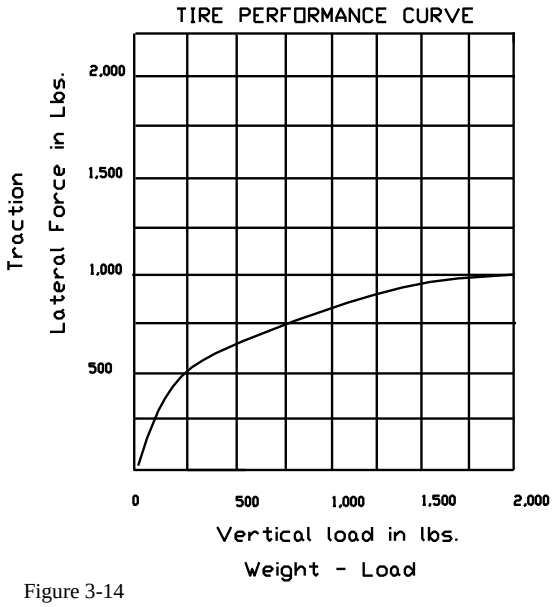


Figure 3-14

Table 3-3 shows a number of results from the Figure 3-14 graph.

Vertical Load vs Percent of Cornering Efficiency				
Vertical Load (lb.)	Lateral Force (lb.)	Traction/Load	Factor	Percent Efficiency
250	500	500/250	2	200%
500	680	680/500	1.36	136%
1000	850	850/1000	0.85	85%
1500	900	900/1500	0.6	60%
2000	1000	1000/2000	0.5	50%

A car's cornering ability is severely hampered when the heavily laden wheel gains traction slower than the lightly laden wheel loses traction. This results in the car losing lateral cornering force. A thorough understanding of the relationship between vertical load and traction is the key to making a car handle under all types of dynamic weight transfer. *A key tuning tool*.

Load versus cornering force efficiency has increased significantly over the past few years which has led to a considerable increase in speed for most race series.

Slip and Regrip - The Two Types - Another look at Chapter Two Concepts

Trailing Edge Slip - Slip at the trailing edge of the contact patch.

- As the tire rotates, the vertical **load** at the trailing edge of the contact patch is removed and that portion of the tread quickly snaps or slips back to its at rest position. Heat is generated by this slip action but it does not play a major role in a tire's heat build up until more than 10% of the contact patch has trailing edge slip.

- Once past its peak coefficient of friction, the slip region begins to expand forward. As grip is lost and the tire begins to slide, heat builds exponentially.

Contact Patch Slip - Slip throughout the contact patch.

- Depending on the design parameters, race tire tread particles are only capable of between 1 and 8 degrees of deflection and peak shear angles are continually getting smaller as tire construction and aspect ratios change. Once the tread particles have reached their shear limit, they begin to slip back to their at rest position. Obviously, if the wheel is turned to an angle of 11 degrees and the tread is only able to shear to 7 degrees, slip will develop. As a tread particle begins to slip back to its at rest position, it also quickly begins to lose its return forces (stored energy). At some point in this slipping process, the tread particle will once again regrip and the whole sequence will begin again. This process is more or less continuous and begins long before the tire reaches its maximum coefficient of friction. *It is far from this simple*.

Regrip - The process of slip and **regrip** allows the tire to achieve a much higher level of lateral force (higher coefficient of friction) than what it would otherwise be capable of. As in the example of the cleaning brush used in Chapter 2, each tread element pushes against the next, which, in turn, supports the first one. This process of one tread molecule leaning against its neighbor prevents the tread molecules from slipping all the way back to their at rest position.

As the peak coefficient is passed, the tread particles will no longer regrip as quickly and will not stay gripped as long. At this point, the tire will begin to slide and much of the molecular bonding will be lost. As vertical load is increased, the tread particles will deflect (shear) further before they begin to slip.

Longitudinal Forces

A tire's coefficient of friction, when considering longitudinal forces, is measured as percentage shear rather than shear angle. The "Percentage shear vs Coefficient of Friction" graph is much the same as the "Shear Angle vs Coefficient of Friction" graph. However, the longitudinal graph **generally** reaches its peak faster. It has a small bubble at the peak and a short flat area before it quickly falls off - Figure 3-15. *It's far from this simple.*

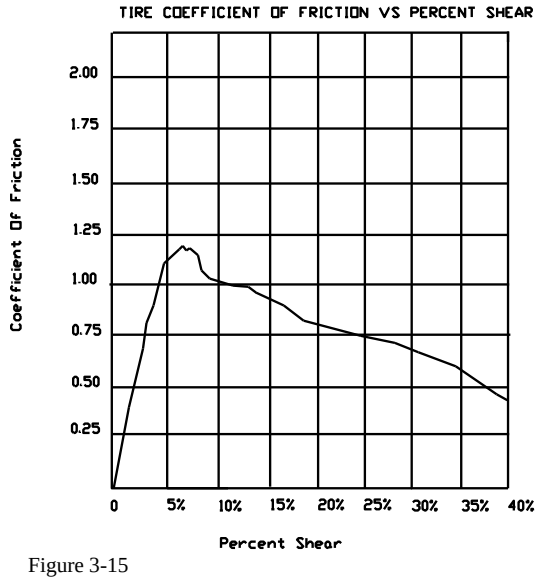


Figure 3-15

How the tire produces shear is the same for both lateral and longitudinal acceleration except that lateral acceleration shears (displaces) the tread laterally while longitudinal acceleration shears (displaces) the tread and sidewall longitudinally - Figure 3-16.

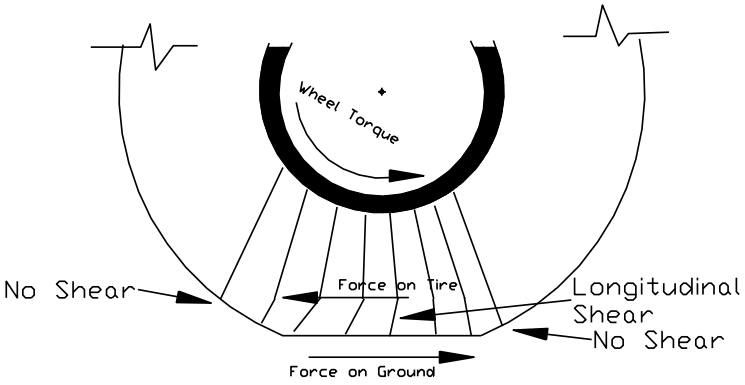


Figure 3-16

Longitudinal Shear - Inline Braking and Acceleration

Tread - The external surface of the tread will have a larger angle of shear than the inner portions, which means the outer tread particles will experience a higher rate of shear (they will move faster and further than the inner portions). Under forward acceleration, the outer tread particles are moving (shearing) faster but their road speed will be slower than the rest of the tire. While the outer tread particles are stuck to the track surface, the rest of the tire continues to move. As the tread molecules reach their elastic limit and/or as the tread particles reach the end of the contact patch and the normal load is removed, they slip or snap back to their original unstretched condition and once again move in sync with the relaxed tire particles - moving at the same speed.

Shear Ratio -The difference in speed between the sheared particles at the tread surface of the contact patch and the particles in the rest of the tread is referred to as the shear ratio. A 10% shear ratio at 100 mph road speed means that the tire assembly is actually rotating at 110 mph. This is far more difficult for data acquisition software to calculate than is generally susposed.

Longitudinal Slip - Slip realignment does not take place instantaneously. As the vertical load at the trailing edge of the contact patch decreases, the stored reaction forces start to overcome the action forces at which time the tread particles begin to snap or slip back into their relaxed state. This speed and shear realignment of the tread molecules is called slip. During inline acceleration, the amount of slip in relation to the contact patch size is the slip percentage. As illustrated in Figure 3-17, the relaxing of the tread particles continues past the trailing edge of the contact patch. This portion of realignment is referred to as spring back. As in the case of lateral acceleration, contact patch slip and regrip also takes place during longitudinal acceleration.

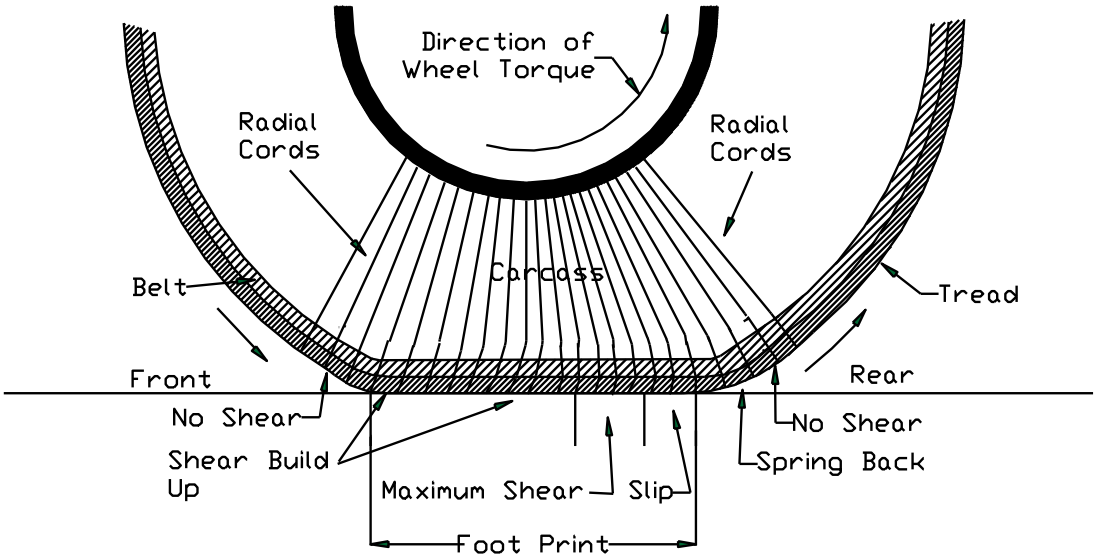


Figure 3-17

The belt (radial tire), sidewalls and carcass also twist (shear or wrap up) during inline acceleration, but at different rates than the tread particles, depending on the tire design and construction. Each part of the tire contributes to the tire's ultimate coefficient of friction, both laterally and longitudinally. The sidewall design plays a significant role in preventing the tire from breaking away during hard forward acceleration. Because the belt (radial tire) is much stiffer than the tread, it normally shears proportionately less. The sidewalls, on the other hand, even though they are normally much stiffer than the tread, may shear more than the tread particles because of the length of the cords. All things being equal, the shorter the side wall cords, the less they will shear. On the other hand, a shorter sidewall cord length will store a higher proportion of energy per degree of shear. This is also true of tread depth. The thinner the tread, the less it needs to shear for a given force but the thinner tread (depending on design) will also store a proportionately higher amount of energy per degree of shear. *It's far from this simple*.

Spring Buffer - The sidewall cords generally relax at a slower rate than the tread molecules and act as a spring buffer between the action and reaction forces, which, in turn, helps prevent break away shock between the tread and road surface (much as a rubber biscuit acts in a trailing arm to help prevent the loads from instantly being fed into the suspension joints).

As the tire reaches its maximum inline tractive capacity, the slip percentage (the amount of tread surface that is slipping) will have grown to approximately 30% of the contact patch area (depending on tire construction, etc.). At this point it takes very little additional tractive force to move beyond the tire's ultimate traction capabilities and into the spin mode. At a slip percentage greater than 50%, the outer portion of the entire tire will now be moving faster than road speed i.e. the tire is spinning. A higher percentage of slip builds a higher percentage of heat. High quality data acquisition software is very helpful in determining the optimum amount of shear and slip.

Vertical Loads On The Sidewall Cords - Action and reaction vertical loads are transferred through the tire sidewalls via the cords from the tread to the rim and from the rim to the tread. The sidewall design significantly contributes to the tire's load carrying capacity. As previously discussed, the tire's design, to a great extent, determines how and when the loads are transferred. The cords act as a series of springs compressing and extending under the combinations of vertical and resultant loads. This spring effect helps prevent abrupt shock loading by slowing the load transfer - Figure 3-18.

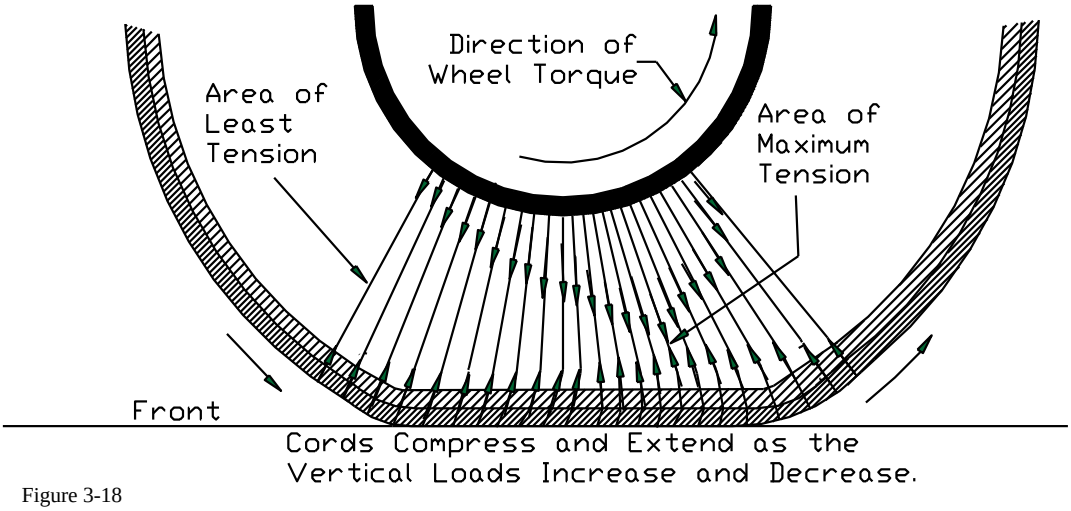


Figure 3-18

As in lateral loading, the tire will reach its maximum coefficient of friction at one particular percentage of shear. Once that peak percentage of shear has been exceeded, the tire will begin to lose traction. Great drivers are able to maintain their acceleration and braking forces right at the peak force while other drivers move from side to side of that force. Visible wheel spin or lock up shows that the peak shear percentage has been exceeded and the tire is below maximum traction.

In reality, the tire will spend much of its time in an intermediate state between lateral and longitudinal loading - Figure 3-19. Because this is a vector formation, the sum of the parts can be slightly greater than its individual components. In other words, the tire is capable of producing a greater overall resultant than its single lateral or longitudinal capabilities i.e. the circle is not actually a circle. However, for the most part, adding inline acceleration or braking while the chassis is cornering will degrade the car's cornering potential i.e. introducing inline acceleration during lateral acceleration will reduce both the lateral and longitudinal peak forces and decrease their individual slope angle. The reduction of lateral forces and slope will be closely proportional to the magnitude of the additional inline acceleration. Using the resultant to determine a tire's potential will be discussed in Chapter 19 i.e. traction (ellipse) concept.

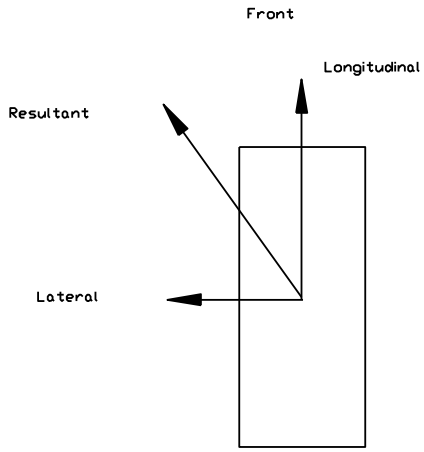


Figure 3-19

Just as in the case of lateral acceleration, adding vertical load will increase traction at the expense of the tire's coefficient of friction - Figure 3-20.

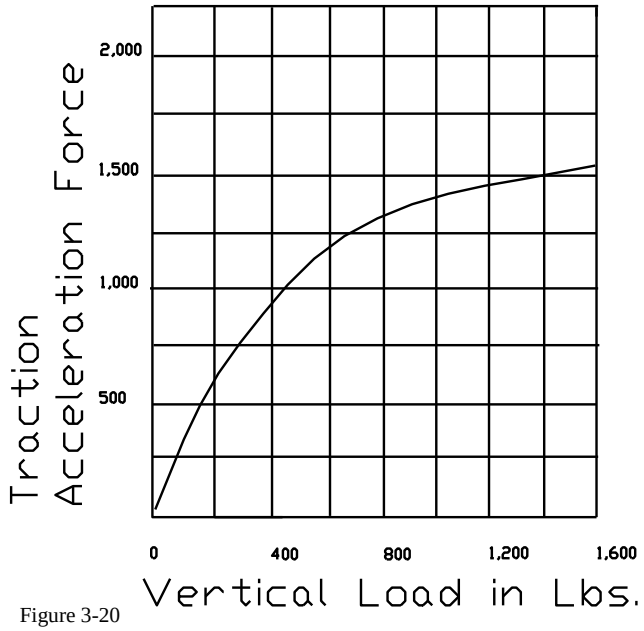


Figure 3-20

Figure 3-21 shows the relationship between coefficient of friction vs vertical load for a rear tire under longitudinal acceleration.

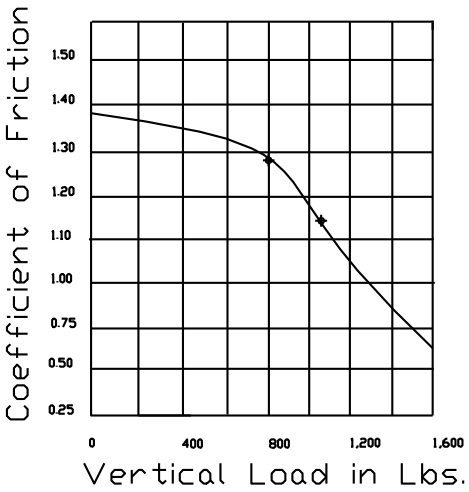


Figure 3-21

Some potential rear traction numbers can be determined from Figure 3-21.

- If both rear tires supported 800 lb. and had a coefficient of friction of 1.28, the two tires, given sufficient torque, would produce a force of (1.28 *800) x 2 = 2048 lb.

- If, through weight transfer, another 250 lb. is added to each wheel, the coefficient of friction is now only 1.15; however, the tire force is (1.15 *1050) x 2 = 2415 lb. So despite the fact that the coefficient of friction went down, the tire force actually increased. In this case, the more weight that can be **evenly** added to the rear tires, the better. Notice, however, that every time weight is added, the resultant proportional increase in traction goes down; as in the case of lateral loading, two tires equally loaded will produce more forward traction than two unequally loaded tires providing the total weight is the same in both cases.

- If the combined load on the rear tires was still 1600 lb. but the right tire carried 1000 lb. and the left only 600 lb., the right tire would have a coefficient of 0.56 while the left tire would have a coefficient of 1.34. 0.56*1600 = 896 lb. plus 1.34*600 = 670 lb., a total of only 1566 lb., 482 lb. less than in the first example. *Key tuning tool.*

Camber Forces

Note: The fact that camber can affect lateral acceleration is pretty well established. How it does so is still up to a considerable amount of debate, even amongst the tire engineers. The following text on the effects of camber angle are theoretical in nature not end all and be all statements.

Two Camber Components

- Non Vertical Resultant Elastic Forces

- Counteracting Centrifugal Forces

Non Vertical Resultant Elastic Forces - If a mounted and aired tire is dropped so that its tread surface strikes parallel to the ground, it will tend to bounce back in a straight up manner. If that same tire is dropped so its tread surface strikes at a cambered angle to the ground, it will tend to bounce back at a resultant angle closely related to the angle the tread struck the ground i.e. it has a horizontal component. Using a device as shown in Figure 2-50, the cambered tire's lateral force component can be measured using hub mounted strain gauges. If fine lines, as described

in Figure 2-51, are scribed into the surface of the tire's tread, a slight outward deflection of the tread surface can be observed as indicated in Figure 3-22. As an aside, if the hydraulic load applied to the hub is quickly released, the tire will jump back at an angle proportional to the camber angle i.e. Non Vertical Resultant Elastic Forces. **Important**: don't try this at home. With a 1500 lb. load on the hub, a 6º camber angle and the tire aired to 50 psi, the result is pretty **bloody** dramatic.

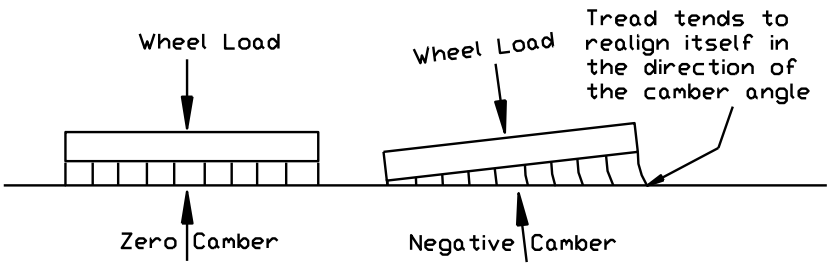


Figure 3-22

Counteracting Centrifugal Forces – As illustrated in Figure 3-22, when loaded, the tire's tread particles tend to be forced out in the direction of the camber angle. Up to the point where the lateral acceleration forces are sufficient to perpendicularly realign the tread particles with the track surface, the shear angle of the tread particles are negative. The lateral acceleration forces must first overcome this negative shear angle of the tread particles before elastic forces as the result of a positive shear angle begin to build. This slight negative alignment of the tread particles allows for a small increase in the tire's performance window i.e. the tire can produce a higher level of lateral acceleration for any given amount of positive tire shear angle

The combination of these two types of camber forces can significantly improve a tire's cornering efficiency by counterbalancing tire and chassis centrifugal forces. Centrifugal forces, in this case, are the reaction forces traveling from the chassis through the tires back to the track. As already discussed, the word centrifugal is somewhat of a misnomer. We should probably not refer to those forces trying to keep the chassis moving along its original line of motion as centrifugal but it has pretty much become sign convention *(I'm not yet willing to circumvent that particular convention).*

The Inner Workings of Camber - When a tire is turned, its tread begins to shear because the tread contact patch is more or less stuck to the track but the forces from the forward motion of the chassis (the forces we have been referring to as centrifugal forces) are still pushing in the original direction of motion. Every part of the chassis, including the rim and tire carcass, wants to keep moving in that original direction. If, as discussed in Chapter 2, the tread was not more or less stuck to the track, it would keep moving in the original direction as well, but, because it is stuck, the tread is pulled, stretched, slipped and sheared. Now, if the tread is pulled, stretched, slipped and sheared too far, for example 5 degrees of shear angle, the tire will begin to become unstuck i.e. the tread shear limit has been exceeded. The combined camber forces will allow the tire to run at a lower angle of shear for any given amount of lateral acceleration force, or, with the inclusion of the combined camber forces, a tire can produce a higher level of lateral acceleration force for any given shear angle.

Shear distortion begins from the outer side of the tire and moves towards the inner side of the tire. Camber counteracting forces move from the inside of the tire towards the outside of the tire if the tire has a negative camber angle. They move in the opposite direction if the tire has a positive camber angle. In the case of an outside tire running negative camber, the camber forces must first be overcome before the contact patch tread surface begins to shear inwardly, not unlike overcoming preload on a spring. As a result, the tire is able to develop a higher coefficient of friction for any given amount of contact patch shear angle - Figure 3-23.

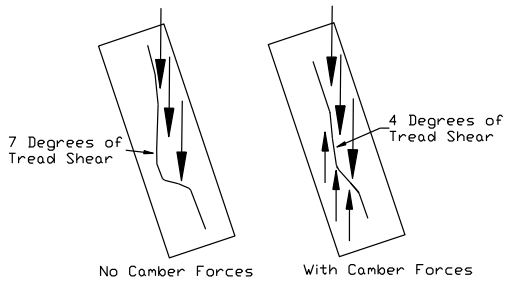


Figure 3-23

A tire's camber coefficient of friction is less affected by increased vertical load, and, up to a point, camber force efficiency is increased by an increase in vertical load. An increase in vertical load sees a proportional increase in camber related visco-elastic distortion at the contact patch. However, as a tire's width is increased, its camber

forces are decreased for any given amount of camber angle. As in the case of shear slope, camber force is defined by an initial slope, which, in turn, is referred to as the camber stiffness.

A tire's camber coefficient of friction varies considerably with tire design and construction. A true radial tire (cords at 90 degrees to the circumference) will produce up to 50% less than an equivalent bias-ply tire. However, because the cords in a modern racing radial are at something less than 90 degrees to the circumference, they are considerably more efficient than their passenger car cousins. In addition, numerous other design characteristics of modern racing radial tires have been developed to improve their camber force coefficient. Nevertheless, all other things being equal, a radial tire normally will need to run at a higher angle of incidence to achieve equivalent bias-ply camber forces.

Camber Analogies:

- The next time you come across a road grader, take a look at how the front tires are cambered to offset the forces from the blade pushing against the dirt. The tires form a parallelogram.

- Participants in a tug-of-war competition are able to produce much, higher forces by leaning backwards and digging in their heels. This is much the same as increasing a tires camber angle.

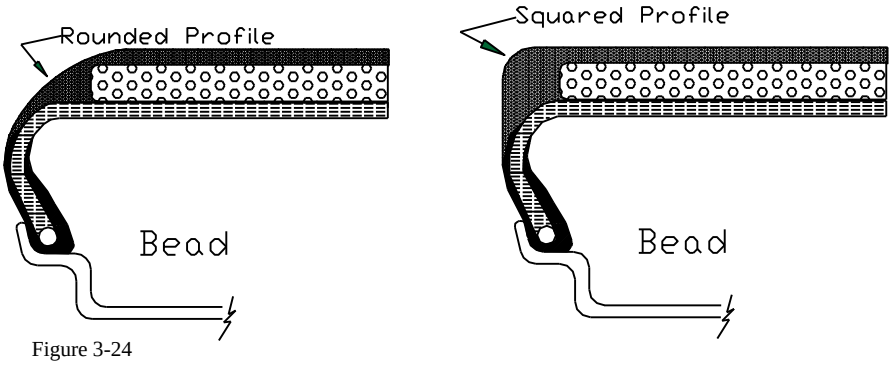


Figure 3-24

Both internal and external tire designs affect a tire's camber coefficient, which, in turn, may require less or more camber angle. For example, tire designers will sometimes deliberately produce a tire with a highly rounded profile in order to achieve less aerodynamic drag –Figure 3-24 (left side). However, a design of this type will generally require a high angle of incidence to achieve the same amount of camber force compared to a design using a squared profile design i.e. the 2001 Formula1 Bridgestone tire with its rounded profile compared to the Michelin's squared design. Formula One chassis using Bridgestone tires generally are required to run a much higher degree of camber compared to the Michelin chassis.

The current BF Goodrich g-Force Trans Am tire is a good example of the squared profile - Figure 3-24 & 3-25. However, in this case, because of other design parameters, many teams actually believe it needs slightly more camber than its predecessor.

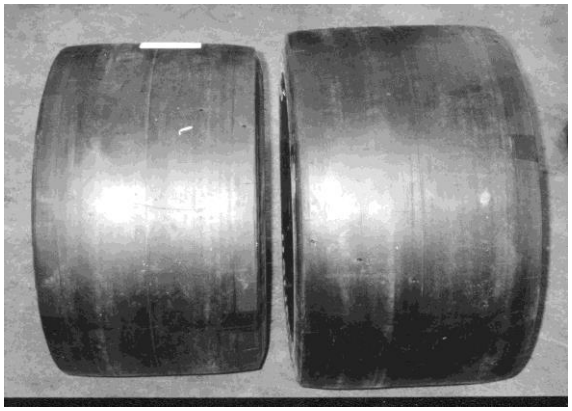


Figure 3-24



Figure 3-25

Camber and Surface Irregularities - Because of their design characteristics, radial tires have a tendency to push the tire towards the lowest part of any track surface irregularity. On the other hand, because camber forces tend to push the tire up and out of a surface irregularity, additional camber can be used to help a chassis from being stuck in a rutted track surface. Bias-ply tires, on the other hand, have a design tendency to climb out of surface irregularities on their own, thereby not requiring additional camber angle while running on a rutted track surface.

Aligning Moments - Caster Torque and Pneumatic Trail

The Contact Patch - Lateral Acceleration - A traditional plan view of a tire's contact patch profile during lateral acceleration can be seen in Figure 3-26.

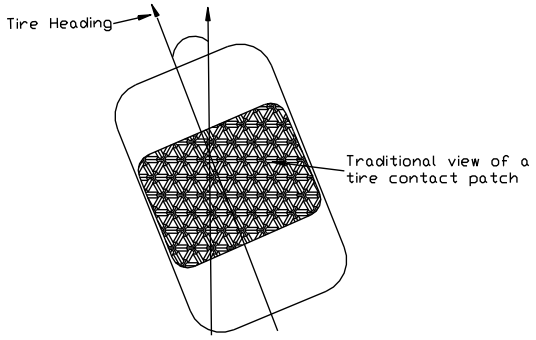


Figure 3-26

However, because of the placement of peak vertical loads and the time it takes to develop peak shear loads, a more realistic representation of a contact patch during lateral acceleration is shown in Figure 3-27. The points of peak vertical loading and peak shear force during cornering and inline acceleration are constantly moving. Note that the point of highest shear deflection/force is aft and to the inside of the tire centerline. The points of highest vertical load and peak shear force are seldom, if ever, at the same place in the tire's contact patch.

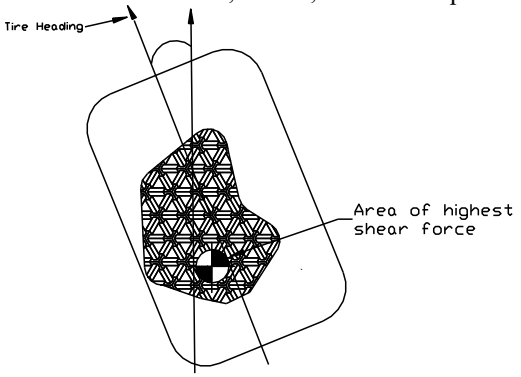


Figure 3-27

Figure 3-28 is a graphic representation of longitudinal shear. As in the case of lateral shear, the highest point of shear deflection/force is aft of the tire centerline.

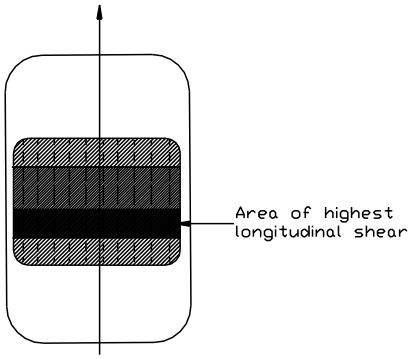


Figure 3-28

Figure 3-29 illustrates the various peaks in vertical load, shear force and slip during forward acceleration. Notice that the peak vertical load is forward of the tire centerline while the shear force peak is aft of the tire centerline. The forward positioning of the vertical load is due to vertical impact force at the leading edge of the contact patch. As the tire tread rotates into the leading edge of the contact patch, it is slowed as it contacts the track surface, also referred to as tread momentum loss. This, in turn, increases ground load at the leading edge as well as increasing drag forces.

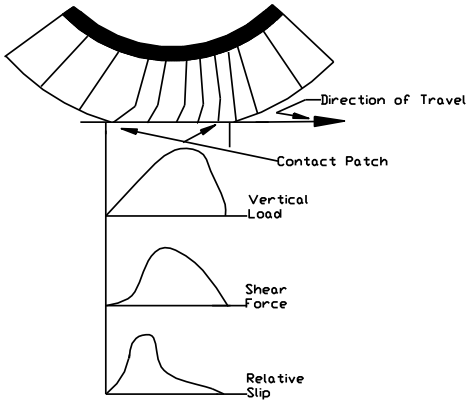


Figure 3-29

Figure 3-30 shows a graphic view of combined lateral and longitudinal shear. The amount and direction of tread shear is indicated by the various lines within the contact patch. The point of highest shear deflection is aft and to the inside of the tire center line, much the same as during lateral acceleration alone. However, as previously discussed, during combined acceleration, the point of peak shear is stretched both laterally and longitudinally. Because the tread molecules only have a given amount of elasticity for any particular tread composition, during combined acceleration the tread can not sustain as high a shear force in either the lateral or longitudinal direction. The peak force is a result of the two individual forces.

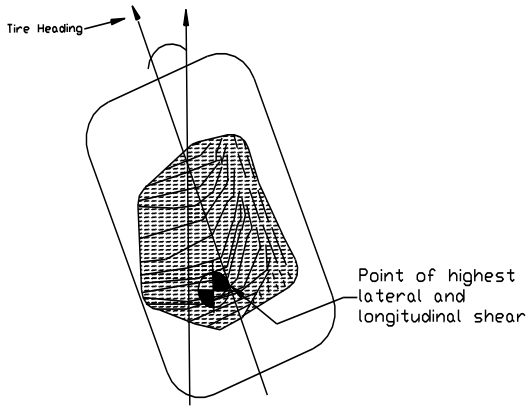


Figure 3-30

Figure 3-31 is a graphic view of a tire's resultant force during combined lateral and longitudinal acceleration. Notice that the center of the tire's peak coefficient force is aft and inward of the tire's center line. As previously indicated, maximum lateral or longitudinal force can not be achieved during combined acceleration. The combined peak force is greater than the individual forces but neither lateral nor longitudinal forces are at their maximum during combined acceleration. *Key tuning tool.*

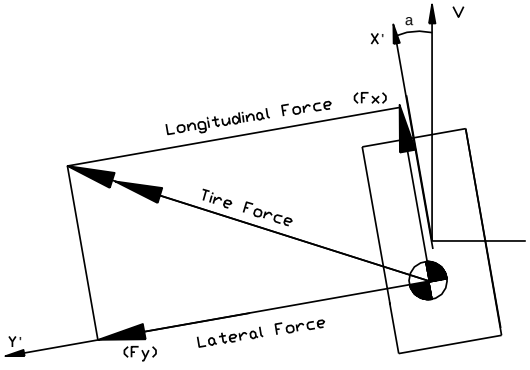


Figure 3-31

Figure 3-32 is another way of displaying the level of shear forces within a tire contact patch. The darker patches represent higher shear forces. As is the case with virtually all graphics within this book, the contact patch graphics only represent a moment or snapshot in time. During actual on track conditions, the forces and positioning of the forces within the contact patch are constantly changing.

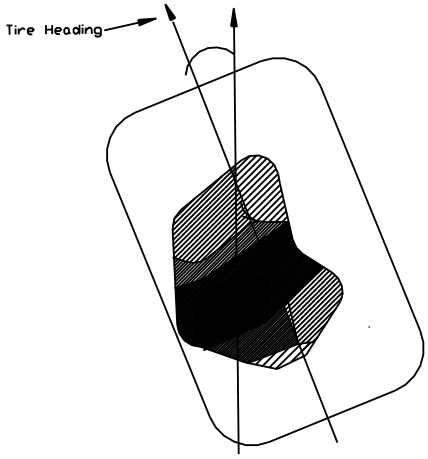


Figure 3-32

Caster Torque (Moment) - Mechanical torque is the result of side view kingpin inclination (caster). During steer, the caster arc causes the inside wheel to lift the chassis as it turns. A chassis reaction force, in turn, tends to push the inside wheel back straight as it attempts to return to its lowest point. This chassis reaction force is referred to as an aligning moment. *It's not that simple*.

Pneumatic Torque - (Tread Moment) - A tire's tendency to steer (realign) about an axis at the centroid of its contact patch (point of maximum shear force) is referred to as pneumatic aligning torque. As a tire is initially turned, it rotates about its centerline but the force trying to realign the tire to its neutral position is at a point aft of the tire's centerline. Because lateral shear forces build from a point forward of the tire center line to a maximum at some point aft of the tire center line, the force distribution is not evenly split forward and aft of the tire center line. It is this uneven distribution of forces that produces a pneumatic alignment torque. Pneumatic aligning torque is measured in foot-pounds for a given shear angle and load. "The steering went all queer and light and then the car pushed up onto the marbles and into the wall". GM test driver

As illustrated in Figure 3-33, the area of the contact patch's highest shear forces are aft of the tire centerline. This point is referred to as the shear force centroid. The distance from this centroid to the tire centerline is called the pneumatic trail. It is from this centroid that the combined sheared tread aligning moment attempts to torque (realign) the tire to its neutral position. The further this aligning centroid is from the tire's centerline, the greater the moment for any given shear force - Figure 3-34. *It's definitely not that simple.*

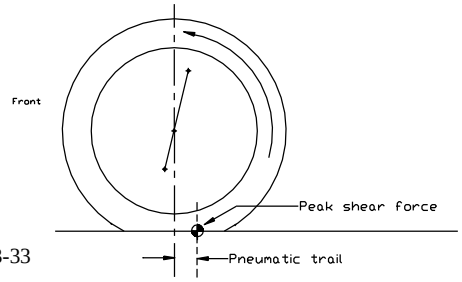


Figure 3-33

In reality, there are two moments - one aft of the tire centerline and one laterally displaced from the tire centerline. As the amount of slip at the trailing edge of the contact patch migrates towards the contact patch's leading edge, pneumatic torque decreases proportionately. As the centroid point nears the tire centerline, the aligning force will drop to zero. If the centroid point moves forward of the tire center line, the aligning forces may actually reverse, thereby attempting to steer the tire into the turn rather than away from it. This is a very important tool to help the driver know where he/she is on the coefficient of friction curve.

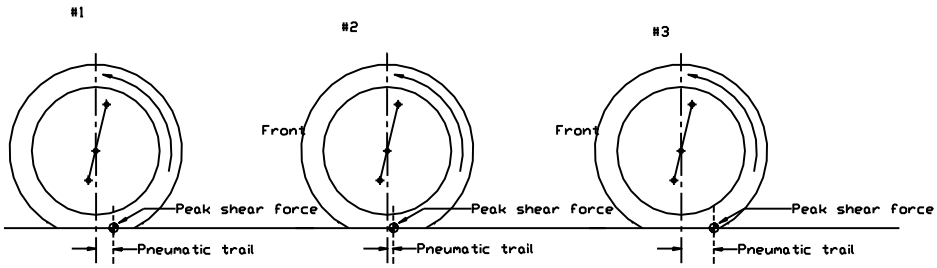


Figure 3-34

Further Definitions

Pneumatic Torque - Carcass (Sidewall and Belt) - The tread, carcass and belt can be seen as a series of lateral and/or longitudinal springs all having centroid moments in relation to the tire's point of rotation, all with different spring rates (stiffness coefficients). Each spring is exposed to the same lateral and/or longitudinal forces and though the carcass and belt have different centroid positions, they are very closely related to the tread centroid.

Combined Pneumatic Aligning Torque - The individual pneumatic aligning torques are a result of force times moment length. The combination of tread, carcass and belt (radial tire only) torques produce the resultant aligning moment. If a suspension corner had no caster angle, (side view kingpin inclination angle), the pneumatic torque would be the only aligning force.

Camber Aligning Forces - Aligning force as the result of camber tends to align the tire in the opposite direction to that of pneumatic trail; however, camber aligning forces tend to be quite small, and, for the most part, are not included in simulation programs. At the most, the engineer would need to add a small amount of positive caster angle to compensate for the small decrease in pneumatic trail torque as the result of camber alignment forces. Bias-ply tires have a higher aligning torque per degree of camber than a radial tire.

Tire Pressure and Aligning Force - Changes in tire pressure also affect pneumatic aligning forces because of resultant changes in the contact patch length, and, to a small degree, the contact patch width. The aligning moment is quite sensitive to changes in the contact patch size. As the contact patch changes in size, the peak shear point also moves.

Vertical Load and Pneumatic Aligning Forces - As in the case of changes in tire pressure, a tire's contact patch size is highly sensitive to changes in vertical load, particularly if the sidewalls have a low stiffness coefficient. Because the peak shear forces occur aft of the center line near the trailing edge of the patch, the pneumatic moment between the tire center line and the aligning centroid will also increase in proportion to the patch's increase in size. An increase in vertical loading will proportionately increase pneumatic aligning force.

Aligning Torque and Its Effect at the Steering Wheel - Aligning torque as the result of the various pneumatic moments only has a small effect on the car's understeer and oversteer balance but it does have a considerable effect at the steering wheel where it can be equal to or larger than caster angle forces in developing steering wheel return torque. Pneumatic moments are large enough that a designer needs to consider their loads when developing the steering system.

Tire Heating - Work produces heat. The higher the friction force, the faster heat builds. Because there is a higher coefficient of friction within the peak shear load region of the contact patch, this is the area that builds the most heat during the slip to regrip process. A temperature increase in this circumferential band region, as measured with a pyrometer, is not necessarily the result of camber, toe or psi imbalance. In the tread portion of the tire, heat generation is primarily from slip and/or sliding, while, within the carcass, it is primarily a function of the rubber and

cords being worked (twisted). Heat generation in the belt is primarily a function of the belt bending at the front of the contact patch during vertical tread impacting at the leading edge.

Energy Loss

Tread Momentum (Energy) Loss, Tire Drag and Tire Rolling Resistance - All three names mean the same thing i.e. the amount the tire's forward motion is retarded (motion resistance force). Rolling resistance (tire drag) is one of the primary factors in motion resistance force. A considerable amount of the power used, as the result of the tire being rolled, is converted into heat generation of the various tire components.

Rolling Resistance (Tire Drag) Factors:

- Tire components, compounding and design
- Amount of tread shear, slip, regrip and slide - all of these factors generate heat
- Side wall deflection - a major contributor to carcass heat generation
- Aero drag - can be a significant factor in determining the amount of overall motion resistance force, particularly with open wheel cars
- Inflation pressure - a tire's psi is one factor determining a tire's level of elasticity (spring rate) - psi in combination with load also controls the amount of sidewall deflection - up to a point, increasing psi will decrease roll resistance, and, in turn, heat build up
- Tire stagger - Chapter 18
- Tire rolling velocity - a tire's velocity drag coefficient increases dramatically at high speeds – Figure 3-35

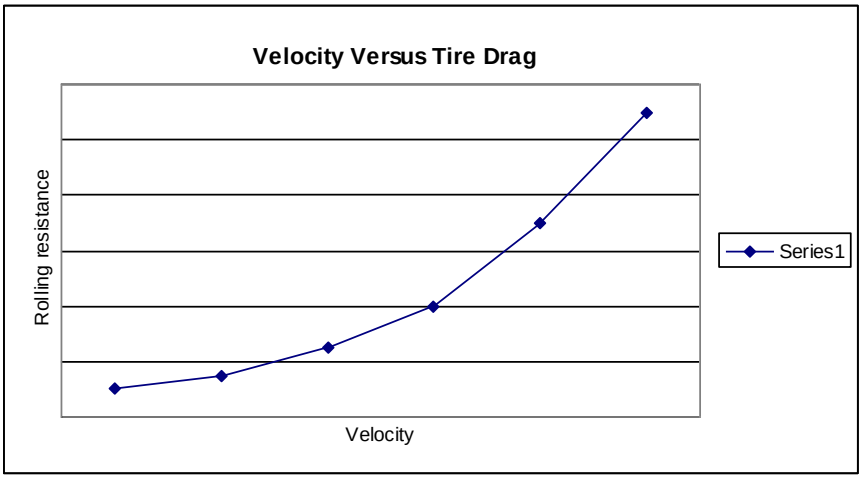


Figure 3-35

- Load sensitivity - this factor is a result of tire design – Figure 3-36

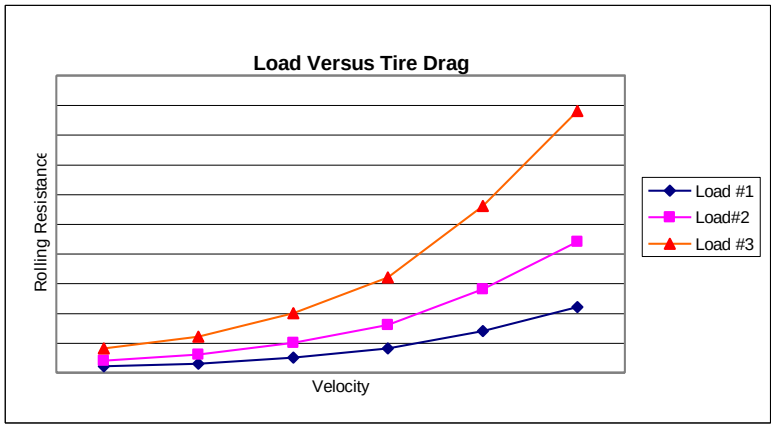


Figure 3-36

Rolling Resistance Coefficient - A car's combined motion resistance forces include tire rolling resistance, aero forces and drive train resistance factors and can be found by running coast down tests in conjunction with on board data systems. The hard part is separating all of the individual factors.

Inline tire drag forces, except for the aero forces, can, to some degree, be determined by tire test apparatus. Tire and chassis aero forces are primarily determined through wind tunnel testing while drive line efficiency is found through comparing engine dyno, drive train dynos and chassis dyno testing. However, even if a team has access to all of the above, the individual on track rolling resistance and inline drag coefficients are extremely difficult to quantify. Most teams first try to quantify the aero and drive train coefficients through wind tunnel and dyno testing and then carry out a series of test pad and on track tests, using their data acquisition system to separate the tire rolling resistance (inline tire drag) coefficients from the rest.

Most of the apparatus used in laboratory tire testing is not able to simulate many of the on track conditions, including aero rolling resistance forces acting on the tires. If the engineer has access to inline tire drag numbers from the lab, he/she will often run parallel on track tests in order to see if there is any usable correlation between the two tests. If usable correlation can be found between lab tests and on track testing, it makes all further track testing a little easier to believe.

Simulation Formulas - There have been many serious attempts to develop formulas that will calculate combined and individual inline and lateral drag coefficients but this is still a developing science. Most formulas need a series of correction factors added to correlate with the various track tested results. At this point in time (2003), most teams take their combined lab and track test results and develop best fit formulas to predict changes rather than using a purely theoretical formula.

The following "Tread Momentum loss" formula is taken from one presented by Chuck Hallum[1] during the 1998 SAE Motorsports Conference. The accompanying table includes a correction factor which was not part of the original formula. The correction factor was included to represent numbers closer to what was found during on track and pad testing and has all drive line and aero factors removed. Through the correction factor results do not represent exact numbers, for obvious reasons, they do correlate with actual numbers quite closely. However, the correction best fit factor is seldom as linear in nature.

As can be seen, even when using a correction factor, the inline drag numbers are quite significant. This type of momentum loss is definitely not something the race engineer wants to ignore. Tires with high aspect ratio numbers normally have a much higher rolling resistance (tire drag) coefficient.

[1] Chuck Hallum, Hallum Racing Enterprises, 1998 Motorsports Engineering Conference Proceedings, Volume 1 Paper 983028 - Published by SAE, 400 Commonwealth Drive, Warrendale, PA, 15096-0001

Note: Though my drag numbers for individual tires often end up approximately half that of the Chuck Hallum model, that is not to say that his numbers or my numbers are wrong. I suspect it has more to do with how each of us corrected for aero, engine and drive train losses.

Tread Momentum Loss in HP

The original equation was developed by "Hallum Racing Enterprises"
The only difference between the HRE equation and this calculator is the inclusion of an on track coast down correction factor

Tire contact patch length = CP	CP	4.55
Tire Diameter = D	D	27
Tread weight per one foot = Tw	Tw	1.5
	mph	150
Correction factor = *ƒ*	*ƒ*	1

Tread deflection angle - β = (ASIN(CP/R)/*ƒ*	β	9.702
Velocity - V = (mph*88)/60	V	220.000
Tread Mass Rate - Tmr = (Tw/32.2)*V	Tmr	10.248
Tread Momentum - Tm = Tmr*V	Tm	2254.658
Tread Momentum - Mv = Tm*SIN(β)	Mv	379.952
Tread Velocity - Tv = V*SIN(β)	Tv	37.074
PH Loss - HP = Mv*(Tv/550)	HP	25.612

The Tread Momentum Calculator, as shown above, can be found on the accompanying CD as part of the Excel **Kinematic Calculator** and as part of all the **Wan31A** and **Wan31B** models. The calculator is found in the **Tire Drag** worksheet.

On Track Testing - Data for all of the following tables is the result of on track testing and is from a right front oval track tire, (no inner liner), with a load of 1500 lb. Tread width is 10" with a 27 " diameter. The tire patch lengths for this example are - 20 psi = 7.68", 30 psi = 5.18" and 40 psi = 3.94". The tread weight was determined by weighing a one foot length of tread including 1/8 of the sidewall on each side. As the tables indicate, a tire, even when rolling at zero lateral or longitudinal acceleration, will create a considerable amount of drag due to the leading edge of the tire impacting the track as it rolls.

Table 3-4 and its accompanying graph illustrate the differences between three pressure settings. Notice the dramatic change in tire drag that accompanies a change in tire pressure. Also remember that an increase in drag produces a corresponding increase in tire temperature. *Key tuning tool*s.

Table 3-4

hp Loss With Correction Factor			
mph	**20 psi**	**30 psi**	**40 psi**
50	1.3	0.6	0.3
75	4.3	1.9	1.1
100	10.1	4.6	2.7
125	19.7	9.0	5.2
150	34.0	15.5	9.0
175	54.0	24.6	14.2
200	80.6	36.7	21.2

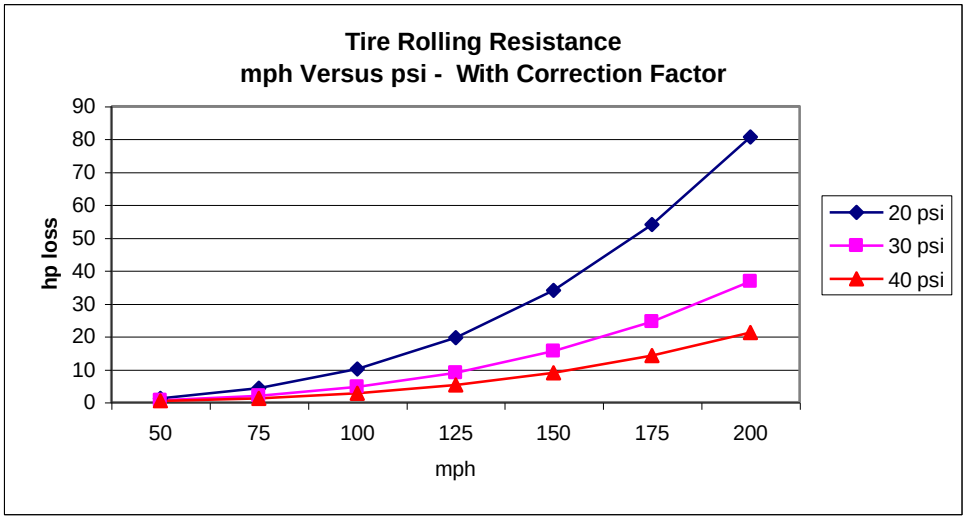
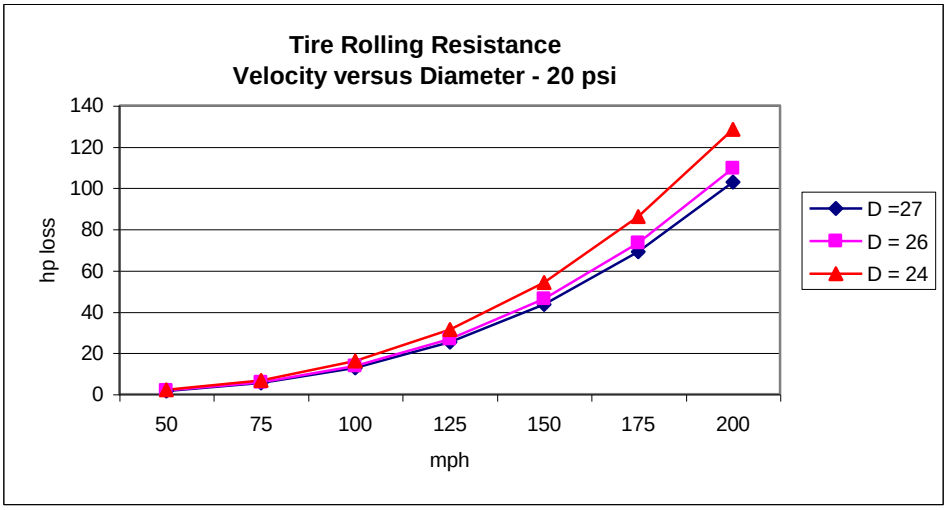


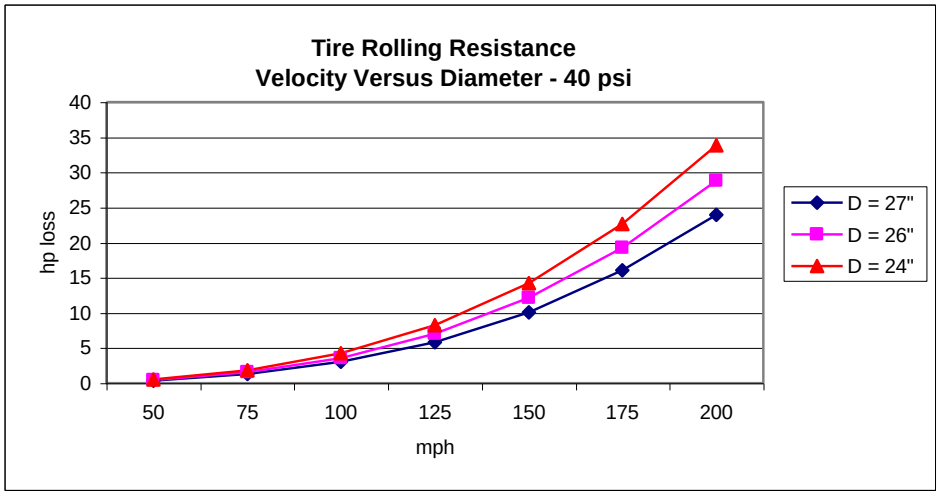
Table 3-5 and its accompanying graph shows the change in drag as it relates to changes in tire diameter.

Table 3-5

D= 27"	Corrected		D = 26"	Corrected		D = 24"	Corrected	
mph	20 psi	40 psi	mph	20 psi	40 psi	mph	20 psi	40 psi
50	1.6	0.4	50	1.7	0.5	50	2.0	0.5
75	5.4	1.3	75	5.8	1.5	75	6.8	1.8
100	12.9	3.0	100	13.7	3.6	100	16.1	4.2
125	25.1	5.9	125	26.7	7.0	125	31.4	8.3
150	43.4	10.1	150	46.2	12.2	150	54.2	14.3
175	68.9	16.1	175	73.3	19.3	175	86.0	22.6
200	102.9	24.0	200	109.4	28.8	200	128.4	33.8



Graph from Table 3-5 (20 psi)

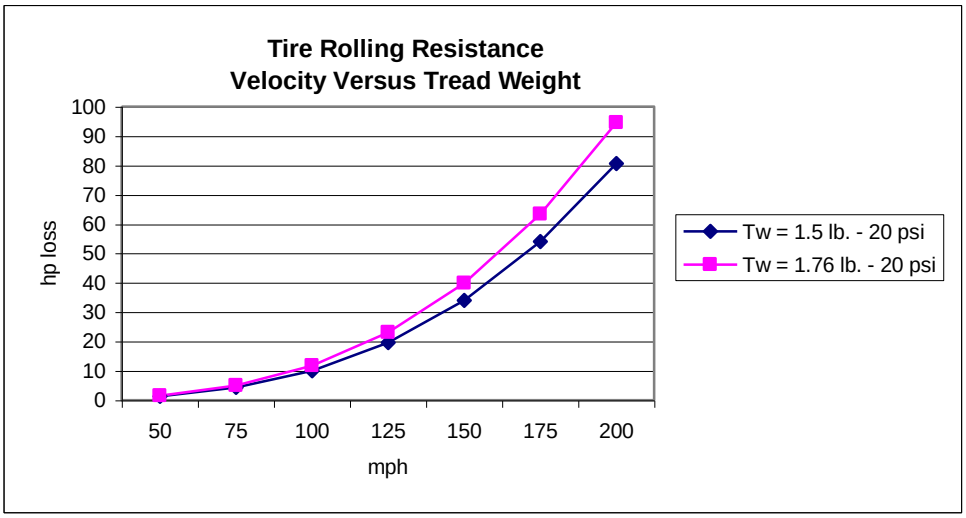


Graph from Table 3-5 (40 psi)

Table 3-6 and its accompanying graph illustrates the change in drag as the result of a change in tread weight.

Table 3-6

Tw = 1.5	Corrected		Tw = 1.76	Corrected	
mph	20 psi	40 psi	mph	20 psi	40 psi
50	1.3	0.3	50	1.5	0.4
75	4.3	1.1	75	5.0	1.3
100	10.1	2.7	100	11.8	3.1
125	19.7	5.2	125	23.1	6.1
150	34.0	9.0	150	39.9	10.5
175	54.0	14.2	175	63.4	16.7
200	80.6	21.2	200	94.6	24.9



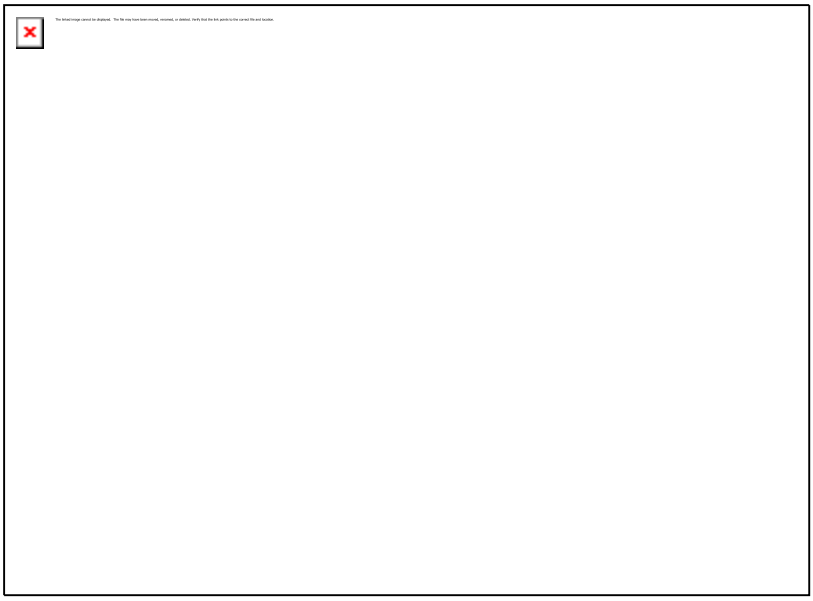
Graph from Table 3-6

Shear Drag and Energy Loss - Shear drag is the result of the tire laterally accelerating. Like momentum/inline drag, shear drag can generate a considerable amount of hp loss. From the formula, a car running 100 mph with a vertical load of 1000 lb., a combined shear percentage of 7% and a lateral acceleration of 1.3 g's, the resulting drag on an individual tire would be in the neighborhood of 0.07*1.3*1000 = 91 lb./ft. Running at a speed of 100 mph, the tire would generate a hp loss of 91*100*88(60*550) = 24.27 hp. This formula is found in numerous vehicle dynamic text books such as Dixon's "Tires, Suspension and Handling"[2]. The formula assumes that the lateral drag forces are closely related to the steered (slip) angle of the tire as shown in Figure 3-37 and that the moment arm seen in the drawing represents the lateral drag force (negative traction force). Like most formulas of this type, the engineer will need to add a number of correction factors after comparing the results with track test data. *Don't forget to develop the needed correction factors from your cost down and skid pad testing.*

Shear hp Loss

Veritcal load on Tire = L	L	1000
Lateral Cornering Force = Ay	Ay	1.3
Front Wheel Shear Angle in % = δ	δ	7.00%
	mph	100
Drag in lb. D = δ*Ay*L	D	91
hp loss hp = (D*mph*88)/60*550)	hp	24.27

The **Shear hp Loss** Calculator, as shown above, can be found on the accompanying CD as part of the Excel **Kinematic Calculator** and as part of all the **Wan31A** and **Wan31B** models. The calculator is found in the **Tire Drag** worksheet.



[2] John C. Dixon, Tires, Suspension and Handling, Published by SAE, 400 Commonwealth Drive, Warrendale, PA 15096-0001

Table (3-7 & 3-8) shows examples of drag and hp loss numbers using the Dixon formula, but, as always, remember this type of number should never be taken as gospel. Every formula, no matter how carefully crafted, will need one or more correction factors added to keep it inline with on track results (See Book Three). A steer angle of 0.08 = 8 degrees of steer.

Table 3-7 (Ay = 1.3 - Shear hp Loss @ 0.08 and 0.05 steer angle)

Lateral Tire Drag due to Shear - hp Loss					Before Corr.		After Corr.
Vertical					factor		factor
Load	Ay	mph	Steer Angle	Drag in lb./ft	hp Loss	Corr. Factor	hp Loss
500	1.3	100	0.08	52	13.87	0.68	9.43
500	1.3	160	0.08	52	22.19	0.68	15.09
750	1.3	100	0.08	78	20.80	0.68	14.14
750	1.3	160	0.08	78	33.28	0.68	22.63
1000	1.3	100	0.08	104	27.73	0.68	18.86
1000	1.3	160	0.08	104	44.37	0.68	30.17
1500	1.3	100	0.08	156	41.60	0.68	28.29
1500	1.3	160	0.08	156	66.56	0.68	45.26
2000	1.3	100	0.08	208	55.47	0.68	37.72
2000	1.3	160	0.08	208	88.75	0.68	60.35
500	1.3	100	0.05	32.5	8.67	0.68	5.89
500	1.3	160	0.05	32.5	13.87	0.68	9.43
750	1.3	100	0.05	48.75	13.00	0.68	8.84
750	1.3	160	0.05	48.75	20.80	0.68	14.14
1000	1.3	100	0.05	65	17.33	0.68	11.79
1000	1.3	160	0.05	65	27.73	0.68	18.86
1500	1.3	100	0.05	97.5	26.00	0.68	17.68
1500	1.3	160	0.05	97.5	41.60	0.68	28.29
2000	1.3	100	0.05	130	34.67	0.68	23.57
2000	1.3	160	0.05	130	55.47	0.68	37.72

Table 3-8 (Ay = 2.0 - Shear hp Loss @ 0.08 and 0.05 steer angle)

Lateral Tire Drag due to Shear - hp Loss					Before Corr.		After Corr.
Vertical					factor		factor
Load	Ay	mph	Steer Angle	Drag in lb./ft	hp Loss	Corr. Factor	hp Loss
500	2.0	100	0.08	80	21.33	0.68	14.51
500	2.0	160	0.08	80	34.13	0.68	23.21
750	2.0	100	0.08	120	32.00	0.68	21.76
750	2.0	160	0.08	120	51.20	0.68	34.82
1000	2.0	100	0.08	160	42.67	0.68	29.01
1000	2.0	160	0.08	160	68.27	0.68	46.42
1500	2.0	100	0.08	240	64.00	0.68	43.52
1500	2.0	160	0.08	240	102.40	0.68	69.63
2000	2.0	100	0.08	320	85.33	0.68	58.03
2000	2.0	160	0.08	320	136.53	0.68	92.84

Track and Skid Pad Testing - Shear drag coefficients are most often calculated from skid pad and track testing - Chapter 19. Shear drag is much easier to calculate than rolling resistance drag. Shear drag tables can normally be developed from only four different combinations of speed versus steer (slip) angle tests. From the on board data, best fit tables and graphs are developed for the required combinations of steer versus speed scenarios. These numbers are extremely important, particularly for fast sweeping corners.

Personal Note: One of the best examples I have come across to illustrate shear drag is from an open wheel race weekend run on a one mile oval. While working with two drivers, one with great talent and one who brought money, the two cars had similar set ups and virtually identical engines but the talented driver was turning considerably faster lap times. This resulted in the second driver who brought money declaring he didn't have the hot set up and his engine was down on power. After looking at the data, it was clear the second driver was continuously putting in too much steer angle, and then, of course, having to take it back out. Once we were able to convince this driver to believe in the set up and gradually try smaller steering inputs, his lap times came down dramatically.

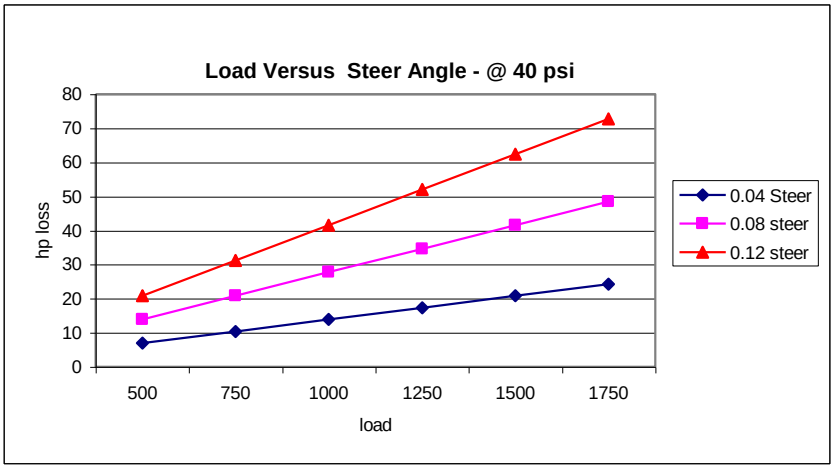
The race engineer, to some extent, is able to manipulate both momentum and shear drag hp losses. *Key tuning tool*. Remember that camber is virtually shear drag free. *Key tuning tool.*

The following shear force tables and their accompanying graphs are quite linear in nature, which is not always the case. When it comes to shear force data, there is no substitute for good on track testing.

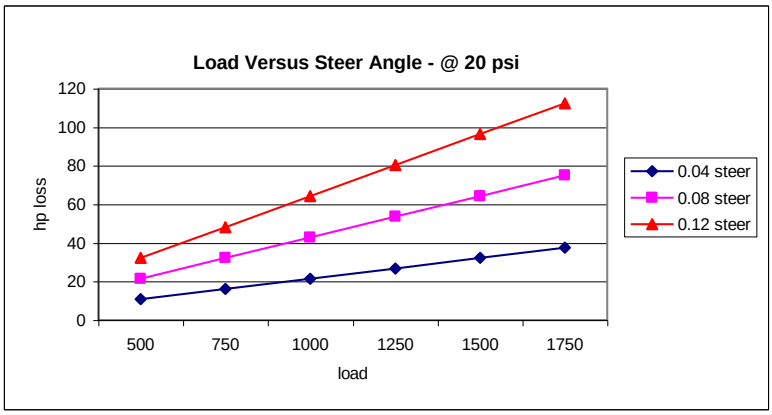
Tables 3-9 and its accompanying graphs illustrate the change in tire drag due to a change in steering angle**.** δ **= 0.04 Represents 4 degrees of steer angle**.

Table 3-9

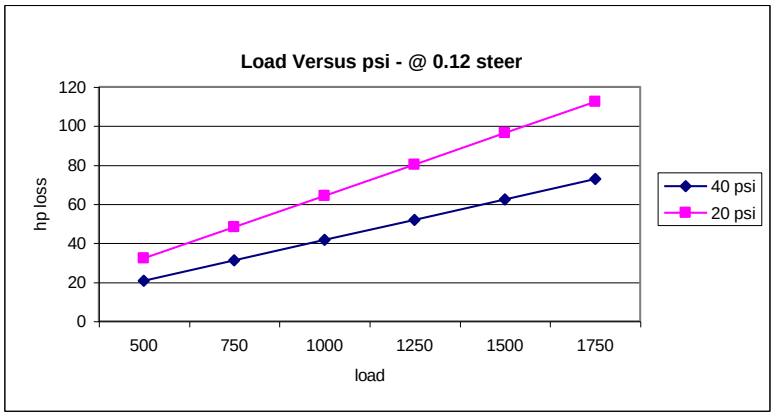
Load	hp Loss	hp Loss	hp Loss	Load	hp Loss	hp Loss	hp Loss
40 psi	$\delta = 0.04$	$\delta = 0.08$	$\delta = 0.12$	20 psi	$\delta = 0.04$	$\delta = 0.08$	$\delta = 0.12$
500	6.9	13.9	20.8	500	10.7	21.4	32.1
750	10.4	20.8	31.2	750	16.1	32.1	48.2
1000	13.9	27.7	41.6	1000	21.4	42.8	64.2
1250	17.3	34.7	52.0	1250	26.8	53.5	80.3
1500	20.8	41.6	62.4	1500	32.1	64.2	96.3
1750	24.3	48.5	72.8	1750	37.5	74.9	112.4



Graph from Table 3-9 (40 psi)



Graph from Table 3-9 (20 psi)

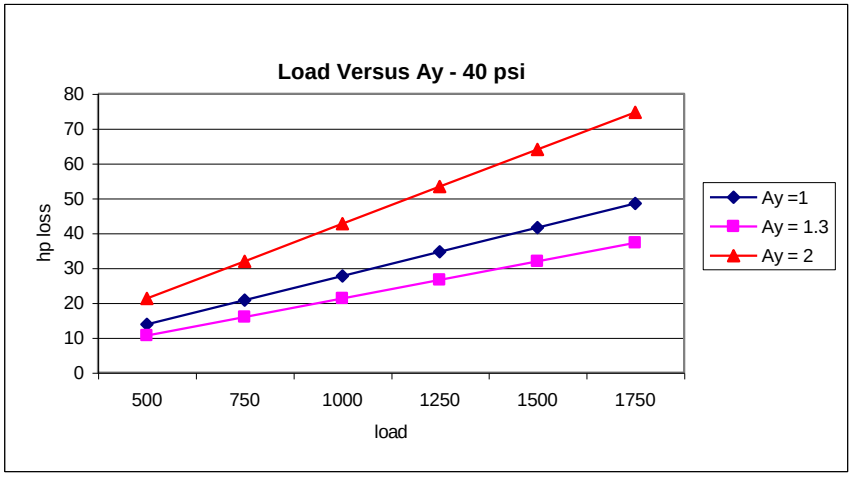


Graph from Table 3-9 (20 & 40 psi @ 0.12 steer)

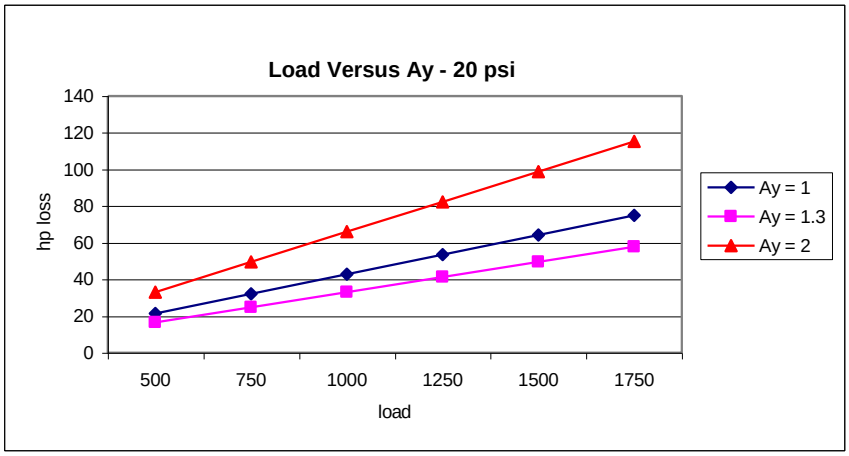
Table 3-10 and its accompanying graph show a change in tire drag in relation to changes in lateral acceleration. All tests are run at a given turn entry speed.

Table 3-10

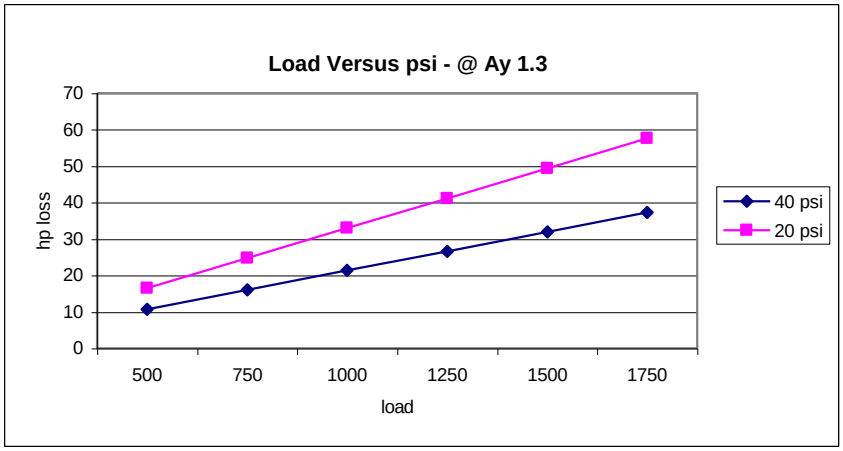
Load lb.	hp loss	hp loss	hp loss	Load lb.	hp loss	hp loss	hp loss
40 psi	Ay =1	Ay = 1.3	Ay = 2	20 psi	Ay =1	Ay = 1.3	Ay = 2
500	13.9	10.7	21.3	500	21.4	16.5	32.9
750	20.8	16.0	32.0	750	32.1	24.7	49.4
1000	27.7	21.3	42.7	1000	42.8	32.9	65.9
1250	34.7	26.7	53.3	1250	53.5	41.2	82.3
1500	41.6	32.0	64.0	1500	64.2	49.4	98.8
1750	48.5	37.3	74.7	1750	74.9	57.6	115.3



Graph from Table 3-10 (40 psi)



Graph from Table 3-10 (20 psi)

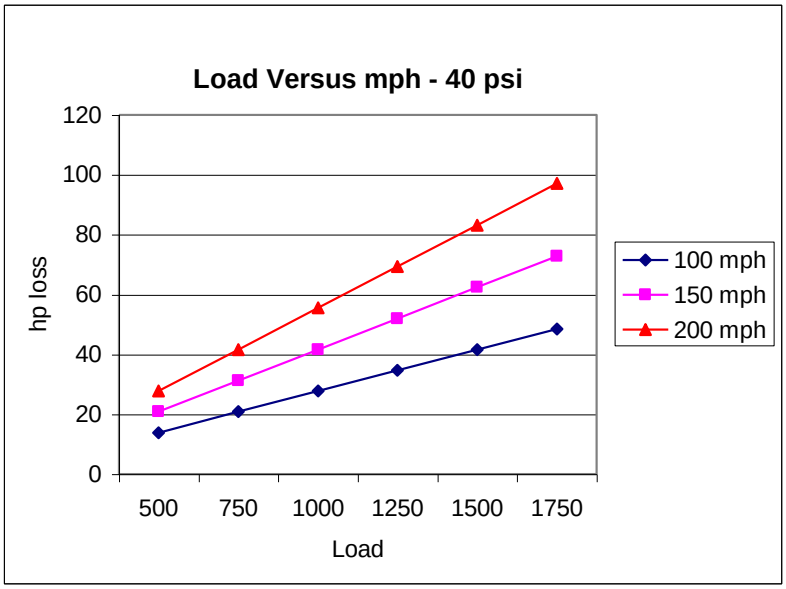


Graph from Table 3-10 (20 &40 psi @ Ay 1.3)

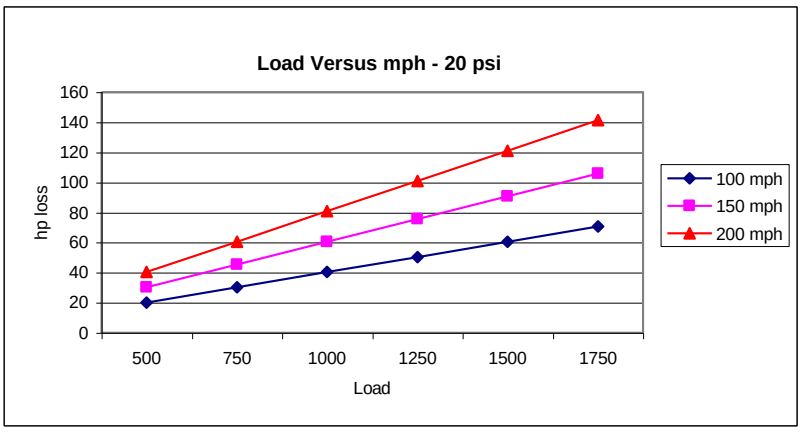
Table 3-11 and its accompanying graphs show the relationship between drag and mph at a constant steer (slip) angle.

Table 3-11

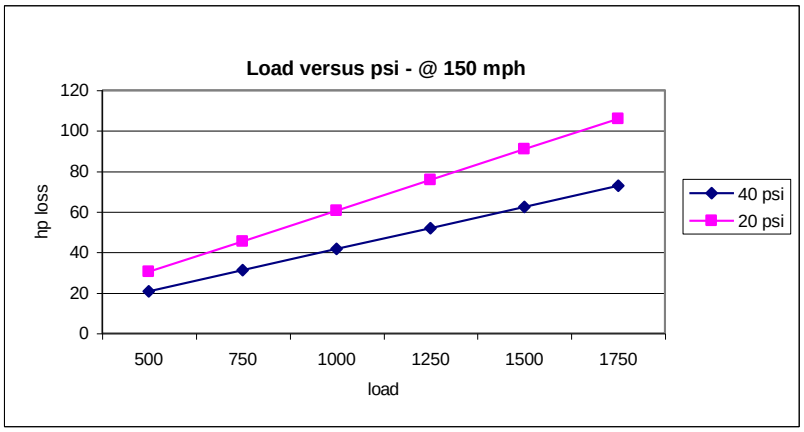
40 psi	hp loss	hp loss	hp loss	20 psi	hp loss	hp loss	hp loss
Load	mph = 100	mph = 150	mph = 200	Load	mph = 100	mph = 150	mph = 200
500	13.9	20.8	27.7	500	20.2	30.3	40.4
750	20.8	31.2	41.6	750	30.3	45.4	60.5
1000	27.7	41.6	55.5	1000	40.4	60.5	80.7
1250	34.7	52.0	69.3	1250	50.4	75.7	100.9
1500	41.6	62.4	83.2	1500	60.5	90.8	121.1
1750	48.5	72.8	97.1	1750	70.6	105.9	141.2



Graph from Table 3-11 (40 psi)



Graph from Table 3-11 (20 psi)



Graph from Table 3-10 (20 & 40 psi @ 150 mph)

Heat Factors and Air Pressure Versus Acceleration Forces - Depending on the tire design, a tire may actually gain a small bit of lateral acceleration force by increasing the tire pressure slightly from the manufacturer's recommended pressure. This may occur because of increased sidewall stiffness or because the tire manufacturer's psi recommendation does not produce the peak forces for your particular set up or driver's style. Running much above the recommended pressure normally requires that the driver use slightly less steer angle to maintain the same heat factor. The stiffer the tire, the faster the car will respond to a change in steer angle. *Key tuning tool.*

In many instances within the usable psi range, running at a lower air pressure increases acceleration forces but also results in a higher heat and drag factor. Like everything else involved in setting up a race car, correct tire pressure balance is not easily achieved. Below the optimum air pressure, trailing edge slip and sidewall shear will increase, lowering acceleration forces and increasing the tread and sidewall heat factors. Above their optimum pressure, trailing edge and overall contact patch slip will increase, lowering acceleration forces and increasing the heat factor. Very carefully planned and executed on track tests are the only way for most teams to develop best case tire pressure scenarios. Changes in the tire performance, as the result of a small psi change, have more to do with the change in contact patch size and how the various tire components are worked than with a change in the tire's spring rate.

Drag and the resultant heat generation play a major role in both short run and long run chassis performance. Tire normal load, Ackerman, steer (slip) angle etc. all affect the performance envelope. Combined lateral and inline drag should be carefully addressed by the race engineer when developing the chassis set up for each individual track and track conditions. Combined drag can have major asymmetric ramifications. Lateral drag heat generation is one reason many set ups use reverse Ackerman at the inside wheel for large radius and/or high banked corners. *Key tuning tool.*

Working towards reducing inline drag coefficients is often far more productive than trying to develop engine torque and hp, and, in fact, tire pressure settings and steer angles are frequently dictated as much by the various drag coefficients as by the acceleration coefficients. *Key tuning tool*.

The Big Three

- Heat
- Normal load
- Psi

Heat, normal load and psi are the three most important factors linked to chassis performance and virtually every test and set up a team designs and runs will have these three factors at the forefront - Chapter 19.