RACECAR ENGINEERING

V9N4 May 1999 | UK £3.70 | USA $7.95

V8 FOR VICTORY

- ELECTRIC RACECAR SERIES FOR USA STUDENTS
- PEUGEOT BACK WITH NEW WORLD RALLY CAR
- PILBEAM'S INTERNATIONAL SPORTS-PROTOTYPE

A LINK HOUSE MAGAZINE
www.linkhouse.co.uk/racecar
9 770961 109036 05

The Design & Tuning of Asymmetrical Racecars

PART 2: DYNAMIC LOAD TRANSFER

In Part 1 of this series (V9N3), we introduced the concepts of tyre load sensitivity, static and dynamic wedge, velocity and acceleration, action and reaction, and couples and moments. We explained how a tyre force, acting horizontally at ground level, and the corresponding inertial reaction, acting horizontally at centre of gravity (CG) height, form a couple. We noted that roll, pitch and yaw moments result from such couples. We closed with the observation that roll and pitch moments react against our tyres and suspension, producing vertical load changes in these components which we call 'dynamic load transfer'.

By MARK ORTIZ

We will now explore these effects in detail. We will examine the ways that suspension design, calibration, and adjustment control load transfer apportionment among the wheels. Load transfer apportionment in turn determines how the car's dynamic wedge varies. And dynamic wedge influences our cornering balance because of tyre load sensitivity. Once we understand the interplay of these phenomena, a very large portion of vehicle behaviour suddenly makes perfect sense.

Many of the ways we apportion dynamic load transfer are applicable to both symmetrical and asymmetrical vehicles. We'll look at these first, then consider some further possibilities that are peculiar to asymmetrical cars.

Fig 1A shows a car cornering to the right, similar to the last example we presented in Part 1. Here, all four tyres are generating centripetal cornering force, shown as a single action force F_A. The centrifugal inertia reaction F_R acts at the centre of gravity height H_{CG}. The couple generates a leftward roll moment $F_R \cdot H_{CG}$.

This moment acts through the suspension and tyres, and finally shows up as a load change at the car's only points of support – the tyre footprints. F_{OP} represents the additional force exerted by the road surface against the two outside tyres. F_{IP} is the equal and opposite decrease in support force against the two inside tyres. F_{OP} and F_{IP} form a couple equal and opposite to the roll couple. This couple's moment arm is the car's track width T.

Therefore, the magnitude of F_{OP} is: $F_R\ (H_{CG}/T)$.

Fig 1B shows the same example, with gravitational forces added. Gravity exerts a downward weight force -W at the CG. The right and left wheel pairs bear equal and opposite support forces $W/2$ from the road.

(Upward forces are assigned a positive sign, per SAE convention. This comes from aeronautical engineering – lift being positive. For mathematical modelling and computer simulation, sign conventions are crucial. For our less formal discussion, it should be sufficient to know our forces' magnitudes and directions.)

When F_{IP} and F_{OP} are summed with $W/2$, we get our wheel pair loadings while cornering. Note that the combined loading still totals W, but the right and left wheel pairs no longer bear W equally. Consequently, we sometimes speak of our loading change due to cornering as 'weight transfer'.

The trouble with this term is that, in casual conversation, 'weight' and 'mass' are used interchangeably. Many informally trained people are under the impression that weight transfer results from movement of the CG as the car rolls. In American dirt track racing, a majority of racers – including some very successful ones – will tell you, "You have to let the car roll in the turns so it will transfer weight and make side bite."

In reality, roll has only a slight influence on weight transfer. Roll does move the CG outward, but usually less than an inch. On a racecar, the difference in inside wheel pair load reduction, or outside pair load increase, between a really soft setup and a really stiff one is 1% of the car's weight or less. Even an 'active' suspension car, programmed to roll inward, transfers almost as much load to the outside wheel pair as a conventional car.

CG movement can assume much greater significance in trucks and buses. The worst cases are tank trucks, dry bulk hoppers, livestock trucks – vehicles where the cargo can shift or slosh.

Not only does a car's CG move a little in cornering, so do the tyre footprints. The tyre side walls allow the contact patch to shift toward the inside of the turn, relative to the wheel rim. In addition, the footprint's 'load centroid' – the point where the support force can be considered to act – usually moves outward from the footprint's geometric centre. All of these effects are real, and all of them affect wheel loading.

But not a lot. In cars, all these movements are so small that we lose only a little accuracy by ignoring them altogether. The origin of a car's major wheel load changes in horizontal acceleration lies in the couples and moments discussed above. That is why we have belaboured our explanation of these at such length.

Note that only three things significantly affect the total amount of load transfer: how much force the tyres generate, how high the CG is, and how far apart the wheels are. The suspension's total roll resistance hardly matters at all.

However, the *comparative* roll resistance of the front and rear wheel pairs matters mightily – not in terms of total load transfer, but in terms of its distribution. This roll resistance relationship governs what portion of the total load transfer is allocated to each end of the car.

The basic rule is that the stiffer end gets the greater load transfer. Load transfer at the softer end is correspondingly reduced. This increases cornering power at the softer end, and reduces cornering power at the stiffer end.

What we are controlling with this roll resistance balance is our dynamic wedge. If the front has more roll resistance than the rear, the car wedges itself more as it corners harder. If the rear has more roll resistance, the car de-wedges itself in response to cornering force. (Recall that we have defined positive wedge as greater inside – not necessarily left – percentage at the rear than at the front. Therefore, these remarks apply for both right and left turns). Our dynamic wedge changes are superimposed upon, or added to, any static wedge the car may have.

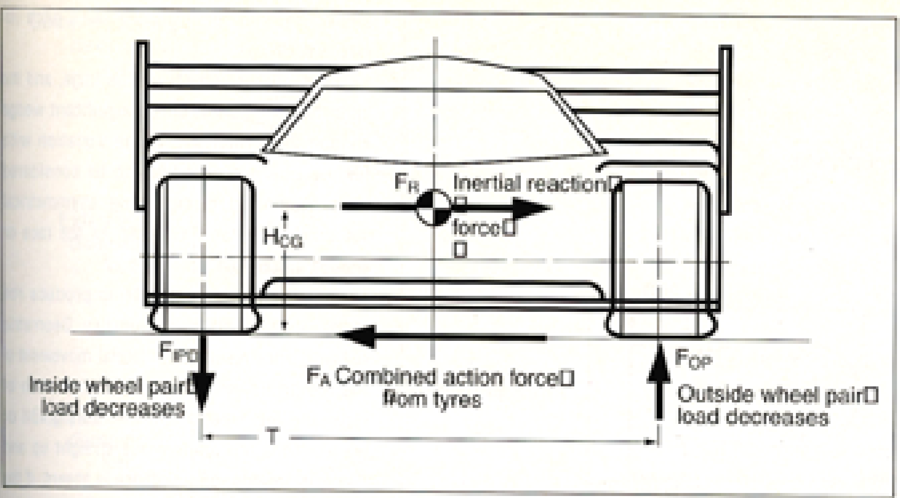


Fig 1A: Origin of load transfer in cornering: the centripetal/centrifugal force couple and the footprint load change couple that counterbalances it.
The sum of the horizontal forces is zero: $F_A + F_R = 0$
The sum of the vertical forces is zero: $F_{IP} + F_{OP} = 0$
The sum of the moments is zero: $F_R \cdot H_{CG} + F_{OP} \cdot T = 0$
The magnitudes opposing moments are equal: $F_R \times H_{CG} = F_{OP} \times T$
Load transfer depends on tyre force, CG height and track width: $F_{IP} = F_A \times (H_{CG}/T)$

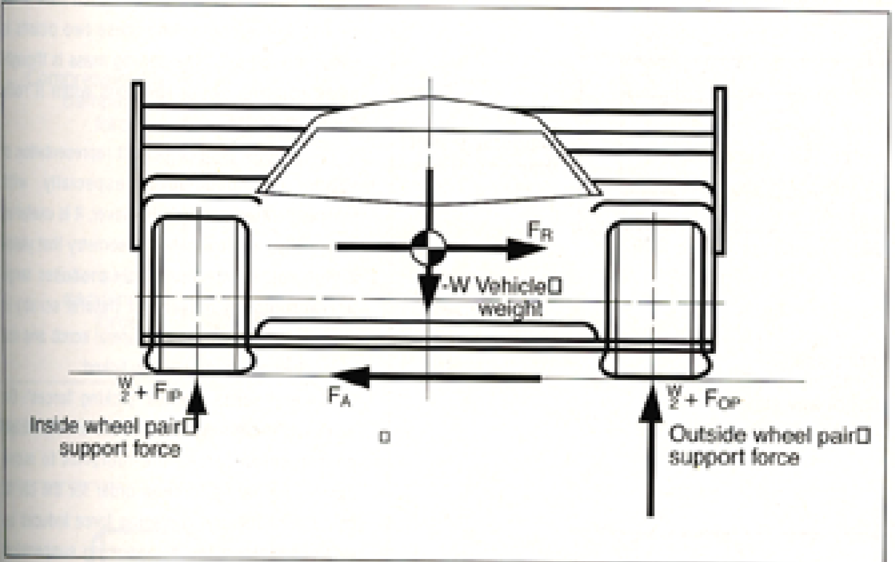


Fig 1B: The same example as Fig 1A, with gravitational effects added
The sum of the vertical forces is zero: $-W + (W/2 + F_{IP}) + (W/2 + F_{OP}) = 0$.
Aerodynamic forces will also be present, but we are ignoring these for simplicity.

Origins of Roll Resistance

Suspension systems resist roll by four basic classes of device or effect. These are the individual wheel springs, inter-connective springing devices such as anti-roll bars (see *Racecar* V7N8), the dampers, and the suspension geometry.

Individual wheel springs and anti-roll bars both resist roll in a position-sensitive manner. We express this resistance as the wheel rate in roll. This rate (in pounds per inch, or Newtons per millimetre) may be very nearly constant through the range of wheel travel, or it may vary with wheel position. It may be the same on the right and left wheels, or not. It is seldom identical front and rear, although it can be.

In steady-state cornering (constant speed, constant radius), the car assumes a roll position, and holds it. The front and rear wheel pairs absorb the lateral load transfer in proportion to the combined wheel rate in roll that each pair possesses. In terms of load transfer allocation, it doesn't matter how much rate the inside or outside member of the wheel pair has – only what their combined rate is.

To help us to discuss stiffness distribution issues, a little additional terminology comes in handy. Cars or wheel pairs can be described as right-stiff or left-stiff, inside-stiff or outside-stiff (relative to a turn), front-stiff or rear-stiff. Or we may describe a right-stiff car as left-soft instead. The idea is concisely to convey how the stiffnesses in a system compare with each other.

As noted in Part 1, a car can be diagonally stiff. A car is diagonally stiff if it is right-stiff at one end and left-stiff at the other. It is also diagonally stiff if it is left-stiff or right-stiff at one end only, or if it is right-stiff or left-stiff at both ends, but unequally so.

A car is not diagonally stiff if (and only if) it has the same combined wheel rate on both diagonal wheel pairs. A car is not right-stiff or left-stiff if (and only if) the right and left wheel pairs have equal combined wheel rates. Similarly, a car is not front-stiff or rear-stiff if (and only if) the front and rear wheel pairs have equal combined wheel rates.

Fig 2 illustrates some possibilities.

As explained in Racecar V7N7 and V7N8, a four-wheel chassis has four modes of suspension motion, and it may have different wheel rates in different modes. Consequently, a car may have differing right-stiffness, front-stiffness and diagonal stiffness in roll, pitch, heave and warp.

For example, a typical front-engine, rear-drive car with independent front suspension, a 'live' axle rear suspension, and a fairly stiff anti-roll bar at the front is usually sprung front-stiff in roll and warp, yet rear-stiff in pitch and heave. For roadgoing application, it will not ordinarily be right-stiff, left-stiff, or diagonally stiff in any mode.

Dampers (shock absorbers) also produce roll-resisting forces. Damping force is usually not position-sensitive, although it is possible to make it so. Primarily, damping force is wheel-velocity sensitive. A damper only generates damping force when its shaft is in motion. Therefore, our dampers add roll resistance only when the car has a roll velocity.

When the car is holding a fixed roll angle, the damping forces do not influence dynamic wedge changes (except those caused by bumps). When the car is rolling rightward, the extension (or rebound) forces of the left dampers and the compression (or 'jounce') forces of the right dampers contribute roll resistance. Conversely, when the car is rolling left, the left-side compression and right-side extension properties are what count.

The relationship between a damper's shaft force and its shaft velocity depends on its valving. The force/velocity relationship at the wheel depends additionally upon the wheel to damper shaft motion ratio.

The damper properties that mainly concern us as a roll resistance distribution factor are the 'low-speed' damping forces. As a rule of thumb, that means shaft speed of 2 inches per second or less. Of course, this is a fairly arbitrary dividing line, and the wheel:shaft motion ratio influences the wheel velocity to which a 2in/sec shaft movement will equate.

With most dampers, it is the bleed properties that determine the low-speed forces. A bleed is a fluid restriction which is often externally adjustable, but does not open or close by fluid pressure, except that it may incorporate a lightly sprung check valve to make it function in compression or extension only. If a damper has no bleeds, its intermediate-speed valves will control its low-speed behaviour.

Fig 3 shows force/velocity curves for linear, progressive, and digressive damping. Actual dampers may have complex force/velocity relationships yielding progressive, digressive and roughly linear properties as velocity varies.

The more progressive a damper is, the more sensitive it is to motion ratio. A linear damper obeys an 'inverse square' law. If its shaft moves 0.5in for 1in of wheel movement, it is a quarter as stiff at the wheel as it would be if it acted 1:1. A digressive damper acting 2:1 is more than a quarter as stiff. A constant-rate spring follows the inverse square law, the same as a linear damper. With complex force/velocity properties,

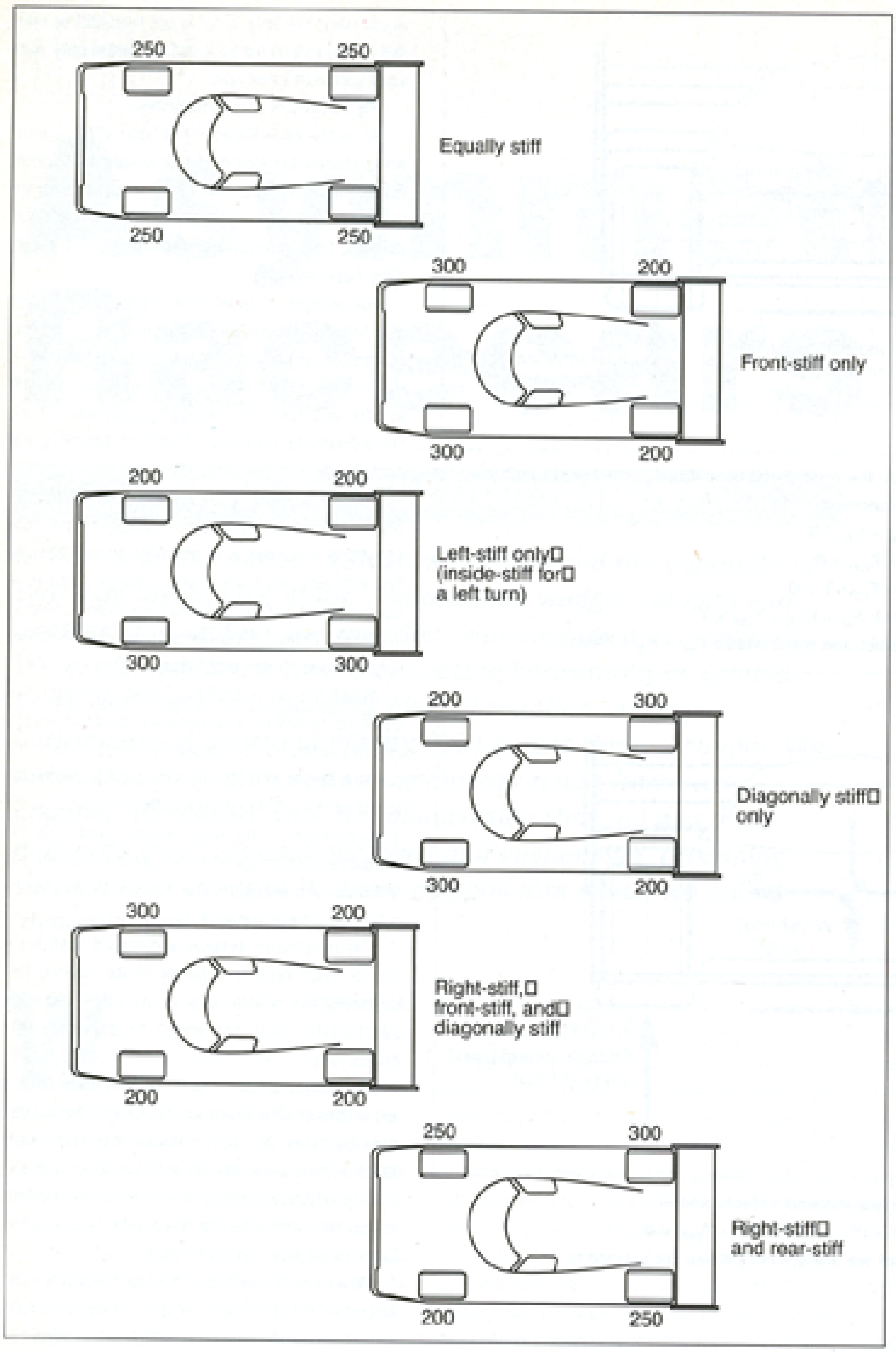


Fig. 2: A Vocabulary for describing suspension stiffness relationships. The numbers represent wheel rates in pounds/inch.

plus position-sensitive motion ratios, we can get some interesting effects!

Pressurised dampers not only damp, but also act as a springing device. Usually, the fluid is pressurised by a gas (normally nitrogen), and we speak of a gas spring effect in these units. Although gas spring forces originate in the damper, they are not damping forces, but auxiliary spring forces. They are position-sensitive, not velocity-sensitive. They do affect steady-state cornering.

Gas springs are rising-rate springing devices. The smaller the gas volume, the more pronounced the rising-rate effect becomes.

Gas spring forces augment individual wheel spring forces. The softer the wheel spring, the greater the relative importance of the gas spring force.

Gas springs are temperature-sensitive. As a gas-pressurised damper heats up, the suspension's wheel rate and ride height increase a little. Again, the softer the other springs in the system, the greater the relative influence of the gas spring.

In a car with diagonally stiff springing, therefore, gas spring temperature-sensitivity can produce some variation in static wedge as the car runs, and as the weather changes. With left-stiff or right-stiff springing, the static roll (or tilt) position will change. With front-stiff or rear-stiff springing, the static pitch (or rake) will change.

None of these effects will be huge, and the latter two should not create significant wedge change. Nevertheless, wedge variation with temperature is a possibility to be considered when diagnosing, or correcting, asymmetrical setups that demand 'chasing' as the race or session proceeds.

Finally, suspension systems produce roll resistance through their geometry. Geometric roll resistance results from lateral movement of the footprints during oppositional motion of right and left wheels (that is, during roll or warp). If the footprints move straight up and down, geometric roll resistance is absent. If the footprints move outboard on the compressing side, the suspension generates anti-roll forces. The greater this lateral motion, the greater the geometric roll resistance.

Many readers will have seen this issue addressed in terms of roll centre height. Many will have seen diagrams such as those in **Fig 4**. The front and rear wheel pairs each have a roll centre. A line connecting these two points is called the roll axis. The sprung mass is thought of as rotating about this axis when it rolls during cornering.

This model is not a perfect representation of actual car behaviour, especially with independent suspension. However, it is useful as a simplified approximation. Geometry that yields a high roll centre invariably produces large lateral footprint movement (lateral scrub) in suspension roll and warp. Lateral scrub and roll centre height are inextricably linked.

Lateral scrub creates 'jacking forces'. On an outside wheel, outward scrub in suspension compression forces the footprint to move against cornering force in order for the car to roll. Consequently, cornering force induces an upward jacking force, opposing suspension compression (see **Fig 5A**).

On an inside wheel, inward scrub in suspension extension also forces the footprint to move against cornering force for roll to occur. As on an outside wheel, a jacking force results – this time downward, opposing extension (see **Fig 5B**).

Note that this is the same phenomenon as in Fig 5A – just backwards.

If the upward and downward jacking forces were equal, roll resistance would occur, and overall ride height change would not. That would make the 'roll centre' model pretty accurate. However, the upward jacking force on the outside wheel is a good deal greater than the downward one on the inside. How much greater the upward force will be depends on how unequal the lateral forces in the tyres are. This, in turn, depends on how unequally they are loaded. It also depends on how unequal their slip angles are. If the wheel pair is toed-in, the outside slip angle is greater, and upward jacking is greater, than if the wheel pair were toed-out.

The higher the roll centre, the greater the net upward jacking. This effect requires us to use low roll

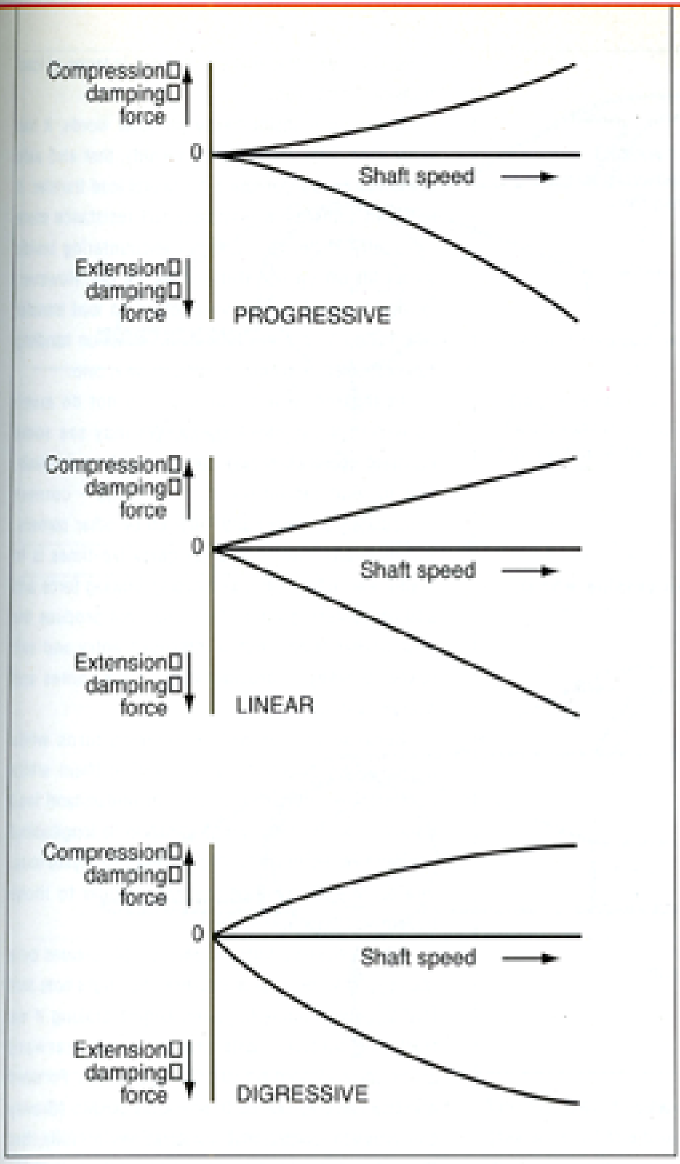


Fig 3: Basic damper characteristics.

centre geometry in an independent suspension. Where class rules require a high-roll-centre suspension at one end of the car, as at the rear of a Formula Vee, we can minimise upward jacking by minimising wheel rate in roll at that end of the car.

Similar forces exist in beam axles, but the axle's rigidity prevents them from lifting the car – at least, up to the point of inside wheel lift. With a beam axle, we can usually use a much higher roll centre, and obtain much greater geometric roll resistance, than with independent suspension. However, we create just as much lateral footprint scrub in suspension roll and warp as we would with comparably roll-resistive independent geometry.

Net upward and downward jacking can occur with beam axles, and it can affect dynamic wedge. We will take these matters up in Part 4.

Returning to the example of a traditional rear-drive road car with live-axle rear suspension, such cars generally have front roll centres near ground level, and rear roll centres near axle height. Therefore, although our hypothetical car is front-stiff in its springing, it will be rear-stiff geometrically. So its total roll resistance will be more evenly apportioned than we might suppose if we looked at its springing or its geometry alone.

Effects of Roll Resistance

All roll resistance, no matter what its origin, affects wheel loads in accordance with the same two fundamental laws:

Total vehicle roll resistance has only a slight effect on total vehicle load transfer.

Front and rear load transfer are directly proportional to total front and total rear roll resistance.

These laws apply not only to the four usual sources of roll resistance, but also to unorthodox devices and strategies such as 'trick' spring mounts, diagonal U-bars, hydraulic interconnections, 'zero-droop' suspension, superimposed suspensions creating inward roll, and computerised 'active' suspension. All roll resistance acts through the tyres. And tyres do not know where their loadings come from.

A Constant-Speed Corner

To illustrate how the factors we have been discussing influence a car, let usconsider a simplified cornering manoeuvre.

The car steers in, proceeds on a constant-radius path for a while, then exits onto a straightaway, all at constant speed. The manoeuvre has transient phases during entry and exit, separated by a steady-state phase.

The damper low-peed properties affect wheel loads only during the transient phases. The geometric and spring-derived roll resistance have an influence that builds, holds, then returns to zero, with cornering force. The greater the cornering force, the greater the dynamic wedge change, due to non-damper roll resistance inequality.

Static wedge, by contrast, is totally independent of cornering force. As noted in Part 1, asymmetrical road racing setups do not normally incorporate static wedge (exceptions may occur in live axle cars – more on this in Part 4). However, static wedge is basic to oval track setup.

Wedge 'tightens' a car, or makes it more inclined to 'understeer'. That is, wedge hurts front wheel pair cornering grip, and helps rear pair cornering grip, due to tyre load sensitivity. Conversely, de-wedging a car makes it 'loose', or inclined to 'oversteer'. A car with equal cornering grip at both ends (proportional to the weight distribution) is said to be 'free', or 'neutral'. Many readers will already understand these concepts.

The tyres do not know, at a particular instant, how much of the car's wedge at that instant is static wedge, and how much is due to dynamic wedge change. The tyres are governed by the sum of the two effects. By controlling how total dynamic wedge varies with conditions, we can control variations in handling balance.

To speak of handling balance variation, we need a little more terminology. If a car tends to understeer more as grip deteriorates, we may say the car 'goes

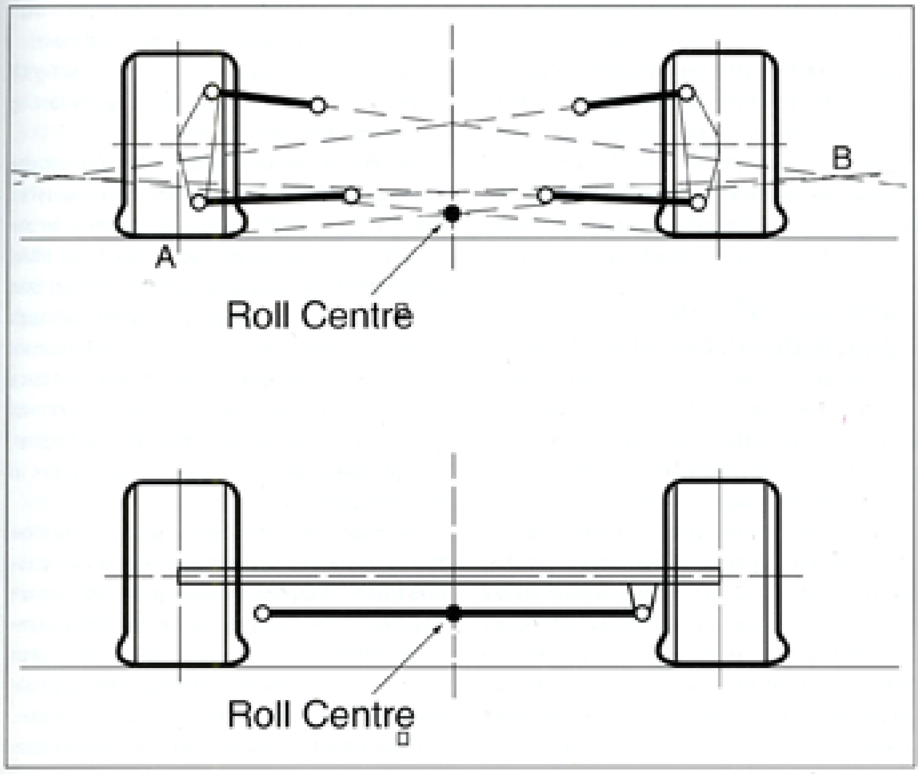


Fig.4: Roll centre locations for typical independent and beam axle suspensions.

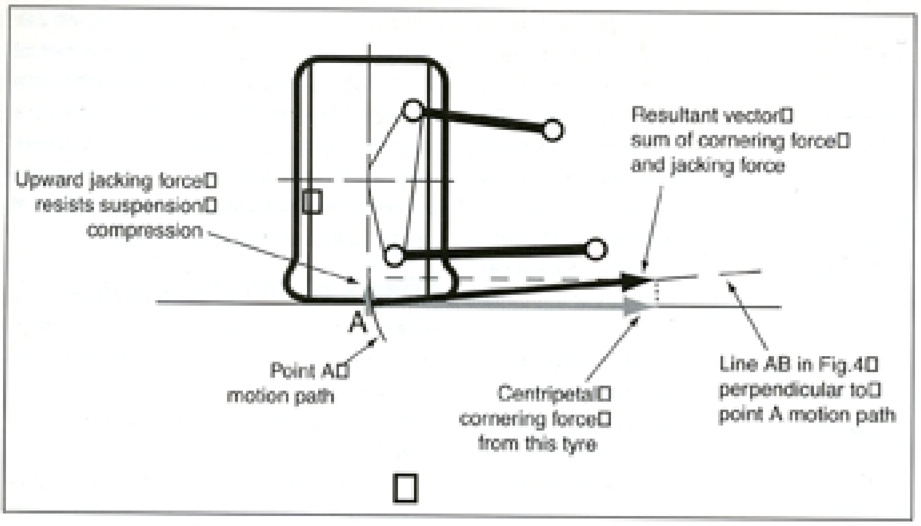


Fig 5A: Origin of geometric roll resistance in upward jacking force on an outside wheel.

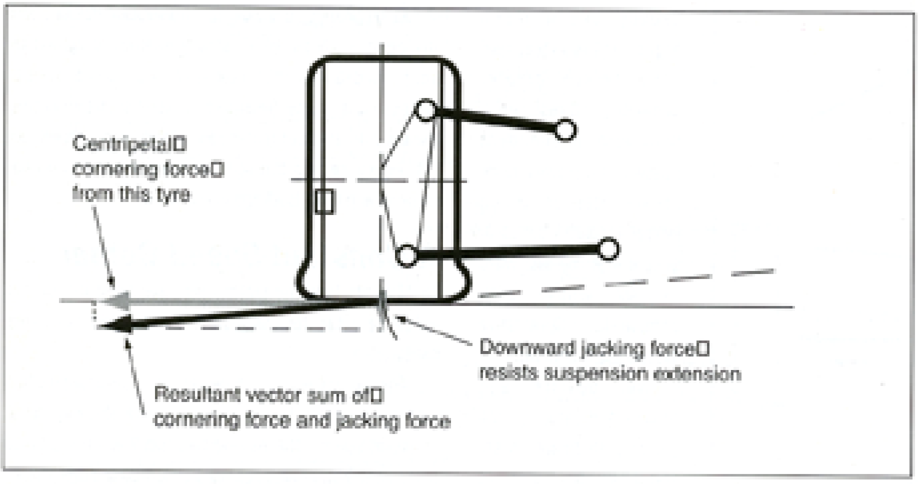


Fig. 5B: Origin of geometric roll resistance in downard jacking force on an inside wheel.

tight on slick'. We may also speak of a car's transient handling as 'loose' or 'tight' on entry or exit.

If we have a tight-on-slick condition, we can improve the car by creating a setup that produces less wedge when cornering forces are modest, with a greater wedge gain as cornering force increases. To obtain this, we would decrease static wedge, and make the overall roll resistance distribution more front-stiff.

For a loose-on-slick car, we do the opposite: more static wedge, with a more rear-stiff roll resistance distribution.

To loosen the car on entry and tighten it on exit, we can make the low-speed damping more rear-stiff. Front-stiff low speed damping yields the opposite effect: tighter entry, looser exit. These approaches are effective on all cars – even road or off-road cars.

On an oval-track car, there are further possibilities. To change entry behaviour only, we can vary low-speed extension force distribution on the left wheels. To change exit behaviour only, we can adjust right extension forces and/or left compression forces.

Low-speed force distribution also affects car behaviour over mid-turn heave disturbances – humps and dips that load and unload all four wheels at once. Most often, it is the times when the tyres unload that trouble us. On these occasions, all four wheels will be moving in extension. Those with the stiffest low-speed extension damping will unload the most. If the rears are stiff, the car goes loose momentarily. If the fronts are stiff, it goes tight.

On a car that goes both ways, we will normally adjust both fronts or both rears in unison. On an oval track, we do not necessarily have to do it this way.

Suppose, for example, that an oval track car momentarily goes loose as it unloads coming off a banked turn. We could stiffen the low-speed extension just on the left front. Not only will the front wheels collectively unload more during extension, the car will momentarily gain wedge as well.

Load Transfer in Longitudinal Acceleration

We have been considering a simplified case in which the car neither loses nor gains speed: longitudinal acceleration is absent. The total front and rear percentages do not change, although the front and rear percentages on each side of the car may change.

We noted that, in this situation, it does not matter whether the car is right-stiff or left-stiff, only whether it is front-stiff or rear-stiff. Before moving on, we might add that springing the car inside-stiff can aid roadholding, because it allows us to spring the heavily loaded outside wheels softer. That will allow the outside wheels to follow bumps better. The inside wheels will be hurt in this regard, assuming we maintain the same total stiffness, but the gain on the outside wheels will exceed the loss on the inside ones. Obviously this matters, but not in terms of load transfer distribution.

If the car is inside-heavy (in other words, it has more than 50% static inside weight), that also aids cornering by compensating for lateral load transfer. It will also allow us to distribute roll resistance more unequally, if desired, without encountering inside wheel lift off on the stiff end of the car. However, inside-heaviness has no direct effect on load transfer distribution, and it has little direct effect on handling balance, when longitudinal acceleration is zero.

In the real world, racecars do not do much cornering at constant speed. We may see some constant-speed cornering around mid-turn on ovals, and in road racing on 'Type 3' turns – corners immediately preceded and followed by other corners. But generally, the way to get quick lap-times is to 'trail-brake' into a turn, decreasing braking force and adding cornering force. This does not produce the best corner times, but it improves entry and exit speed, thereby improving straightaway times and overall lap-time.

Because the car will be entering turns while accelerating rearward, and exciting them while accelerating forward, it is vital to understand load transfer and dynamic wedge variation in longitudinal acceleration. Fortunately, this need not be mysterious. The principles involved are very similar to those governing lateral acceleration.

Fig 6A shows a forward pitch couple created by a rearward tyre force, as in braking. We might note that tyre drag while cornering, and engine braking if we trail the throttle, will create some rearward acceleration without actual brake application. Forward acceleration only comes from engine power. (Gravity can produce longitudinal accelerations on hills, but these do not produce much load transfer or wedge change. *Overcoming* gravity with the brakes or the throttle can generate significant wedge changes.)

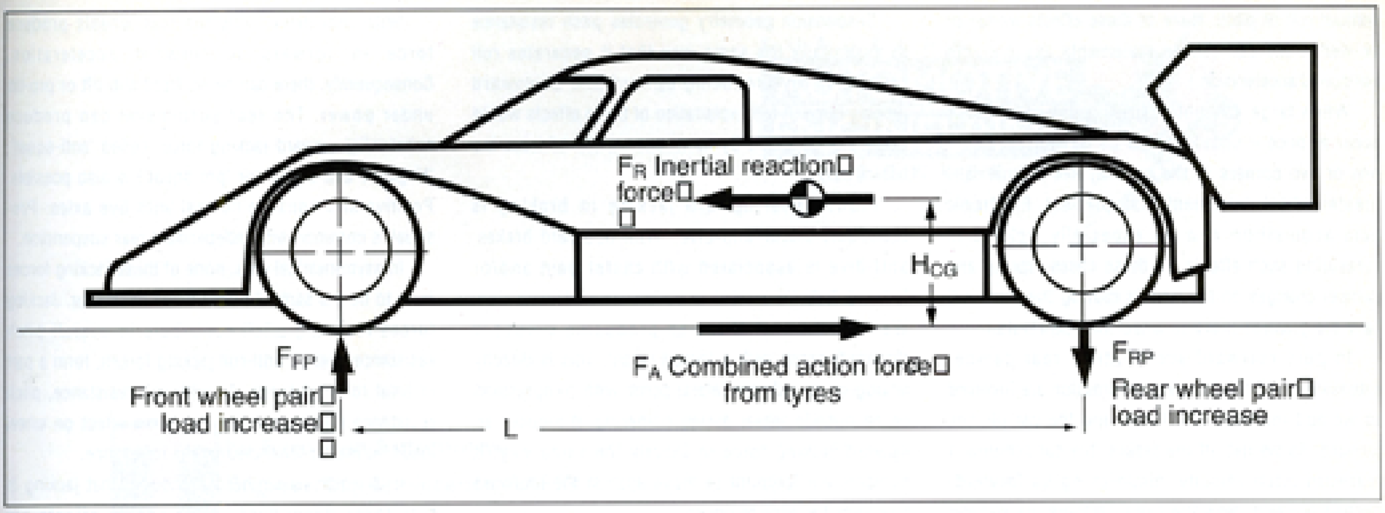
Fig 6B shows the same example, with gravitational effects added (assuming a level road).

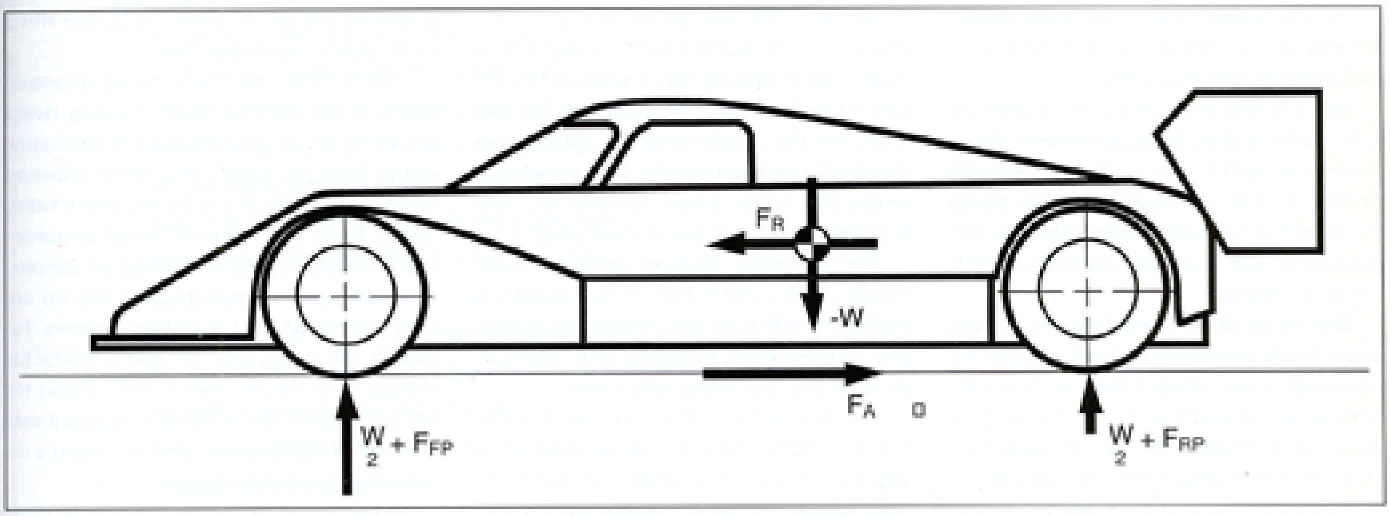
The similarity between these diagrams and Fig 1A and Fig 1B should be obvious. We really are looking at the same thing, turned 90deg.

As with lateral acceleration, total load transfer depends entirely on how much force the tyres generate, how high the CG is, and how far apart the wheels are. For longitudinal acceleration, the wheel separation of interest is wheel base length rather than track width. The CG height is the same, although some rear suspensions can raise it an inch or more under power or lower it in braking. The tyre force shown is equal in both examples. However, a car will generally brake with more force than it will corner, and will generate considerably smaller forces in forward acceleration.

Although total load transfer depends on the three factors above, the suspension determines how the load transfer is apportioned between the right and left wheel pairs. The rule is that wheel pair load transfer is allocated in proportion to wheel pair pitch resistance. This is directly analogous to the way front and rear roll resistance apportion lateral load transfer.

Pitch resistance, like roll resistance, comes from individual wheel springs, inter-connective springing devices, damping, and suspension geometry.





Top Fig 6A: Load transfer in braking.

The sum of the horizontal forces is zero: $F_A + F_R = 0$

The sum of the vertical forces is zero: $F_{FP} + F_{RP} = 0$

The sum of the moments is zero: $F_R \cdot H_{CG} + F_{FP} \cdot L = 0$

The magnitudes of opposing moments are equal: $F_R \times H_{CG} = F_{FP} \times L$

Load transfer depends on tyre force, CG height and wheelbase: $F_{RP} = F_A \times (H_{CG}/L)$

Fig 6B: The same example as Fig 6A, with gravitational effects added. The sum of the vertical forces is zero: $-W + (W/2 + F_{FP}) + (W/2 + F_{RP}) = 0$

As a general rule, road-racing cars will have identical wheel rates on both sides. However, if we ballast a road-racing car to get more than 50% static inside weight for the dominant cornering direction, we may want to use proportionately higher wheel rates on the heavier side. This will maintain similar sprung mass natural frequencies on both sides of the car.

Oval-track cars are almost always right-stiff or left-stiff. Cars running on steep bankings are often right-stiff. The combination of downward (compressive) heave and rightward roll produces large compression movement on the right wheels. Stiff springs are used to keep this movement within liveable bounds. On the left side, the effects of right roll and downward heave are subtractive. Usually, the suspension extends but the movement is relatively small. With soft springs, the movement will be larger, but not dramatically. Therefore, it is logical to use softer springs on the left, simply because we can.

Note that this logic contradicts the previous argument in favour of inside-stiff springing, which makes sense everywhere except steep bankings. The point is that there may be reasons for using right-stiff or left stiff springing which may have nothing to do with load transfer allocation. Nevertheless, such springing always creates wedge changes in longitudinal acceleration.

If the springing is inside-stiff, the inside wheels experience more load change than the outside ones. In rearward acceleration, this de-wedges the car. In forward acceleration, it wedges the car.

Therefore, inside-stiff springing loosens entry and tightens exit. Outside-stiff setups do the opposite: more wedge and tighter on entry, less wedge and looser on exit.

Fore-and-aft suspension interconnections are uncommon in racing. In *Racecar* V7N8, we described various ways in which such effects can be achieved. All such devices can contribute equal or unequal amounts of wheel rate at each corner of the chassis, and all of them essentially act in two modes. They have no effect in the other two modes, if they act through equal motion ratios on the two connected wheels.

When an inter-connective device has unequal motion ratios at its two ends, it does have a small effect in the two modes which it otherwise would not influence. For example, suppose we have a rear anti-roll bar with a short lever arm on the inside wheel and a long arm on the outside wheel. Such a bar will add wedge when the rear suspension compresses synchronously, and it will subtract wedge when the rear wheels extend synchronously. The former effect occurs exiting turns, the latter on entry. Consequently, our bar will tighten exit a bit, and loosen entry a bit, compared with a bar of identical total stiffness acting equally on both wheels.

A similar effect occurs in heave – the other mode the bar normally would not influence.

We will leave it to the reader to ponder the effects of dissimilar motion ratios in various other inter-connective springing devices. The key to understanding all such effects is to determine whether the device makes the car right-stiff or left-stiff, front-stiff or rear-stiff, in the modes that are at play in the situation we are looking at.

As noted earlier, a diagonally stiff car gains and loses wedge in heave. It does not wedge or de-wedge in longitudinal or lateral acceleration. But it does roll a little as it pitches in longitudinal acceleration, and it does pitch a little as it rolls in lateral acceleration. This is sometimes called 'diagonal roll' or 'diagonal pitch'.

A front-stiff or rear-stiff car pitches as it heaves. A car whose stiffnesses are proportional to its static weight distribution, front/rear and left/right, heaves without roll or pitch. None of these effects wedge or de-wedge the car. (These statements assume zero horizontal acceleration.)

When large amounts of roll, pitch, and heave occur at once – usually with a bit of warp as well – one or two corners of the car may see dramatically greater wheel movement than the rest. Electronic data-acquisition can be especially helpful in measuring such effects. In these cases, spring and damper changes on the wheels moving the most will have the greatest effect on wheel load distribution.

In pure rearward acceleration, rear damper extension and front damper compression are involved as we add rearward tyre force, and the car pitches forward. As we get off the brakes, the car acquires a rearward pitch velocity, diminishing its forward-pitched position. Now the front dampers are moving in extension, diminishing their compressed position. The rear dampers are moving in compression, diminishing their extended position.

What is at issue here is not the car's acceleration, as is true for springs, but its acceleration *change* (sometimes called 'jerk'). We add rearward acceleration – then we reduce it. These two distinct phases both occur during corner entry. The first phase may actually be complete before the car starts turning, or it may not.

When we are adding braking force, the car gains wedge if front compression and rear extension are outside-stiff. It loses wedge if these are inside-stiff. When we are reducing braking force, the car gains wedge if front extension and rear compression are inside-stiff. It loses wedge if these are outside-stiff.

Since dampers oppose any suspension motion, they create a lag between stabilisation of vehicle acceleration and stabilisation of suspension position. The car is still gaining forward pitch position for a while after braking force has maximized and it also returns to the original pitch position slightly after braking force has zeroed. This matters, because the dampers are making force as long as they are moving.

The softer the dampers, and the smoother the driver's style, the less the dampers affect corner entry. The stiffer the dampers, and the more abrupt the driving style, the greater the influence of the dampers.

In racing, forward acceleration begins while the car is cornering, increases steadily through the entire corner exit, then falls off gradually while the car is running straight. This means that the direction of pitch motion on exit is the same as during the last phase of braking: rearward.

Correspondingly, the role of the dampers is very similar. If front extension and rear compression are inside-stiff, the car gains wedge. If these are outside-stiff, it loses wedge. Again, the effect is greatest with stiff dampers and abrupt driving.

The foregoing comments assume that the track is fairly flat, and that suspension geometry and other factors allow significant suspension movement at all four corners. Steep bankings can change wheel motions considerably. This makes some wheels disproportionately important. On entry, outside front compression damping counts most. On exit, outside front extension counts most.

Suspension geometry generates pitch resistance in essentially the same way that it generates roll resistance: by producing upward and downward jacking force. A full explanation of these effects would make a good article by itself, but we can address the basics here.

Front wheel upward jacking in braking is commonly called 'anti-dive'. With outboard brakes, anti-dive is associated with caster gain and/or forward hub movement in suspension compression. We may think of these as producing outboard (forward) scrub with a locked wheel. This is directly analogous to outward lateral scrub with compression on an outside wheel during cornering. It induces an upward jacking force in exactly the same way: it compels the footprint to move against the rearward tyre force for pitch to occur.

Similarly, the rear wheels can produce a downward jacking force in a manner analogous to an inside wheel in cornering. This is known as 'anti-lift'. Anti-lift forces can be so great that the rear suspension may actually compress, and the entire chassis will experience downward heave exceeding its forward pitch. For this to work, rear brake force must be limited to a value well short of wheel lockup.

Rear wheel lockup, combined with severe anti-lift, usually creates wheel hop. This is especially troublesome with a live axle, because this design's large unsprung weight encourages wheel locking on any rough patch encountered while braking.

It is possible for front suspensions to create downward jacking force in braking ('pro-dive' or 'negative anti-dive'). Rear suspensions can create upward jacking ('pro-lift' or 'negative anti-lift'). Pro-dive is common on NASCAR Winston Cup cars when the front suspension is compressed. Pro-lift is sometimes seen on Dirt Late Models where the brake calipers are mounted on 'floaters'.

Actual jacking forces depend not only on geometry, but also on brake bias and wheel loading. This is analogous to the influence that toe and wheel loading exert upon jacking forces in cornering.

With rear-drive, only the rear wheels produce force in forward in forward acceleration. Consequently, there can be no front anti-lift or pro-lift under power. The rear suspension can produce substantial upward jacking force, called 'anti-squat'. 'Negative anti-squat' (or 'pro-squat') is also possible. Positive anti-squat is normal with live axles. Pro-squat is common with independent rear suspension.

In asymmetrical cars, none of these jacking forces have to be the same on both sides of the car. Jacking forces resisting pitch contribute to overall pitch resistance, just as anti-roll jacking forces, form a part of total roll resistance. As with roll resistance, pitch resistance distribution has the same effect on wheel loads no matter what creates the resistance.

It is worth remembering, though, that jacking is tyre force dependent, while spring effects are suspension position dependent, and damper effects are suspension velocity dependent.

Jacking forces that reverse normal suspension motion, as with severe rear anti-lift in braking, reverse the normal effects of spring stiffness distribution. Jacking forces that merely reduce normal suspension motion dilute the effects of spring and damper tuning. Jacking forces that exaggerate normal suspension motion increase the influence of springs and dampers.

Increasing roll or pitch resistance from any one source dilutes the effect of changes elsewhere. For example, stiff springs dilute the effect of anti-roll bar changes. A torsionally flexible frame dilutes the effects of all differences in roll, pitch, heave and warp resistance. A rigid structure, conversely, makes a car responsive to suspension changes.

As complex as all these interrelationships are, they are made up of simple effects. With enough thought, they can be figured out.

We have now covered effects that try to make the car roll and pitch, and their effects on handling balance through load distribution and tyre load sensitivity. In Part 3, we will consider effects that act around the third axis – the vertical one – and directly create yaw moments.

DAMPERS ON A BUMPY SURFACE: A SPECIAL CASE

OUR DISCUSSION of the effects of dampers on wheel load distribution in the main body of this article focuses on situations where the dampers are primarily influenced by the car's horizontal accelerations, and by large-scale road contours. In such conditions, damper shaft motion is predominantly low-speed.

When the surface has a lot of small scale curvature – holes, lumps, patches, ripples – this roughness may become the predominant cause of suspension movement. Speed of this movement will vary but, on average, it will be much higher than what we would see on smooth pavement.

The low-speed motions are still there. The suspension's behaviour is the sum of its low-speed and high-speed components. Analytically, we may separate the components.

But – *the dampers don't know this!* They generate force entirely on the basis of their shaft velocity at a particular instant. Their low-speed properties only matter when their shaft speed really is low.

This means that a bumpy surface largely obliterates the damper-related load distribution effects we would otherwise see. The bumps impose a whole new set of rules. Damper-related load transfer is now controlled by the surface, not the driver.

We may see dynamic wedge change over a series of bumps if our dampers have dissimilar high-speed properties. Disproportionately stiff high-speed rebound damping on one wheel will cause that corner of the car to 'jack down' on rough patches. Disproportionately stiff high-speed compression damping will make it 'jack up'. This will dynamically change the car's wedge in the same way as if we made a similar ride height adjustment when setting up the car at the shop. The wedge change will grow and persist while the surface is rough, and rapidly fade when the surface smoothes out.