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BIKE TECH

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AERODYNAMICS

The True Hour Record Holder . . . Is Bracke!

Claude Genzling

Editor's note: Of course, the true hour record holder is Francesco Moser, but a lively debate is underway as to whether engineering or athletic prowess played the greater role in Moser's stunning victory. Claude Genzling explores this question by using the fundamental equation that relates bicycle speed to air drag power loss, combined with a few very plausible, though not completely tested, assumptions. Genzling calculates the theoretical performance of Moser, Eddy Merckx (hour recordman before Moser), Bernard Hinault, and Ferdinand Bracke (hour recordman in 1967) on a variety of bicycles and at both high and low altitudes. The results are truly surprising! Unfortunately, some of the assumptions needed for this calculation are untestable, in a sense. Practical techniques for measuring wheel air drag and tire rolling resistance, for example, are still under development. This does not detract from the value of Genzling's study, but it does point out the areas where more work is needed.

Reprinted from Le Cycle (no. 99), *March 1984, Paris, France. Translation assistance by John S. Allen. Thanks to Eric Hjertberg for bringing this article to our attention.*

In an article published more than four years ago, "Aerodynamics or Light Weight"[1] I estimated that if Eddy Merckx had been riding a Renault-Gitane "Profil" bicycle, he could have ridden 51.470 km in an hour, instead of 49.431 km, with the same energy expenditure. My hypothesis, based on data provided by the manufacturer, was that this bicycle would have reduced by 70 watts the energy needed to overcome air resistance.

Francesco Moser carried out his record attempt on a bicycle even more aerodynamic than the "Profil": a bicycle with its wheels sheathed in plastic fairings. It is interesting that Moser's record distance (51.151 km) is slightly less than the distance I had calculated for Merckx on the "Profil" (51.470 km). It seemed that the question of the aerodynamic bicycle called for further evaluation. I had already collected additional data, notably on the Gitane "Delta e" bicycle.[2] This new data now permits us to calculate the importance of reduced air resistance in Moser's record-breaking ride.

Principle of the Calculations

The power which a cyclist expends in overcoming air resistance is given by the formula:

$$P = \rho \cdot C_D \cdot A \cdot (V/3.6)^3/2$$

where P = power (watts)
ρ = air density (kg/m^3)
C_D = drag coefficient (dimensionless)
A = frontal area (m^2)
V = speed (km/hr)
3.6 = factor for converting km/hr to m/s (See footnote 3.)

[1] *See* Le Cycle, *no. 51, November 1979.*
[2] *See* Le Cycle, *no. 95, November 1983.*
[3] *Editor's note: The original French article gives all speeds in meters per second (m/s). Our translation gives speeds in kilometers per hour (km/hr), because this number tells immediately how far the rider would travel in one hour if he maintains constant speed.*

IN THIS ISSUE

This equation can be rearranged to solve for the rider's speed in terms of the power he expends against air resistance:

$$V = 3.6\sqrt[3]{2\cdot P/(\rho\ C_D\cdot A)}$$

From this second equation, we can compute exactly how much the rider's speed (V) will increase if either the air density (ρ) or the effective frontal area ($C_D\cdot A$) are made smaller.[4] It is precisely these two factors which were crucial in Moser's record-breaking ride:

—air density, which decreased from 1.225 kg/m^3 at sea level to 0.961 kg/m^3 at the altitude of about 2,000 meters in Mexico where Moser achieved the record, assuming equal temperature;

—effective frontal area, which was reduced by improving the position of the rider on the bicycle and by aerodynamic streamlining of the bicycle and its wheels.

The actual values of effective frontal area ($C_D\cdot A$) for typical man/machine combinations are well-known today, thanks to the work of the Aerotechnic Institute of Saint-Cyr-l'Ecole, directed by Maurice Menard.[5] He has provided precise measured data on this subject. For example, we know that Eddy Merckx on a traditional bicycle in the record-seeker's low riding position exhibits an effective frontal area of 0.39 m^2. I have used this and other test data[6] to derive reasonable assumptions about the effective frontal areas of Francesco Moser and Ferdinand Bracke (an exceptional but often forgotten hour-record holder) riding various bicycles.

We are now in a position to calculate the distance which these three champions would have traveled on different bicycles and at different altitudes, given only that their power output would be the same in our fictional record attempts as it was in their real ones.

Actually, there's one additional problem we must face in trying to compare record attempts in Mexico against others in Rome or Milan: altitude reduces the cyclist's oxygen consumption and so reduces his maximum possible power output. We will deal with this physiological question below, after first discussing the purely mechanical effects of air drag and altitude.

Hinault as Experimental Baseline

Since 1979, the Aerotechnic Institute of Saint-Cyr-l'Ecole has been testing a variant of the "Profil" bicycle intended for a possible future hour record attempt by Bernard Hinault. This bicycle, with a 600 mm front wheel and 700 mm rear wheel, is virtually identical to the one which Francesco Moser used in Mexico, except that it lacks fairings on the spokes.[7] We know that Bernard Hinault would have to develop 550 watts at sea level to overcome air resistance at a speed of 50 km/hr on a "Profil" bicycle. We also know the differences in effective frontal area between the traditional bicycle, the "Profil" bicycle, Hinault's "hour record" bicycle (which later became the "Delta e") and Moser's bicycle with its faired wheels. Consequently, we can calculate the power consumed by air drag with Hinault riding each of these cycles:

$$P = 1.225(50.000/3.6)^3(C_D\cdot A)/2$$
$$P = 1{,}641(C_D\cdot A)$$

From this, we have derived Table 1, which applies to Bernard Hinault riding at 50 km/hr at sea level. The faired wheels alone achieve a savings of approximately 18 watts, reducing by half the power consumed by "aero" wheels currently used in track racing. Although the figures in Table 1 are only approximations, they can be used as the basis for our further calculations.

Table 1. Bernard Hinault riding at Sea Level*

Bicycle	Power (watts) consumed by Air Drag	Effective Frontal Area (m^2)	Speed (km/hr)
traditional	591	0.36	50.000
"Profil"	550	0.34	50.000
"hour record"	510	0.31	50.000
"hour record" with faired wheels	492	0.30	50.000

*Air density ρ = 1.225 kg/m^3

[4]*Editor's note: The term "effective frontal area" refers to the product of drag coefficient (C_D) times frontal area (A). In this article, the values quoted for effective frontal area pertain to the combination of bicycle plus rider. Effective frontal area is perhaps the most useful single number for specifying the bicycle's aerodynamic qualities, for two reasons.*

First, it accounts for both the shape of the bicycle/rider combination (through the C_D term) and its size (through the A term). Second, experimenters can easily measure effective frontal area with simple instruments, but have great difficulty in measuring either drag coefficient or frontal area separately. The physical units of effective frontal area are square meters (m^2) in the SI system, and square feet in the English system.

[5]*See* Le Cycle, *no. 52, December 1979.*

[6]*Editor's note: For those who work with English units instead of SI units: the effective frontal area of the least streamlined bicycle/rider combination quoted in this article (0.39 m^2 for Merckx on a traditional bicycle) equals 4.20 ft^2 in English units. The most streamlined combination mentioned (Hinault on "hour record" bicycle with faired wheels) has an effective frontal area of 0.30 m^2, which equals 3.23 ft^2. By comparison, a tourist riding in the upright position has an effective frontal area of about 4.5 ft^2, while recumbent cycles with full-coverage fairings (such as the Vector single) have an effective frontal area of about 0.5 ft^2.*

[7]*Editor's note: This bicycle will be designated the "hour record" bicycle throughout this article.*

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Table 2. Eddy Merckx riding in Mexico*

Bicycle	Power (watts) consumed by Air Drag	Effective Frontal Area (m^2)	Speed (km/hr)
traditional	485	0.39	49.431
"Profil"	485	0.37	50.306
"hour record"	485	0.34	51.744
"hour record" with faired wheels	485	0.33	52.262

*Air density = 0.961 kg/m^3

Table 3. Francesco Moser riding in Mexico

Bicycle	Power (watts) consumed by Air Drag	Effective Frontal Area (m^2)	Speed (km/hr)
traditional	441	0.38	48.303
"Profil"	441	0.36	49.182
"hour record"	441	0.33	50.629
"hour record" with faired wheels	441	0.32	51.151

*Air Density = 0.961 kg/m^3

Extrapolation to Other Racers

If we know the $C_D \cdot A$ of a racer on his bike, we can calculate his speed from the power he develops; conversely, we can calculate his power from his speed.

When record-seekers used bicycles with very similar coefficients of air resistance, the record belonged to whomever was strongest, given only that his position on the bicycle was sufficiently aerodynamic. But today, this is no longer so, because the new design of competition bicycles is capable of reducing power demands on the rider by as much as 15 percent at the same speed of 50 km/hr.

If we accurately estimate the $C_D \cdot A$ of Eddy Merckx, Francesco Moser, and Ferdinand Bracke, we can calculate the power which they had to put out to achieve their records, whether in Mexico or, as in Bracke's case, in Rome. For Eddy Merckx—who is, apparently, interested in the question himself—this is very easy, because he himself has given us his value of $C_D \cdot A$, 0.39 m^2. For Francesco Moser, who is taller than Bernard Hinault and who doubtless has a more aerodynamic riding position than Eddy Merckx, we will hypothesize a $C_D \cdot A$ of 0.38 with a traditional bicycle. Given that small differences in bicycle frame size have very little effect, we can use the $C_D \cdot A$ of Bernard Hinault on his highest-performance bicycle to estimate that of Francesco Moser on his strange machine: 0.30 for Hinault, and so 0.32 for Moser; the difference between the two extreme values is preserved. Finally, we will hypothesize a $C_D \cdot A$ of 0.37 for Ferdinand Bracke, since, like Eddy Merckx, he used a traditional bicycle and was of intermediate size, between Moser and Hinault.

If we neglect rolling friction, we can easily compare the performances of Francesco Moser and Eddy Merckx, since these both took place in Mexico. But is it legitimate to neglect rolling friction? We think so, if we are simply trying to calculate the distance Moser would have ridden on Merckx's bicycle, or vice versa. The power necessary to overcome rolling friction—approximately 40 watts in the case of Eddy Merckx—varies no more than 8 percent as speed rises from 48.6 km/hr to 52.2 km/hr. The difference is about 3 watts, negligible in comparison with the effects of changing $C_D \cdot A$.

Comparison of Moser with Merckx

The average power Eddy Merckx required to overcome air resistance on his record-breaking ride in Mexico at 49.431 km/hr is given by:

$$P = 0.961 \cdot 0.39 \cdot (49.431/3.6)^3$$
$$P = 485 \text{ watts}$$

We can also calculate the speed which the Belgian champion would have reached had he ridden any of the three other bicycles which we discussed earlier. We assume the same power, 485 watts, as in his actual record performance. The results given in Table 2 show that King Eddy, given the preparation he had in 1972, would have become the holder of a 52 km hour record if he had ridden Francesco Moser's bicycle. It is likely that Merckx would have pushed the record even further if, in addition, he had had Moser's three months' training program and medical assistance. We do not hesitate to affirm that Eddy Merckx would have exceeded 52 km, since our air resistance calculations have not accounted for two factors which considerably augmented Francesco Moser's performance, namely:

—plastic coating of the track, which saved him approximately 15 watts—half the net rolling resistance of Moser's ultra-narrow 17 mm tubulars;

—weighting of the rear wheel, giving it a flywheel effect to help with pedaling through the "dead center" positions, thus permitting a higher gear ratio.

We remember that Eddy Merckx started as if in a kilometer race, quickly putting himself into oxygen debt, and he struggled energetically for the remainder of the hour—while Francesco Moser modulated his efforts according to a computerized plan, especially in his first attempt. It would not be a bad bet that Eddy Merckx, given these same conditions of medical assistance, would have exceeded . . . 53 km! On his traditional bicycle, Eddy Merckx rode the last lap at 52 km/hr, putting out a power of 565 watts to overcome air resistance. With Moser's bike, this would have driven him to 54.750 km/hr. To finish in a sprint after an hour of effort in a state of muscular asphyxiation says a lot about the deep resources of the exceptional champion who was Eddy Merckx.

Conversely, we have computed the distance that Francesco Moser would have covered with Eddy Merckx's bicycle, and with the assistance of his own medical team: that is to say, all other things being equal, Francesco Moser would have barely exceeded 48 km, as shown in Table 3. Are we too harsh in assigning the $C_D \cdot A$ of 0.38? The $C_D \cdot A$ of

Bernard Hinault on a traditional bicycle, 0.36, is certainly lower than that of Moser on the same bicycle; but if we use it anyway to avoid "bias," we find that "Cesco" would have only flirted with Merckx's record, give or take a few meters.

The Role of Altitude and Ferdinand Bracke

We will now attempt to compensate Merckx's and Moser's records for the altitude of Milan, approximately 2,000 meters lower. In addition to accounting for the changes in drag force due to changes in air density, we will also consider that Merckx's and Moser's riding power was diminished by the 2,000 meter altitude, by an amount that depends on the length of their period of acclimatization.

Eddy Merckx made his record attempt only a few days after his arrival in Mexico. Under these conditions, his power output was reduced by 7 percent, according to Doctor Jean-Pierre de Mondenard, in an article in *Le Cycle*[1]. Merckx's total power output in Mexico can be estimated at 525 watts (485 watts air drag plus 40 watts rolling resistance). It would have been 7 percent higher in Milan or Rome, or about 565 watts, of which 525 watts would be consumed by air resistance. (In the November 1979 issue of *Le Cycle*, we estimated Merckx's total power as 570 watts, very nearly the same.) With 565 watts of muscle output, Eddy Merckx would certainly have ridden farther than 46.800 km in an hour.[8]

This is indeed a surprise, since it falls in between the records of Roger Riviere, who rode 46.923 km in 1957 and 47.347 km in 1958! The explanation is that Eddy Merckx would have paid very dearly at sea level for his too-rapid start and his struggle to beat the clock; in fact, in Mexico, he was saved by the altitude.

Suddenly, in this light, the performance of Ferdinand Bracke, who covered 48.093 km in Rome in 1967, stands out in bold relief. How would Bracke have done in Mexico? A quick calculation establishes his total power output in Rome at 580 watts, including 540 watts against air resistance, with a $C_D \cdot A$ of 0.37. Without acclimatization, in Mexico, he would have ridden 50.800 to 51.000 km in an hour on his traditional bike, and from 53.300 to 53.600 on Moser's bike. Was Louis Caput right, then, when he stated that, in his opinion, the true holder of the hour record was Ferdinand Bracke?

And how would Moser have done in Milan? If we grant him the same 7 percent power

[8]*Author's note: For a more precise calculation, we would need to know what Merckx's actual rolling resistance was in Mexico, compared to what it would have been in Milan or Rome.*

AERODYNAMICS

Spoke Drag

Glen Brown

Editor's note: Bicycle wheel air drag has suddenly become a hot topic. Bicycle air drag, as opposed to rider drag, has traditionally been waved off as minor. But in January, Francesco Moser proved otherwise. Moser demolished Eddy Merckx's hour record on a radical looking bicycle built for minimum wind resistance. Far and away the most important wind-cheating feature on this bike was its solid disk wheels. Many people believe that without those wheels, Moser would not have broken the record.

While the debate about how much the equipment really did help Moser will probably never be settled, the evidence for minimizing bicycle wheel air drag is mounting. Many Olympic teams will come to Los Angeles prepared to equip their bicycles with solid disk wheels. If disk wheels are not allowed, the teams will use traditional spoked wheels built with as few spokes as possible. Wheel air drag is suddenly big stuff in the eye of some of the world's leading bicycle engineers.

Our own Olympic team will have a set of disk wheels in reserve, their merit bolstered by the wind tunnel testing of wheel aerodynamics conducted by research scientists like Chester Kyle. Data from this testing are currently held under tight security wraps, but it is clear that wheel spokes are responsible for a major portion of wheel drag. We hope to present this Olympic data in future issues of Bike Tech, but for now we'll have to whet your appetite with an interesting "back of the envelope" calculation done by aerodynamicist Glen Brown on the magnitude of spoke drag.

There is plenty of concern these days about spoke drag. Fewer spokes, flat spokes, and wheel disks are becoming the trend on special competition bicycles. Francesco Moser stirred up great controversy when he rode a bicycle equipped with solid disk wheels during this successful attempt at the hour record earlier this year. Even sport-touring bicycles are appearing with only 32 spokes. But how significant is spoke drag? Is spoke drag a large enough proportion of the total bike/rider drag that all performance-minded cyclists should consider methods of reducing it, or does it comprise only a minor percent or two that need concern only top kilo riders? This article presents a mathematical estimation of spoke drag that will put its contribution to total drag in perspective.

Wind tunnel testing is the best way to derive numbers for the magnitude of spoke drag. But wheels are tricky to test because they rotate as well as translate through air. (Readers of *Bike Tech* will recall the special test jig built to measure the aerodynamic performance of the Roval wheels in the December 1983 issue.) Rotating wheel drag is especially difficult to measure directly; even in a wind tunnel a special rig is required to measure the direct drag on the wheel separately from the torque required to keep it spinning.

But spoke drag is straightforward to calculate if you ignore interference effects. That is, you must ignore the disruption of the flow passing the spoke caused by the other components of the bicycle such as the tire, rim, and fork blades. It turns out that these effects are smaller than you'd think. The relative wind is greatest when the spoke is vertical above the hub, but here interference from the rim is the least. Rim interference is greatest in the forward horizontal spoke position, but at this point, the spoke presents virtually no frontal profile so only its rotational velocity component is a source of drag. The rotational velocity component is unaffected by rim interference.

The air flow past the spoke ranges from approximately zero velocity to twice vehicle speed and varies cyclically through each revolution. Luckily, spoke diameter is small enough that, even at twice vehicle speed, the spoke never approaches a transition Reynolds number.[1] In short, the drag on each smaller element of spoke is propor-

[1]*A Reynolds number is a value assigned to an object of aerodynamic interest. It is calculated from the ratio of one dimension of the object (e.g., the diameter of a cylinder) and the relative velocity between it and the fluid, to the viscosity of the fluid.*

Within a family of common shapes, like cylinders, different diameters and/or relative velocities result in proportionally different Reynolds numbers and drag values. This proportionality dependence is called the scale effect. *Scaling is vital in aerodynamic work because it allows the measured drag for one cylinder at some relative velocity to be scaled up or down to predict the drag of another cylinder under different conditions.*

As long as the Reynolds number of a cylinder is below a critical value, scaling works. But over this value, the flow characteristics of the fluid around the cylinder change drastically and drag cannot be predicted by scaling. In the case of spokes spinning on a wheel through air, their Reynolds numbers are well within the safe range.

loss in Mexico—biasing the results in his direction since the acclimatization after three weeks considerably reduces the power loss—he would have ridden no more than about 48 km even given the luxury of his advanced machine.[9] He would have ridden about 45.5000 km on Fausto Coppi's bicycle; Coppi might have held onto his earlier record!

The Pioneering Italians

Our evaluations are only approximate, since we have neglected certain factors, and we do not know the exact measured performance of the racers we have studied.

However, the errors should hardly exceed 400 meters, preserving the relative positions. We have undertaken several recalculations, and Maurice Menard has confirmed the approximate accuracy of our figures. According to him, the cumulative advantages of the aerodynamic bicycle and of the altitude in Mexico account for a gain of approximately 5 km in the hour record. Since it's very unlikely that anyone would have exceeded 53 km in Mexico, Bracke's record of more than 48 km in Rome is really the "true" record. Bravo Louis Caput!

Our analysis in no way detracts from Francesco Moser's accomplishment in overturning the world hour record twice within four days. He did this in keeping with the ancient ideal of heroism, unifying intelligence, courage, and athletic prowess: an open mind is necessary to lend trust to a team of scientists who design equipment and strategy; much courage is necessary to keep going in the face of wind drag and fatigue; and athletic talent as well as physical preparation is necessary to ride at high speed regardless of any difficulty.

In France, we have reason to regret that Renault-Gitane has done nothing to help Bernard Hinault overturn Eddy Merckx's first record. In 1979, Renault-Gitane set off the trend in aerodynamic innovations, with the "Profil" bicycle. In 1980, plans for the "hour record" bicycle, which was to supplant this earlier model, were ready. But in 1984, it is the Italians who are using this technology in a pioneering way. This departure represents a new possibility for the Italian bicycle industry.

And that's nothing for a Frenchman like me to crow about.

Author's note: The purpose of this article is to nourish thought and increase awareness, but there is no intention to establish an artificial hierarchy of performances or racers.

[9]*Author's note: Lack of a plastic coating on the track in Milan or Rome would have put Moser at a further disadvantage. Note that Moser went approximately 46.400 km/hr during his training in Milan.*

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tional to the square of the normal component of the element's relative velocity.

(The relative velocity of a small element of spoke is composed of two parts: one part due to forward bike velocity and one part due to wheel rotation. This can be expressed as: $v = V \cos \Theta + Vr/R$, where Θ is the spoke angle (vertical=zero), r is the distance from the wheel center, R is the spoke length, and V is the bike speed.)

Total spoke drag is proportional to the square of the relative velocity averaged over the length of the spoke and around one wheel revolution. This is expressed as the double integral:

$$\overline{v}^2 = \frac{1}{2\pi R} \int_0^{2\pi} \int_0^{R} v^2 \, dr d\Theta.$$

Since the spoke extends from the center of the wheel to the edge of the tire, the integration can be simplified, which yields the interesting result that,

$$\overline{v}^2 = 5/6 \, V^2.$$

This means that the drag of the spokes on a rotating wheel is 5/6 of the drag of the same spokes when the wheel is held broadside to a wind of the same speed.

A bicycle with 72 spokes of 0.075 inches diameter, each 11 inches long, will have an effective frontal area (using $Cd = 1.2$; an accurate drag coefficient for cylindrical spokes) of 0.50 square feet.

To put this value in perspective, consider that a mounted, fully crouched rider has a drag area of approximately 3.2 square feet. Of that total, at least 75 percent is due to the rider, leaving around 0.8 square feet of drag area owing to the bicycle. Spoke drag therefore appears to be over half of the drag of the bicycle!

I find this result incredible. While this calculation is simplified and does tend to overestimate, the numbers are just too large to ignore their significance. Those little buggers really churn up the air!

An interesting result of the integration is that 60 percent of the spoke drag comes from forward velocity and 40 percent from rotation. The two effects are mathematically separable because the vector cross-product of the two velocities drops out of the integration. Therefore, in a wind tunnel test, even if the wheels are spun for the test, the torque to rotate the wheels has to be measured (a real pain) and added to the drag, or else spoke drag will be underestimated by 40 percent.

How much did the wheel covers help Moser in his recent hour record? Not only was there no spoke drag (except perhaps for some internal pumping) but tire/rim drag was also reduced. I figure his aerodynamic advantage was perhaps 15 percent. That's 15 percent of the total drag! If wheel covers are now allowed in competition, I predict many records will fall quickly.

Another interesting implication is that small-wheeled bicycles should have a five to seven percent advantage just because of their shorter/fewer spokes. This aerodynamic power savings easily offsets the slightly higher rolling resistance of small wheels.

It's evident that any performance-oriented cyclist should strive to reduce spoke drag; its contribution to total drag is apparently second only to the rider's own drag. Wheel covers should give the largest reduction in drag, although their sensitivity to crosswinds may make them an unwise choice for normal road use. More modest savings are possible by using conventionally spoked wheels with fewer spokes. Or, following the lead of the Roval wheels, drag can be reduced not only by using fewer spokes, but also by using ones that are flat in cross section and laced radially.

I must add a cautionary note about reducing wheel drag by reducing the number of spokes in your wheels. A well-built 36-spoke wheel is a sturdy structure; reducing the number of spokes without compensatory changes in other parts of the wheel—like the rim—will jeopardize the wheel's strength. You may invite wheel collapse in pursuit of improved aerodynamics.

Eric Hjertberg feels that both 32- and 28-spoke wheels are acceptable for road racing, but that 28-spoke wheels should be used only by light riders or by riders in hill climbing and time trialing events. He asserts that the two most important variables that determine wheel strength are how well the wheel is built and what rim is used. A rim should be chosen with rider weight and road conditions in mind. Heavy riders and rough roads require stronger, more rigid rims that will necessarily be heavier, have a deeper cross section, and/or be made of a higher strength aluminum alloy.

And how about 24-spoke wheels? Hjertberg says that only expert riders need apply. Great discretion is needed when using these wheels on the road; their durability will be limited no matter who the rider is or what the road conditions are. Many rims are drilled for 24 holes, but Hjertberg suggests that you ask around about which rims are reliable before buying. Don't, he concluded, short-circuit the intentions of the rim designer by lacing up a 36-hole rim with only 18 spokes. All you'll end up with is a very dangerous wheel.

Painting With Imron, Part II

Les Lunas

Imron® paint is a polyurethane enamel developed and marketed by DuPont for industrial applications in which a glossy, easily cleaned, and long-lasting coating is needed. Imron can be applied with conventional spray-painting equipment but the non-industrial use of Imron is not officially endorsed by DuPont because, during pouring, mixing, and spraying, Imron releases isocyanate vapors which are very harmful if inhaled in any quantity. Only rigorous attention to air filtering and ventilation will guarantee a painter's health. (See Harvey Sachs's sidebar following this article on the risks, and precautions necessary, when working with Imron paint.) Its physical properties—toughness, high gloss, good adhesion, and resistance to weathering, fading, and common solvents—can exceed those of the high-quality, but more equipment-intensive, baked-on enamels. Since these desirable qualities can be achieved without a baking oven, Imron has become a very popular paint among custom framebuilders and automotive paint shops, in spite of its health hazards.

Cost

Imron is an expensive paint, costing the framebuilder an average of $27 a quart. Depending on the method of application and the particular color used, a frame painter can paint anywhere from three to five frames per quart of Imron. (Metallic colors don't stretch as far as solid colors. Also, only about two ounces of paint ever make it onto a frame; the balance ends up in the spray booth.) Including the primer paint, the clear top coat, and other expendable items like sandpaper and respiratory mask filters, a complete frame paint job costs the painter/builder about $30 in materials.

Imron is currently available in over 3,000 colors but, because of its intended use, many of the colors are industrial hues of blue, green, and yellow. Flamboyant colors, such as candy-apple red and pearlescents, are not available. Hence, some frame painters use other brands of enamel paint with only a clear top coat of Imron.

Polyurethane

Imron paint is classified as a two-part catalyzed[1] polyurethane enamel. Polyurethanes are a group of chemical compounds that have exceptional toughness and reasonably good strength; they are widely used in components that must resist heavy-duty abuse. Skateboard wheels, hockey pucks, and bumpers on tugboats, autos, and airplanes are made of polyurethanes. For paints, polyurethanes represent a major improvement over earlier paint formulations, which tend to be brittle and are likely to crack under impact.

The toughness of polyurethane paints can be explained by looking at their molecular structure. On a microscopic level, polyurethane molecules resemble a tangled mass of long, tightly coiled springs. When subject to an impact load, these molecules simply flex and then spring back into their original position. The tangles between adjacent molecules are actually chemical bonds called cross-links, which provide strength to the paint film and prevent it from tearing. These cross-links in the paint are not formed until the two separate liquid components are mixed together.

Imron is not the only catalyzed polyurethane paint on the market. I have found four other paints that have about the same impact resistance as Imron. They are: *Sunfire 421,* made by Sherwin-Williams Co. of Cleveland, Ohio; *Nitram,* made by Martin-Senour Co., also of Cleveland, and sold by NAPA automotive distributors; *Delstar* paint with *Delthane* catalyst, made by Ditzler Automotive Finishes Division of PPG Industries, of Troy, Michigan; and *Miralon,* made by Acme Automotive Finishes Co. You must add reducer (solvent) when mixing these paints for spray application, whereas Imron's instructions specify that no reducer is to be used. Your final choice between these paints should be governed by cost, color choice, and availability.

Curing

The curing reaction of Imron paint begins as soon as the catalyst is mixed with the binder. Most other paints cure in a different manner, by processes of oxidation, baking, or reaction with moisture, and do not begin to cure until the paint has actually been applied to the surface. The "pot life" of Imron is usually about eight hours, but can be a bit longer in cold weather. Toward the end of its pot life, the batch of Imron will quickly gel into a thick mass.

On the bicycle frame, Imron takes a long time to cure completely. Although the painted surface will feel dry to the touch three to four hours after spraying, you should not handle the frame or attach components for at least seven to eight days, as the paint's mechanical properties are slow to develop.

You can shorten the curing time of Imron by heating the paint in a warm-air oven at 180° -200° F for about 1½ hours. Although heating is no longer officially recommended by DuPont, my experience is that heating can cut the overall curing time of Imron in half, with no loss in physical properties. (Note that traditional alkyd enamels are "baked" at higher temperatures than I use for heating Imron.) After heating, the frame should be allowed to cool for at least twelve hours before being handled. Infrared "heat lamps" should definitely not be used to speed the curing process, since they can cause hot spots and discoloration.

Clean Surfaces

As we saw in Part I of this article, proper preparation of a frame for painting begins with the bare steel. The steel must be free of all rust, grease, and other surface pollutants. Steel frames are usually cleaned by either particle blasting or acid pickling. Both methods are effective in removing all surface contaminants as well as providing a sufficiently rough surface for mechanical bonding, but as Mario Emiliani pointed out in his series of articles on frame surface finishes, care must be exercised when using either process. It is easy to pit or damage the steel surface by using a coarse particle grit, a too-high particle velocity, or leaving the frame in the pickling tank too long. (See the December 1983 and February 1984 issues of *Bike Tech*).

Phosphate Coat

In Part I, we also saw that the best initial coat to apply is a phosphate coat, either iron or zinc phosphate. A phosphate coating provides the first layer of corrosion protection, adds a buffer layer between the inflexible steel and the relatively flexible primer coat, and also provides a good "tooth," a sufficiently rough surface to which the primer coat can mechanically bond. This tooth develops as jagged iron or zinc phosphate crystals grow onto the steel surface as the phosphate coating dries (see Figure 1). Iron phosphate is most commonly used for bicycle frames, because it can be applied at room temperature, while zinc phosphate requires a carefully controlled hot bath solution. The zinc formulas, however, are somewhat more protective. Precautions should be taken in handling the frame after phosphating to pre-

[1] *A catalyst is a chemical substance that initiates a chemical reaction and drives it to completion under conditions in which the reaction would not normally occur. Typically, catalysts are used to make reactions occur more quickly and at lower temperatures than would otherwise be possible.*

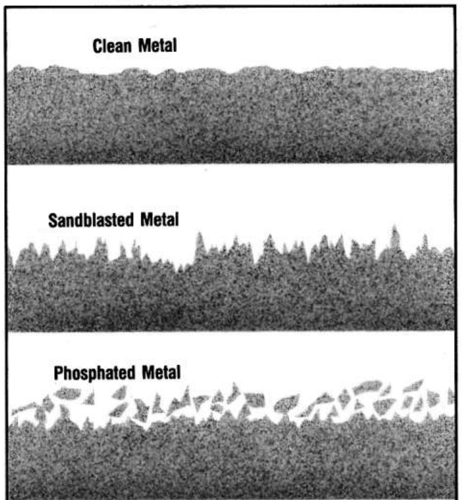


Figure 1: Effects of sandblasting and phosphating on microscopic metal surface.

vent contamination by grease, oils, or salts (from sweat). The typical phosphate layer is attacked by moisture in the air, and so must be prime-coated with paint within 24 hours.

Primer Coat

The primer coat is designed to do various tasks. First, it provides the next layer of corrosion protection; second, it provides a uniform color base for the top coat; third, its heavy pigments will fill in small surface irregularities to smooth out the surface for the top coat; fourth, the primer has a porous surface that provides the necessary "tooth" for the top coat to grab onto; fifth, it should be elastic; lastly, the primer may help to seal in small amounts of chemical contaminants and oils that might have been picked up at an earlier step. The top coat will not hide any imperfections in the previous layers, and so the prime coat is sometimes given a light sanding to remove any roughness or spatter marks. With a smooth, clean primer coat, the top coat will go on smoothly and will adhere well.

It is important to use compatible primer and top coats. The two paints must be chemically compatible for best adhesion. Also, they must have similar elasticity, i.e., the coats must flex the same amount or else they will separate under impact and the adhesive bond between them will break (see Figure 2). When this occurs, the top coat separates from the primer, a small blister forms, and eventually the top coat chips away, exposing the primer.

DuPont's *Corlar* two-part epoxy primer, approximately $35 a gallon, is probably the best match for use with Imron. I have also used DuPont's *Multi-Purpose Primer/Surfacer 1008,* which DuPont claims is compatible with Imron, but I've found that Imron adheres much better to Corlar primer. Compatibility is also important if you plan to use Imron as a clear final coat on top of a non-Imron color coat. Make sure that your color coat is physically and chemically compatible with Imron; otherwise, you might find that the finish is less chip-resistant than you'd like.

Painting with Imron is not especially difficult, but it is different in some important ways from painting with traditional alkyd enamel paints. If you take the time to do it right, you'll be very pleased with the glossy, durable finish that comes with an Imron paint job.

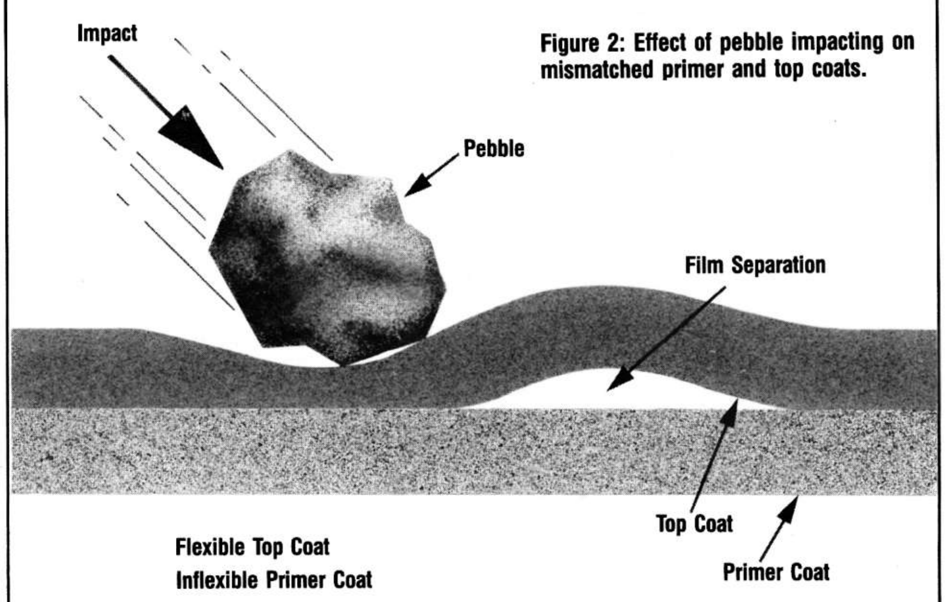


Figure 2: Effect of pebble impacting on mismatched primer and top coats.

Using Imron Safely

Harvey Sachs, Ph.D.

Imron is a polyurethane enamel that has become famous for its wet look, durability, and resistance to damage. However, getting first class results without doing damage to yourself requires a lot more than deft use of the spray gun. You have to read the fine print on the paint can.

The Imron activator (catalyst) contains polyisocyanates which in large quantities can kill you and in small quantities can cause local skin and mucous membrane irritation, permanent eye damage, and other serious health effects including asthma-like symptoms and permanent sensitization. That's why the label on the can reads:

DANGER!
VAPOR AND SPRAY MIST HARMFUL.
MAY CAUSE LUNG IRRITATION AND ALLERGIC RESPIRATORY REACTION.
USE ONLY WITH ADEQUATE VENTILATION.

But how much exposure to isocyanate vapor is too much? What constitutes adequate ventilation?

Federal regulations limit peak exposure to isocyanate vapor to 0.02 parts per million—this is the equivalent of one-inch distance in 800 miles! Basically, once you smell the vapor, you are getting too much. Vapors will be given off when you pour, mix, and spray the stuff, so don't even open the activator can without wearing an approved mask and having adequate ventilation. Remember, some people are more sensitive to these vapors than others, and you can't predict your own reactions in advance. If you know that you are chemically sensitive or asthmatic, don't do any spray painting.

Minimum Standards

I strongly recommend that anyone wanting to paint with Imron adopt the following minimum standards:

1. Always ventilate with enough fresh air. This means having a paint booth equipped with a fan capable of supplying an air flow of at least 100 feet/minute across the work area and exhausting this air to the outside. Included in this ventilation system are paint arrestors, which are filters in the exhaust duct that trap over-spray paint particles.

For a work area with a four-foot by six-foot paint arrestor area, minimum ventilation requires an air flow rate of 4ft × 6ft × 100ft/

min = 2400 cubic feet per minute. I recommend doubling that rate to 5000 cfm. The fan motor must be out of the air stream to avoid igniting the volatile vapors. (Typical bathroom and kitchen fans supply only 50-150 cfm and sit right in the air stream; they are much too small and very dangerous.)

2. Use an approved respirator or mask. For occupational use, a supplied air (SCUBA-type) system which brings a steady flow of fresh air to the worker's face is best. The approved units of this type are classed as NIOSH/MSHA TC-19C air-line units.

Many workers use negative pressure respirators (dust masks) that filter air with activated carbon, but there are important restrictions to these masks:

—Use *only* the approved NIOSH/MSHA TC-23 vapor and particulate masks that the manufacturer states are designed for use around free isocyanates. 3M Company recommends their #8711 disposable mask if there is enough ventilation to keep vapor concentrations in the air to less than 0.2 parts per million. Be careful; many other manufacturers of TC-23 masks do not recommend their products for painting with isocyanates, so read the directions carefully.

—The mask must fit well so that air comes through the mask rather than around it. This is difficult for people with small faces and impossible for people with beards. Facial hair between the mask and skin absolutely prevents a good seal. If you have a beard, your choice is simple: either shave the beard, use a supplied air system, or don't work with Imron or other hazardous products.

—Keep spare masks at hand, and change masks at the first whiff of paint odor. If you can smell it, then the activated carbon filter is saturated and is no longer effective. With care and adequate ventilation, my 3M mask lasts for about 40 hours. I seal my mask in a clean glass jar between uses.

You can work safely with Imron® and similar products, but you can't do it without some conscientious investment in protecting your health. Building a fully rated spray booth is very expensive, but I have developed some low-cost alternatives for hobby use. For instance, the fire code can be met by building the booth out of fire-rated gypsum board rather than more expensive sheet metal. And a low-cost exhaust fan can be cobbled together out of an old 1/3 to 1/2 hp washing machine motor driving an aluminum fan blade. The blade needn't be aluminum, but it must be made out of a non-ferrous material so it won't spark.

For a copy of my sketches and specifications for a low-cost spray booth, send a check for five dollars to cover handling costs to the address below:

Harvey Sachs
29 South Main St.
Cranbury, NJ 08512

Editor's Note: Harvey Sachs is a senior consultant to the National Indoor Environmental Institute. He consults on a wide range of indoor pollution topics.

DESIGN CRITERIA

An Analysis of Front Fork Flexibility

Raymond Pipkin

Editor's Note: As a former framebuilder, I am happy to say that almost all of the conclusions reached by Mr. Pipkin are confirmed not only by my own experience, but also that of other builders. However, the changes in fork flexibility shown in Figure 6 should actually correspond to differences in comfort that riders would easily notice. Experience always seems to underscore a difference in the comfort level of racing and touring frames. Readers' comments would be welcome.

Lightness buffs, take note! A careful reading will once again delineate the fundamental trade-off between weight and flexibility—at least for steel frames. A choice between the two should be made on the basis of your own needs, not those of fashion.

Jim Redcay

Like most cyclists, I have been curious about the relative influence upon fork flexibility of such factors as head angle, fork rake, fork radius, and tube shape and thickness. Racing cycles tend to ride harshly, but is their steep head tube angle and minimal fork rake the reason for this harshness? On the other hand, touring bicycles use more fork rake and a shallower head angle to reduce road shock — but by how much?

I have also been curious about the magnitude of forces in a fork. For instance, how are the stresses of vertical road loads distributed in a fork? What is the maximum stress in a fork under normal riding conditions and how much of the total yield strength does it represent? What stresses are involved under braking?

Decreasing Head Angle

This article uses engineering analysis to answer these questions. And the results are satisfyingly complete. For example, we can determine the percentage increase in fork flexibility if the head angle is decreased from 75 to 71 degrees, while the fork rake is simultaneously increased so as to maintain constant trail.

Moreover, the analysis enables me to give a rough estimate of the flexibility of the fork tip, expressed in units of, say, millimeters of fork tip deflection per kilogram of force applied at the dropout.[1] Deflection and stress at various points along the fork blade can also be calculated, giving additional insight into the magnitude of bending stresses withstood by the fork, and on the surprisingly minimal effect of the popular fork stiffeners so often found underneath the crown.

Tube Geometry

There are too many fork tubes in the world for me to run calculations on them all, so I selected four popular ones which have dissimilar cross-sectional shapes and thicknesses. I referenced DeLong's *Guide to Bicycles and Bicycling*, for the gauges and external dimensions listed below[2]:

—Reynolds 531, standard section oval: 18/21 gauge (1.2/0.8 mm), 29 × 16 mm;

—Reynolds 531, round: 17/20 gauge (1.4/0.9 mm), 22 mm round;

—Reynolds 531 SL, wide-section oval: 19/24 gauge (1.0/0.5 mm), 28 × 19 mm;

—Columbus SL: 20 gauge (0.9 mm), 28 × 19 mm oval.

Figure 1 defines the geometry of these fork tubes, and Table 1 lists the actual tube dimensions used in this analysis.

All these fork tubes taper to 12 mm O.D. round at the tip. The tubing gauges listed here refer to the thickness of the cylindrical fork tubes prior to the tapering operation. Rolling the tubes into a conical shape increases their thickness at the tip — a factor which I took into account.

The overall geometry of the front fork is specified in Figure 2.

Fork Design

My methodology, which will be of interest to mathematically inclined readers, is in-

[1]*Editor's note: Vertical flexibility is defined as the vertical distance (mm) which the fork tip moves per unit vertical force (Kg) applied to the dropout. Another term for flexibility is "deflection rate." Roughly speaking, "rigidity" and "stiffness" mean the opposite of "flexibility," within the context of this analysis.*

[2]Bike Tech *checked the Reynolds and Columbus catalogues and discovered that the nominal dimensions of Reynolds 531 and 531 SL oval tubing are slightly different than shown in Table 1. Reynolds records the oval dimensions as 28.5 × 16.5 mm for the 531, and 27.5 × 20 mm for the 531 SL. While these different external dimensions will slightly alter the fork tubes' moments of inertia and all subsequent calculations, they do not significantly change the spirit or conclusions of this analysis.*

Figure 1: Fork Tube Geometry. Linear taper and elliptical cross section is assumed.

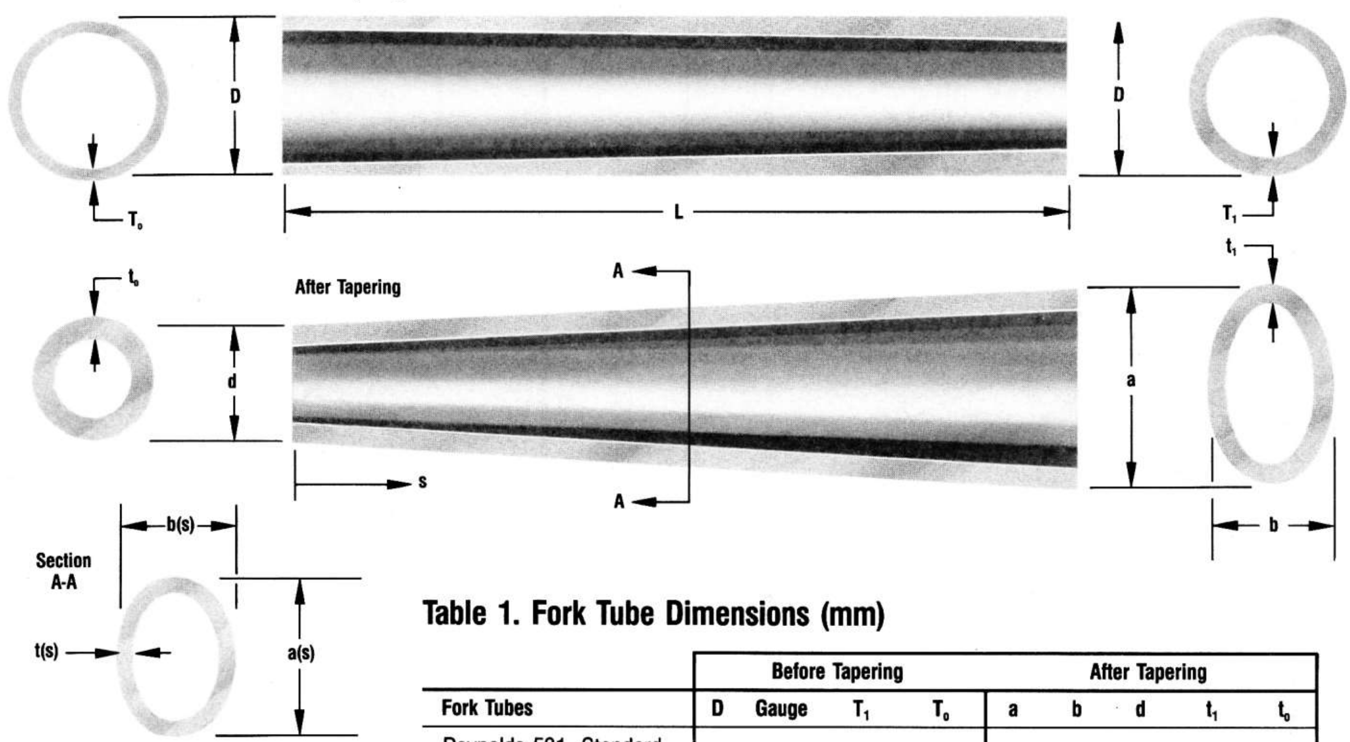


Table 1. Fork Tube Dimensions (mm)

	Before Tapering				After Tapering				
Fork Tubes	**D**	**Gauge**	**T_1**	**T_0**	**a**	**b**	**d**	**t_1**	**t_0**
Reynolds 531, Standard Section Oval	22	18/21	1.22	0.81	29	16	12	1.22	1.5
Reynolds 531 SL, Wide Section Oval	24	19/24	1.02	0.56	28	19	12	1.02	1.1
Reynolds 531, Round	22	17/20	1.42	0.91	22	22	12	1.42	1.7
Columbus SL	24	20	0.91	0.91	28	19	12	0.91	1.8

cluded in the accompanying sidebar. The results produced by this methodology are given in Figures 3–10. One observation worth noting is that regardless of the type of tubing used, the magnitude of the vertical fork deflection is pretty small. An average value for a typical frame design is 0.065 mm per kg, which translates approximately to 0.7 mm of deflection for a pair of fork blades loaded vertically with a force of 50 pounds.

Figures 3–6 show how changes in the head angle, fork rake, and fork radius will affect the fork's vertical flexibility. In Figure 3, only the head angle is varied; in Figure 4, only the rake; in Figure 5, only the radius of curvature used to produce the rake; in Figure 6, the trail is held constant while rake and head angle are varied.

Figures 3 and 4 are of mostly academic interest, since they represent conditions that most framebuilders would try to avoid (wide variations in trail). Still, they isolate the influence of fork rake and head angle on vertical flexibility.

Figure 2: Front Fork Geometry.

F = fork rake
D = wheel diameter
H = head angle
M = dropout-to-crown distance, arbitrarily set to 360 mm for this analysis
r = fork radius of curvature
L = trail

This analysis divides the fork into four distinct segments:
— L0: Fork dropout, assumed perfectly rigid. Length L0 set to 2 cm for all cases.
— L1: Lower straight segment. Length L1 arbitrarily set to 5, 4, and 3 cm for fork radii of 12, 16, and 20 cm respectively.
— L2: Central curved segment. Length $L2 = r \cdot \Theta$
— L3: Upper straight segment. Length L3 chosen to obtain desired dropout-to-crown distance M.

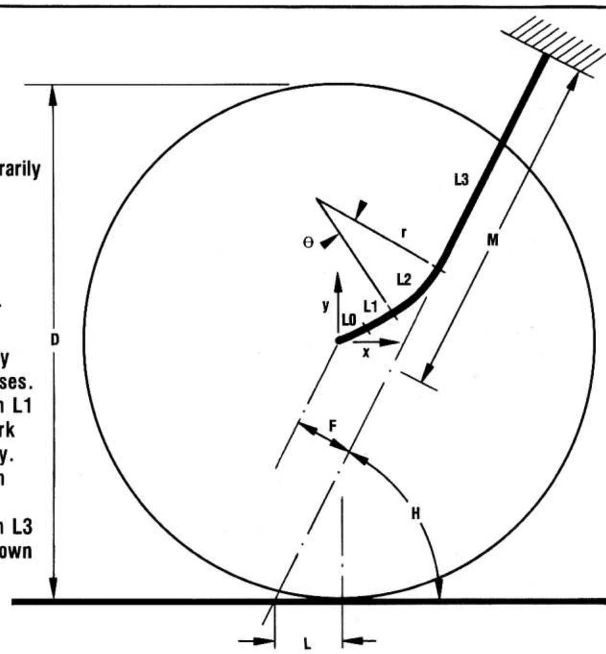


Figure 3. Vertical Flexibility vs Head Angle

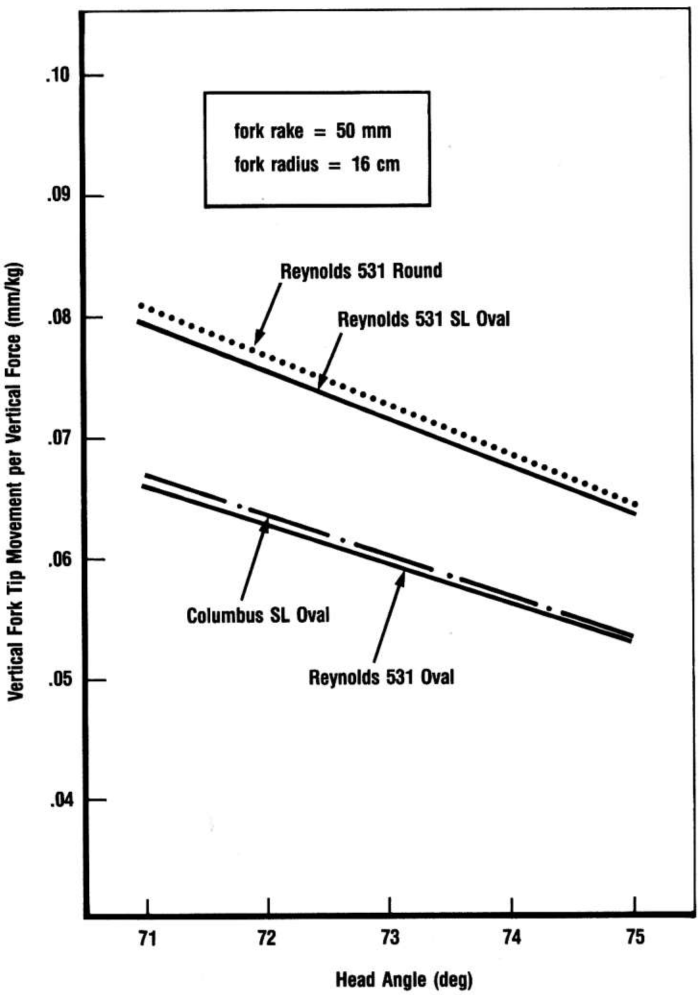
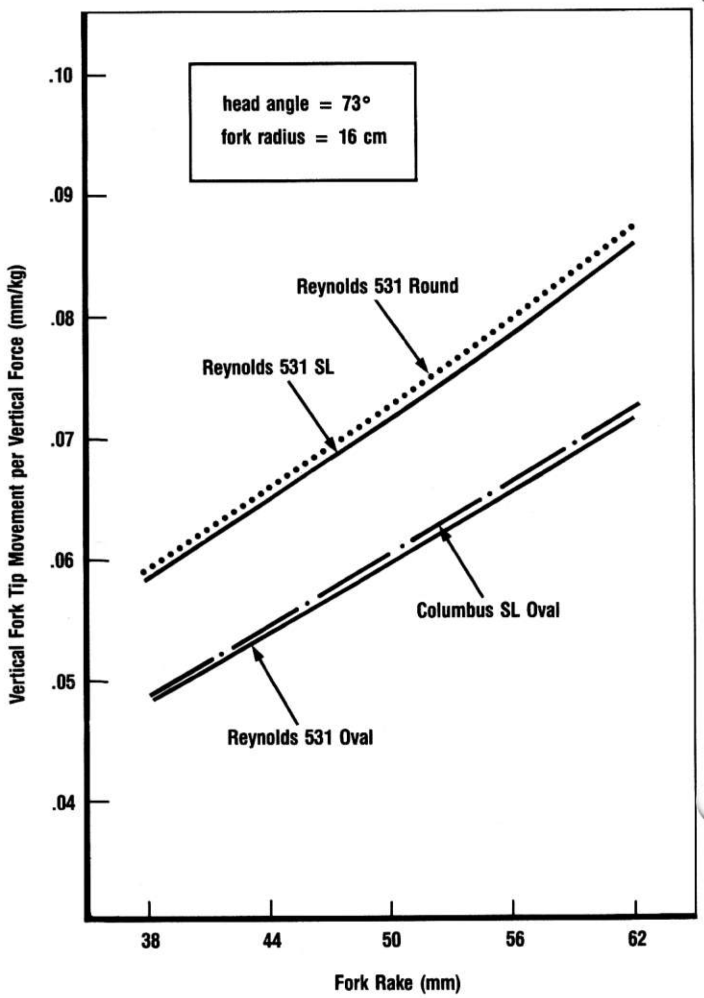


Figure 4. Vertical Flexibility vs Fork Rake



Shock Absorption

Figure 5, however, explodes an old myth. Framebuilders have traditionally raked touring forks with small radii of curvature for more shock absorption, and racing forks with greater radii for less shock absorption. My calculations show that varying this radius affects fork flexibility so little that it's hardly worth the trouble. The tiny differences visible in the graph would not be discernible on the road, particularly with pneumatic tires!

Figure 6 gets to the heart of fork design; it shows that variations in steering geometry that fall within conventional limits produce approximately a two-to-one change in fork flexibility. The fork in a frame with a 71-degree head angle will deflect a bit less than twice as much as a fork in an otherwise-identical frame with a 75-degree head angle (holding trail constant).

Braking Load

Figure 7 gives some indication of how much a fork flexes under a braking load. If we assume that the braking deceleration is about 0.5 G (which is about as quickly as you can stop without inviting pitchover), and that the entire retarding force is applied through the front wheel, then a 150 lb. rider/bicycle generates a rearward force of 75 lb. (34 kg) shared evenly by both fork tubes. Using an average deflection rate of 0.21 mm/kg from Figure 7, the total deflection of the two fork blades under this extreme braking condition is between 3 and 4 mm.

Vertical Displacement

Figure 8 displays the contribution of each incremental element of the fork tube to the total vertical displacement of the fork tip (for the mathematically inclined, it is a graph of $x^2(s) / E \cdot I(s)$ versus s). Using the Reynolds 531 oval tube as an example, we see that a one millimeter segment of the tube located at a distance of 15 cm from the fork tip as it bends under a vertical load of one kilogram, contributes 0.00195 mm of vertical fork tip movement.

An interesting application of the information in Figure 8 is gauging the effect of reinforcing the fork at the crown. Most high-quality frames have stiffening tangs at the fork tube/crown junction, but Figure 8

Figure 5. Vertical Flexibility vs Fork Radius

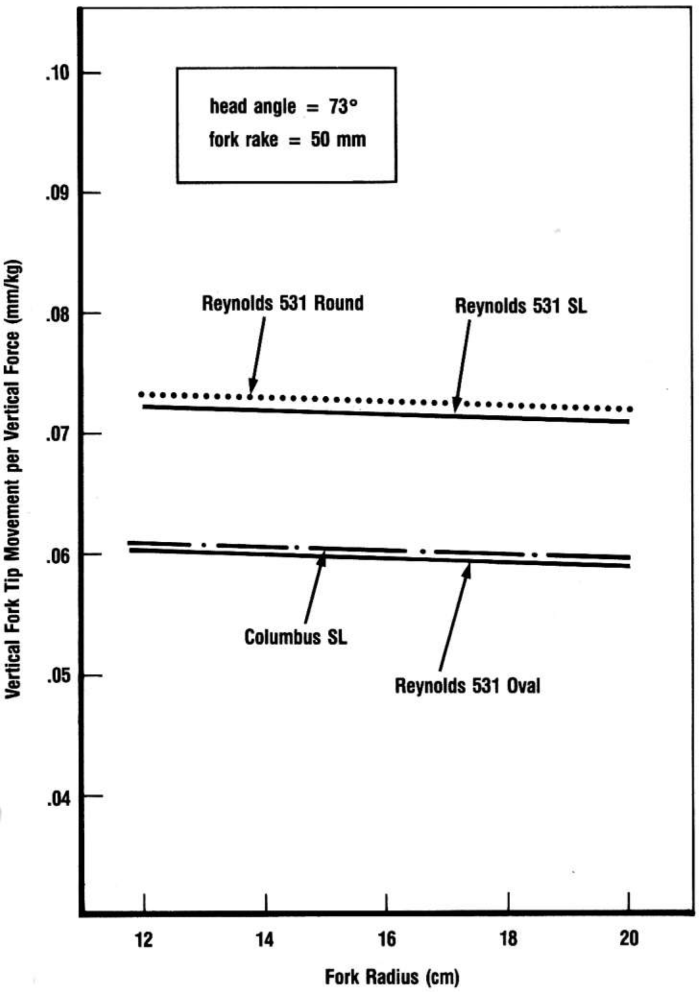
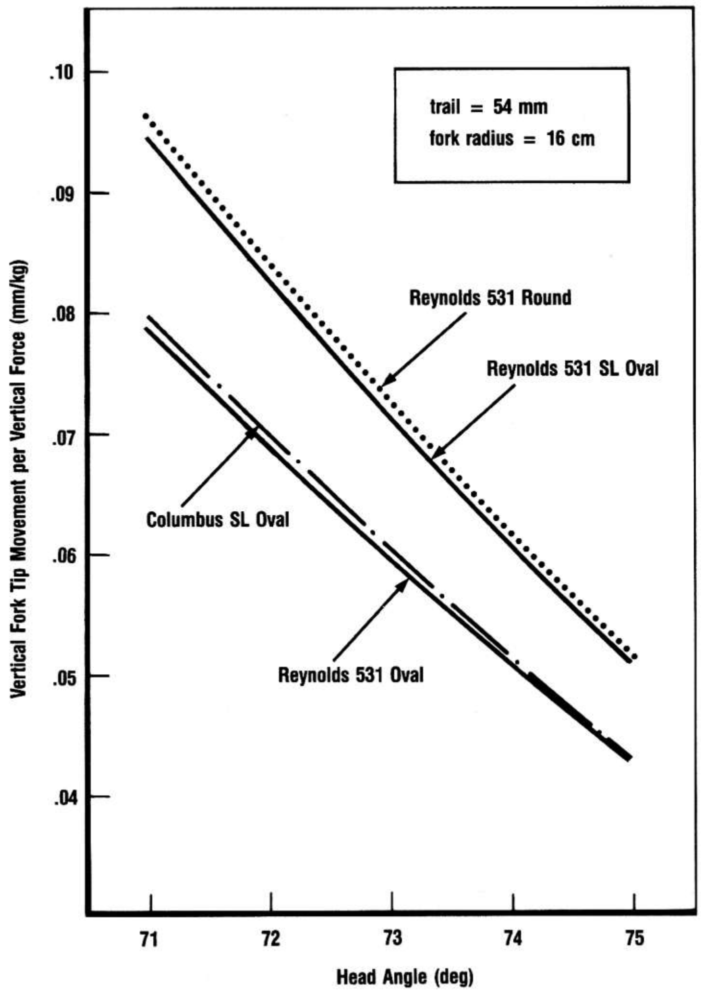


Figure 6. Vertical Flexibility, Constant Trail



shows that these tangs have only minimal effect on stiffening the fork in the vertical plane. For example, if the last five cm of the fork tube were made perfectly rigid, the vertical deflection rate would be reduced by an amount equal to the area under the curves lying between $s = 30$ and $s = 35$ cm (the approximate length of the tube). In the case of the Reynolds 531 oval and the Columbus SL tubes, the vertical flexibility would decrease by about 16 and 19 percent, respectively. But remember that this assumes a *perfectly rigid* sleeve wrapped around the top five centimeters of the fork tube. A slim, tapered tang down one side of the tube can increase rigidity in the vertical plane only moderately. In defense of their value, I surmise that the tangs may help distribute the stresses at the fork tube/crown junction, and may improve lateral rigidity.

Bending Stresses

Figures 9 and 10 give an indication of the magnitude of stresses in a fork under actual riding conditions. These two graphs display the maximum bending stresses at each location of the fork tube when subjected to unit vertical and horizontal forces acting at the dropout (again, for the mathematically inclined, Figures 9 and 10 represent the functions $^1/_2ax/I$ and $^1/_2ay/I$ respectively, where *a* denotes the tube's major diameter at location *s* along the tube's length). As an example, Figure 9 indicates that at a distance of 15 cm from the fork tip, the Reynolds 531 SL tube experiences a maximum bending stress of 0.48 kg/mm^2 when the applied vertical force is one kilogram. Or, looking at the right hand scale, we see that a one pound vertical force induces a stress of 310 lb/in^2 at the same point. So a 50 lb. vertical load on the front wheel causes a compressive stress of almost 8000 lb/in^2 15 cm from the tip.

Combined Stresses

Similarly, Figure 10 shows that a one-pound horizontal force at the fork tip produces a maximum bending stress of 450 lb/in^2 at the point 15 cm from the tip. So, for the 0.5 G deceleration rate that we considered earlier, the 75 lb. rearward force at the front wheel creates a tensile stress of about 17,000 psi on the front of the fork blade, at the point 15 cm from the tip.

What is the net effect if these two stresses act simultaneously? It is important to realize

Figure 7. Total Flexibility, Horizontal Force

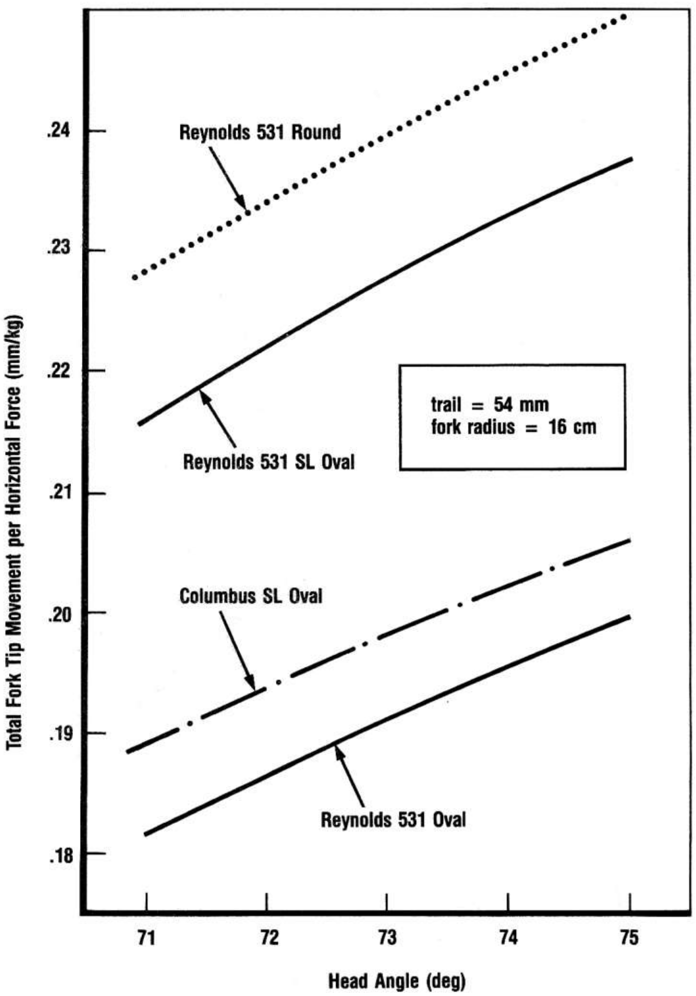
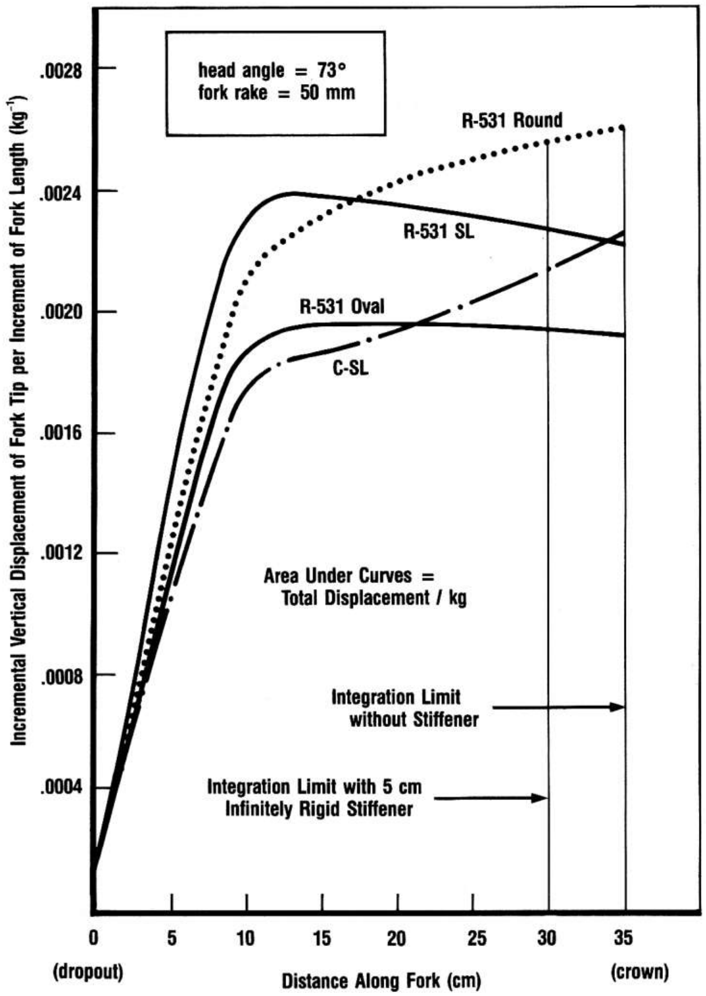


Figure 8. Incremental Displacement, Vertical Force



that these two stresses are of opposite mathematical sign (tensile stresses are considered positive; compressive stresses negative), so the resultant stress would be the difference between 17,000 and 8,000 psi, or only 9,000 psi. With a typical yield strength of about 100,000 psi, a fork blade is only operating at about nine percent of yield. That's a comfortable margin. It is curious to note that, for those forks made with "taper gauge" tubing, the maximum stresses generated by vertical loading do not occur at the tube/crown junction, but peak at a point 10 cm from the tip.

More to Consider

Unfortunately, Figures 9 and 10 do not tell the whole story about the amount of stress in a fork blade. For instance, they do not take into account the stress concentrations at the crown due to the sudden change in cross section at the tube/crown junction. Nor do they consider the residual locked-in stresses from brazing. These additional stress concentrations would be particularly important to gauge when investigating fatigue failure.

I believe that the foregoing analysis fairly accurately describes the relative influence of head angle, fork rake, and fork radius upon fork flexibility, short of performing actual load/deflection tests. I do not know the amount of variation in Young's modulus among the brands of tubing (I doubt that it varies by more than ten percent), but if this information becomes available from the manufacturers, the curves could be shifted vertically to incorporate the new values.

Mathematical Model of a Fork Tube

I assumed that the cross section of the tubes could be modeled as two concentric ellipses whose dimensions vary linearly along the length of the fork (see Figure 1). The resulting slight variation in thickness around the perimeter of the tube was considered negligible. These two assumptions allowed me to compute the area moment of inertia $I(s)$ about the minor axis of the tube's cross section as a function of the distance s along the tube.

This analysis assumes that the neutral axis

Figure 9. Maximum Stress, Vertical Force

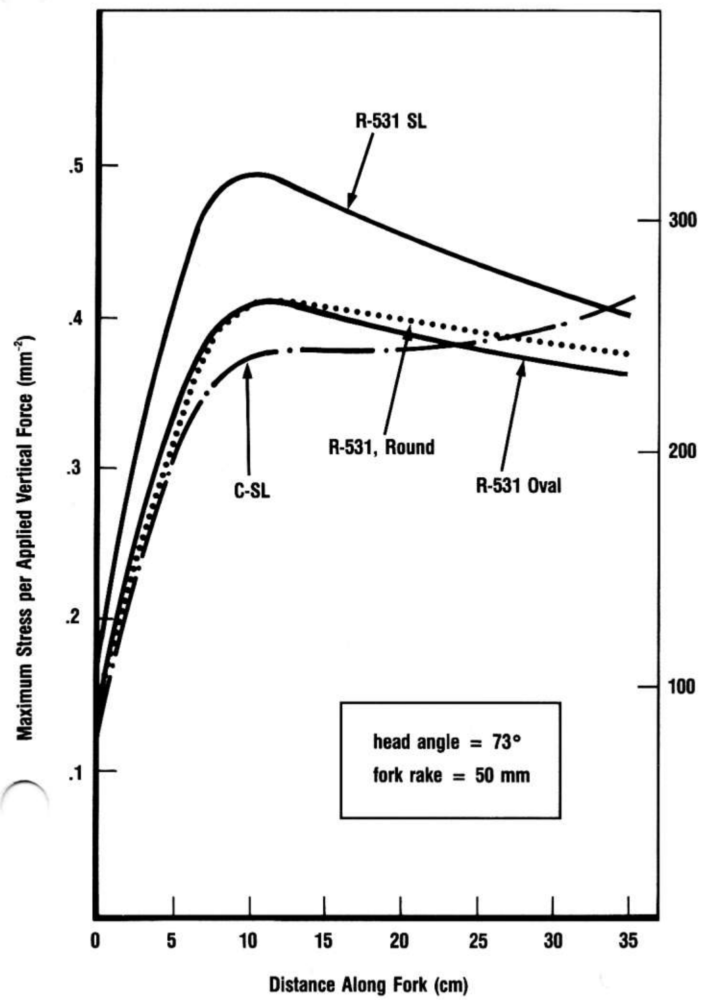
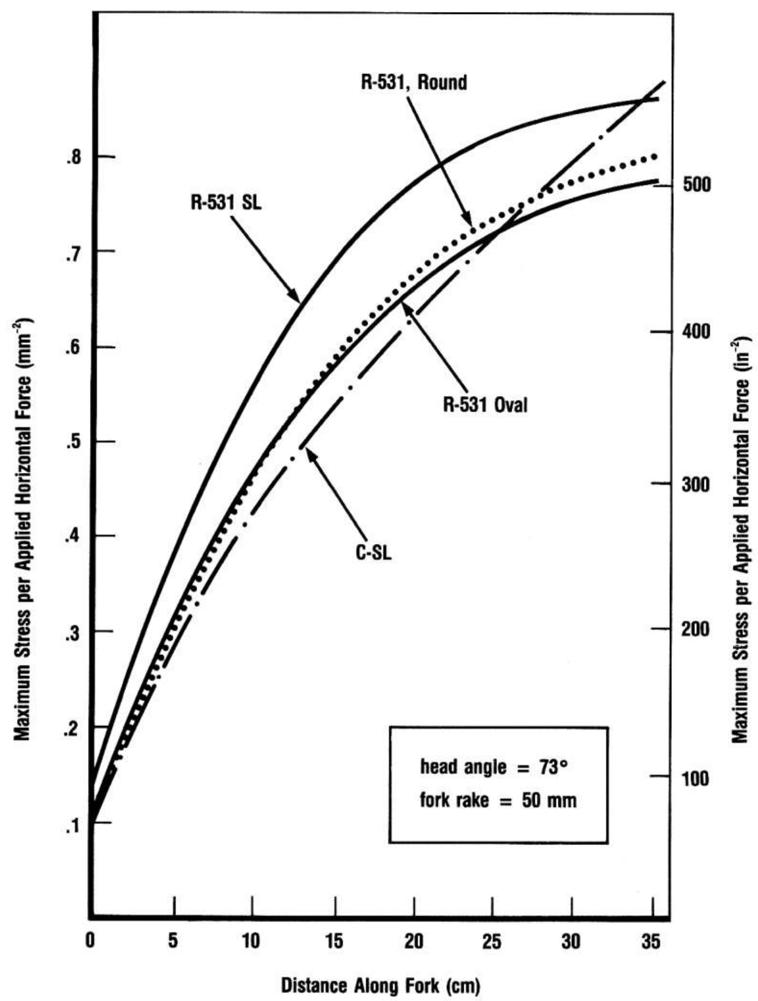


Figure 10. Maximum Stress, Horizontal Force



coincides with the center line of the tube's cross section. This is not strictly true for curved beams, as the neutral axis is shifted toward the center of curvature and the maximum stress on the inner edge of a curved symmetric beam becomes greater than the maximum stress on the outside of the curve. However, in this case it can be shown that the displacement of the neutral axis from the area center line is less than one millimeter and that the excess stress is less than three percent.

I next developed a set of parametric equations[1] expressing the positional coordinates x(*s*) and y(*s*) of any location s along the fork with respect to an x-y coordinate system whose origin coincides with the fork dropout (see Figure 2). The y-axis is directed vertically; the x-axis is directed horizontally rearward. The functions x(*s*) and y(*s*) depend upon the values chosen for head angle, fork rake, and fork radius.

Finally, I assumed that the fork could be modeled as a cantilever beam rigidly supported at the fork crown. To calculate the graphs for Figures 3-6, I substituted the expressions for I(*s*) and x(*s*) into the cantilever beam formula to obtain an expression for the vertical flexibility:

$$D_{yy}/F_y = \int [x^2(s)/E \cdot I(s)]\, ds$$

where D_{yy} denotes the vertical deflection of the fork tip due to a vertical force F_y acting on the dropout. (A concomitant forward deflection was not computed.) The integrals were evaluated numerically over the length of the fork using the trapezoidal rule. Young's modulus of elasticity E was assumed to equal 21,000 kg/mm^2 (about 30×10^6 lb/in^2) for all tubes.

The total fork tip deflection in response to a horizontal force was calculated for the graph of Figure 7 in the main article. The total deflection consists of a horizontal rearward component,

$$D_{xx} = F_x \cdot \int [y^2(s)/E \cdot I(s)]\, ds$$

and a vertical downward component,

$$D_{yx} = F_x \int [x(s) \cdot y(s)/E \cdot I(s)]\, ds$$

where F_x denotes the horizontal force. The total deflection equals the square root of the sum of the squares of these magnitudes.

Ray Pipkin

[1] *For those readers interested in the equations behind this analysis, send an SASE to Front Fork Analysis c/o Bike Tech, 33 E. Minor St., Emmaus, PA 18049.*

A Musical Test for Correct Spoke Tension

John S. Allen

As described in Jobst Brandt's book *The Bicycle Wheel,* the optimum spoke tension must be high enough so that the cyclical stresses on the spoke during riding do not slacken the spoke completely, but not so high that the spoke or rim is stressed beyond its yield point. Spokes tensioned between these two extremes have the longest fatigue life and provide the greatest resistance to rim damage from accidental impacts.

Long-time experience is usually necessary to build durable wheels consistently because it's not easy to tell by eye just when spokes are tensioned properly. In practice, most wheelbuilders adjust spoke tension at first by guess, and with more experience, by feel. Pulling on the spokes, however, is not an accurate tension-measuring technique. The hands are not very sensitive measuring instruments, and the appropriate spoke tension varies depending on the gauge of spoke. Tools that measure spoke tension are available, but they are not in wide use, nor are they particularly easy or fast to use.

Measurement Without Gauges

However, spoke tension can be gauged accurately enough—to within a few percent—by its musical pitch when it is plucked. The only tool you need is a pitch pipe, available at any music store. Obviously, you can't true your wheels with a pitch pipe—the vagaries of rim and spoke dimensions prevent this—but you can easily use musical pitch to determine whether your spokes are within the correct range of tension. Surprisingly, the musical pitch corresponding to optimum spoke tension is the same regardless of the spoke gauge. This is so because the same musical pitch results from the same level of tension per cross-sectional area—*the same stress on the steel of the spoke*—whether the spoke is thick or thin. A typical spoke stress of 50,000 psi (3.5×10^9 dyn/cm^2)—approximately one-third the yield strength value of the steel used in bicycle spokes—results in a specific musical pitch, or frequency of vibration, in a plucked spoke that can be calculated by an equation found in any elementary text on the physics of waves.

The fundamental frequency of vibration for lateral motion of a plucked string or wire (like a guitar string or bicycle spoke) under tension is described by the formula:

$$\frac{1}{2\pi L}\sqrt{\frac{T}{k}}$$

where L is the length of the string, T is the tension, and k is the mass per unit of length. Notice that, for two different strings of equal length, one thick and another thin, the fundamental frequency of vibration is the same if the tension per unit of cross-sectional area is the same. If we use a thicker string, the mass k and the tension T increase in the same proportion to give a constant frequency of vibration.

A simple way to look at this is to consider two strings lying side by side, of equal thickness, and under equal tension. Both of these strings, when plucked, will vibrate identically at the same frequency. Now let's imagine the two strings merged together; the two strings will now vibrate as one, but there is no difference in the frequency of vibration because, while the mass of the new string has doubled, so has the tension. In the same manner, we can "merge" any number of strings together, and the frequency would always stay the same.

This vastly simplifies the measuring of spoke tension. In order to determine whether a spoke is optimally tensioned, we don't have to measure the thickness or, what is more difficult, the tension. Checking the musical pitch is enough; it translates directly to the tension per unit of cross-sectional area. If we want to determine the actual tension of the spoke, we need only multiply by the cross-sectional area. We can easily determine this using a micrometer caliper.

Calculated Frequency

Let's plug some numbers into our equation and derive a typical value for spoke pitch. For a steel spoke, of density 7.87 gm/cm^3, length 30 cm, under a tension of 3.5×10^9 dyn/cm^2, we arrive at a frequency of 354 Hz (cycles per second), musically an F above middle C. The free vibrating length of the spoke is somewhat shorter, as the spoke is essentially rigid where it is inside the nipple and where it overlaps the hub flange. The larger-diameter ends of a double-butted spoke also decrease the effective length somewhat, though not to their full extent. Also, the spoke's resistance to bending throughout its length raises its frequency a few percent. This bending stiffness can be modeled quite accurately as a shortening of effective length.

The outcome of these corrections for effective length is to make the pitch somewhat higher, as shown in the accompanying table. The corrections can be worked out theoretically using some very complicated formulas, but to avoid serious number crunching, I determined the corrections empirically by clamping some actual spokes at different points along their lengths and measuring the excess increase in pitch beyond what is calculated with the formula. For common spokes, the tension should be within a few percent of that given in Table 1.

Tension of a spoke that is not at any of the pitches in the table can also be calculated. For each musical octave (12 steps or semitones) of increasing pitch, the tension approximately quadruples; for each six semitones, it approximately doubles; for each semitone it increases about 12.25 percent. Typically, spokes in the left side of a rear wheel will have a musical pitch five semitones lower, corresponding to a tension hardly more than half that of the right-side spokes. The musical calculation of the tension ratio between the sides of a rear wheel agrees very well with a calculation based on the bracing angles.

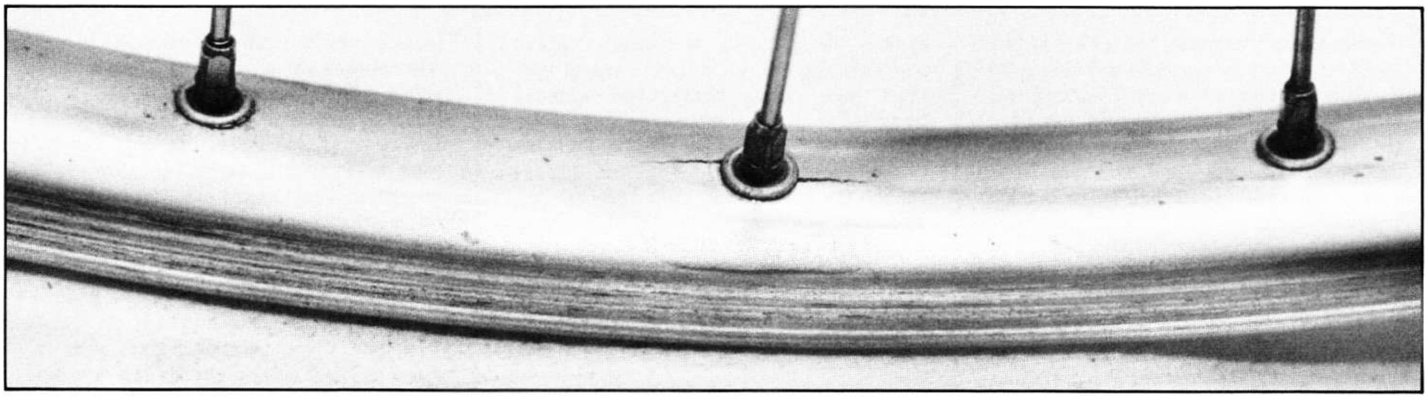
Checking Spoke Pitch

It is easiest to hear the musical pitch of spokes in a wheel with radial or unlaced spokes which vibrate independently of one another. In a wheel with laced spokes the two spokes of each laced pair will vibrate together. Pluck them where they cross; if the tension of both spokes is nearly equal—as it will be in a well-built wheel—they will vibrate as a unit. If the tension of one spoke is much different from that of the other, they may vibrate separately, and you may not hear a clear musical pitch. You can confirm this tension difference by pulling the two spokes toward each other with your hand to see whether one is slacker, or by lifting one clear, then plucking the other.

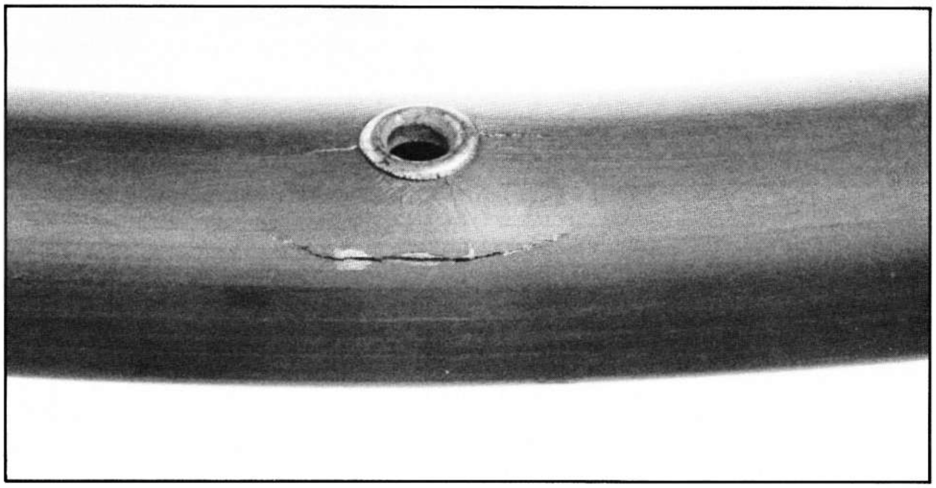
No spokes are perfectly uniform, and no rim approaches perfect roundness without some coaxing from the spokes. Therefore, when you check spoke tension, do ***not*** try to get the tension or the musical pitch of all of the spokes perfectly equal. Rather, you true the wheel as usual, to eliminate hop and wobble. You use musical pitch to check the general level of tension.

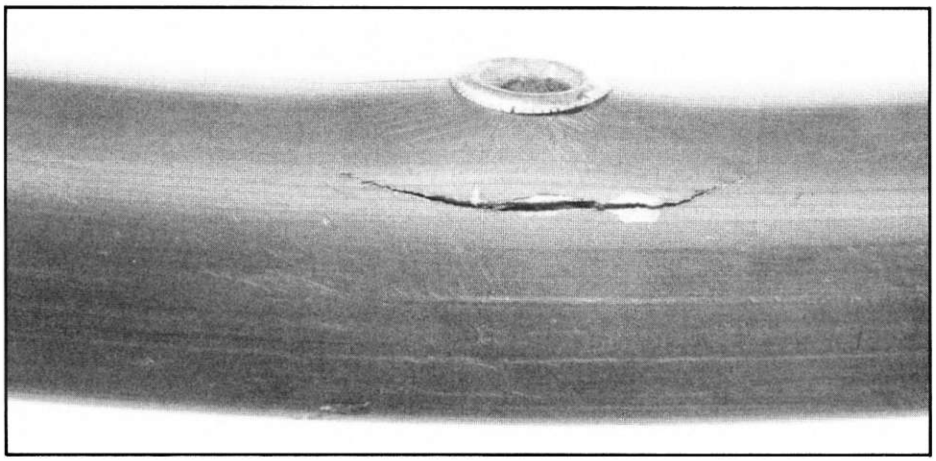
In a well-built wheel with a good rim, you will find that the musical pitch of the spokes will not vary by more than two or three semitones (musical steps), corresponding to a tension variation of 25-35 percent. If a wider range of pitch is necessary to true the wheel, then the rim is warped and should be bent back into shape or discarded. Avoid raising any spoke more than one step above the pitches in the table. If necessary, lower the overall pitch of the wheel a step or two, but recognize that this will weaken the wheel somewhat.

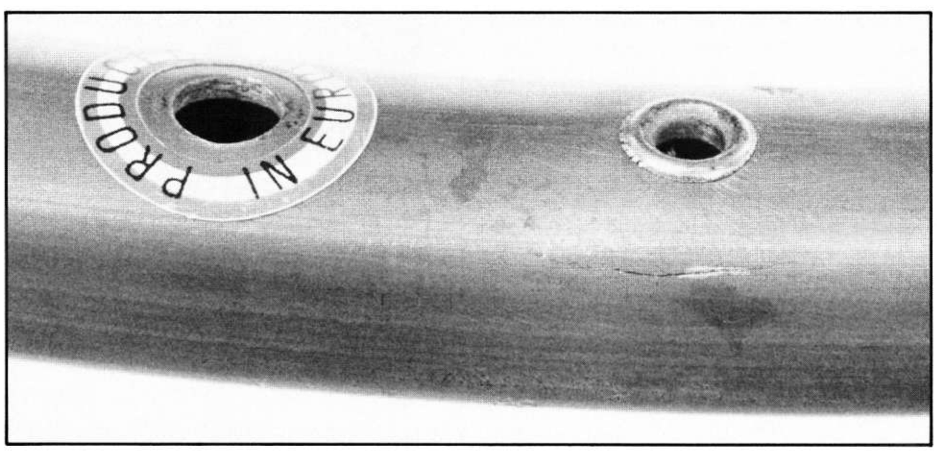
On a dished rear wheel, you only can bring the right-side spokes up to the tension indicated in the table. Do not raise the tension of the right-side spokes higher in an attempt to



Heavy spokes under high tension in a light rim lead to troubles like those shown here. (Rear wheel, DT 2 mm spokes, 13 mm aluminum rim, tensioned somewhat above optimal.) The rim pulled up around the spoke holes, and the wheel went out of true. Eventually, the rim cracked along a line between the spoke holes. The author is aware of two identical wheels that have failed this way. One belongs to a 215-pound rider, the other to a 110-pound rider, so the failure was not clearly attributable to excess loading of the wheel.







Spoke Length (MM)	Steps (Semitones) of Musical Pitch for 3.5×10^9 dyn/cm^2
326	G above middle C
308	G#
292	A
276	A#
262	B
248	C
236	C#
224	D
212	D#
201	E
191	F
181	F#
172	G
163	G#
156	A
147	A#

Table 1: Musical pitches of properly tensioned spokes. The pitches listed in this table are for double-butted spokes; use a pitch two semitones lower for plain gauge spokes. Don't be put off if you can't find your exact spoke length in this table. Notice that the gap in length between spokes of 292 and 308 mm (the range in which you'll find most spokes for 27 inch and 700C wheels built with large and small flange hubs laced cross-three) corresponds to only one musical semitone. Even if your ears can discern this pitch difference, you can't buy a pitch pipe that plays this "in-between" note. Be satisfied that if your spokes "play" between G# and A, they will be in the correct range of tension.

"compromise" between the sides of the wheel: the right-side spokes will simply be too tight, and they will either break or cause damage to the rim. Unless special measures are taken—like lighter or fewer spokes on the left side—the left-side spokes must be at a lower pitch because of their larger bracing angle.

Choosing Spoke Gauge

One major issue about spoke tension is left unanswered by the musical test: whether the spokes are the correct gauge for the wheel.

The strongest wheel results when the spokes are as heavy as possible without overstressing the rim in normal use at optimum spoke tension. Heavier spokes cannot be raised to optimum tension without risking damage to the rim when lateral loads increase spoke tension during use. Lighter spokes can be raised to their optimum tension, but some of the rim's strength is wasted on them. There is no serious problem with this—most wheels are built this way. Generally, any medium- or heavy-weight rim can handle straight 14-gauge (2 mm) spokes, the heaviest ones in common use.

With light rims or heavier spokes, experimentation is necessary. The experiment is, as Jobst Brandt suggests, to raise the spoke tension until the rim will not hold its true when spokes are pulled toward each other to stress-relieve them. The usual way a rim fails is to pucker out around the spoke holes. Unfortunately, once this has happened you have, in my opinion, sacrificed the rim, since you have overstressed and work hardened the aluminum around the spoke holes. Perform the experiment on a rear wheel, in which the uneven spoke tension places greater demands on the rim.

If the wheel warps when you stress it after raising it to the musical pitch given in my table, you must use lighter spokes with this model of rim. Failure to heed this warning can lead to the type of failure shown in the photographs.

IDEAS & OPINIONS

Practical Vehicle Considerations

Glen Brown's piece on HPV aerodynamics in the February issue of *Bike Tech* was very good. In the future, I would like to see someone expand on it and address the issue of vehicle stability as it relates to the shape of the vehicle.

I gather from articles I have read about car design that for stability in cross winds, the center of wind pressure (the point at which all the air pressures acting on the body are considered concentrated) should always be behind the car's center of gravity. This arrangement may not be too critical for HPVs intended only for 200 meter time trials, but it is an important consideration for machines intended for road racing or for machines that are built with the intention of being practical commuting HPVs.

John Riley
Iowa City, Iowa

Touring Bike Insights

In response to "New Design Needed" in the February 1984 issue of *Bike Tech,* I have three comments:

—Many manufacturers make bicycles designed for packed touring. Take for example, the Specialized Expedition. In general, the frames of these bicycles are made with heavy gauge tubing, are shod with 27 × 1 3/8 or 700 × 38 tires, and have their forks drilled for mounting "low riders."

—The design that Patrick Warfield writes of is not experimental, but is similar to a standard delivery bicycle, available, for example, from Worksman Cycles. These bicycles typically have a 26-inch rear wheel and a 20-inch front wheel, with a box mounted to the frame over the front wheel. These bikes are not lightweight, but neither are they designed for touring. The point is that this design is well tested, and the bikes are stable when loaded.

—Even a very lightweight frame can pack 40 to 50 pounds simply by not attaching the load to the frame. Instead, use a bike trailer, like the Cannondale Bugger. These trailers do present some problems with stability during braking, but they barely affect steering and don't increase frame whip.

Eric Schweitzer, Head Mechanic
Larry and Jeff's Bicycles Plus
New York, New York

Let Us Hear

We'd like *Bike Tech* to serve as an information exchange — a specific place where bicycle investigators can follow each other's discoveries. We think an active network served by a focused newsletter can stimulate the field of bicycle science considerably.

To serve this function we need to hear from people who've discovered things. We know some of you already; in fact some of you wrote articles in this issue. But there's always room for more — if you have done research, or plan to do some, that you want to share with the bicycle technical community, please get in touch.