Materials • Mechanics • Physiology • Engineering • Aerodynamics

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MATERIALS

The Metallurgy of Brazing, Part 1

Mario Emiliani

Have you ever wondered what you're doing when you braze two metals together? What makes them stick? Why is a well-brazed joint so strong when the filler metal is so weak? What factors affect the strength of the joint, and how sensitive are they?

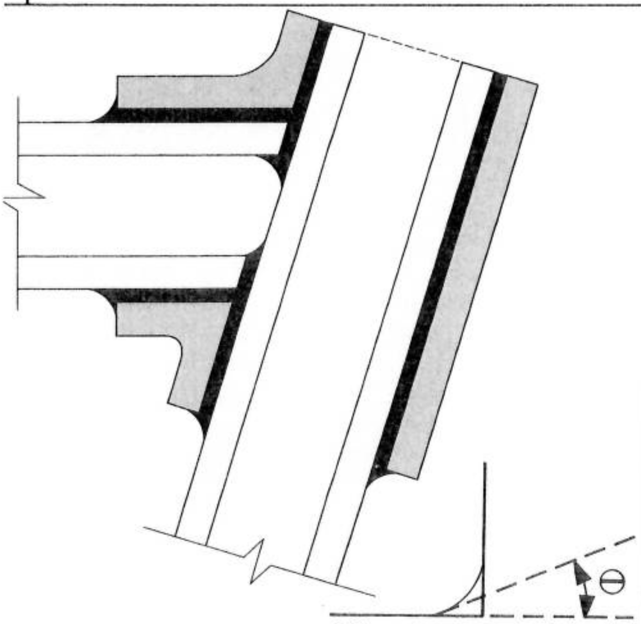
Brazing isn't a difficult skill to learn, and it's not even really necessary to know much about it to produce a well-brazed frame. But knowing a few details of the process can only help you produce more consistently sound joints, and satisfy your curiosities. "The Metallurgy of Brazing" is a series intended to thoroughly explain brazing. The above questions are but a few that will be answered in this series.

History of Brazing

It's difficult to say when brazing was invented, let alone who invented it. Brazing, like most other manual arts, was something handed down from generation to generation. Apparently nobody thought enough of brazing to document it, or perhaps documentation would have led to a loss of one of the world's first trade secrets. In any event, its origins are a mystery.

The earliest examples of brazing are found in jewelry and other types of adornment from about 2,500 years ago. Pieces of pure gold were joined together using lower-melting alloys of gold and silver.

About 900 years ago, when it was discovered that zinc was a separate metal, brazing with brass filler metals became popular. It was through the use of these filler metals that the term "brazing" came about. Originally, the process of joining metals using lower-melting brass filler metals was called "brassing." Through the centuries, this term evolved to the word "brazing."



Effect of wetting on brazing alloy penetration.
A: good wetting; low contact angle; full penetration.

Definitions

Before we continue, we must go over a few definitions. Brazing is a process which joins metals by heating them to a suitable temperature, then introducing a non-ferrous filler metal. The filler metal must have a liquidus* *above* 840°F (450°C), but below the solidus of the base metals. The filler metal is distributed through the joint by capillary attraction.

Soldering is the same as brazing, except the non-ferrous filler metal has a liquidus *below* 840°F (450°C). Because a few silver brazing alloys melt at very low temperatures, and because silver brazing was originally termed hard soldering, silver brazing is usually referred to as silver soldering. The difference in

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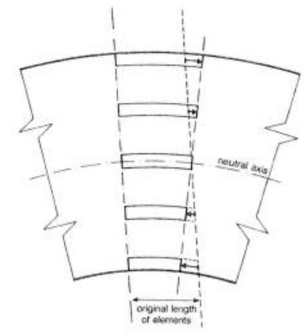

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liquidus between a brazing alloy and a soldering alloy has so crucial an effect on the nature of the joint that the term silver soldering should never be used when referring to silver brazing.

(There are several factors that contribute to this difference; most of them are too complex for a brief description. One major factor, though, is simply that soldering alloys are generally much weaker than brazing alloys, and this difference is reflected in the strength of the joint.)

Welding differs from brazing and soldering in that the base metals are melted, and capillary attraction isn't a factor. Welding may or may not use a filler metal, depending on the process.

Braze welding, as the name implies, is akin to both brazing and welding. The similarities are (to brazing) that the base metals aren't melted, and (to welding) that the filler metal (which may be ferrous or non-ferrous) isn't distributed by capillary attraction. An example of braze welding is two pieces of steel which are joined by simply building up a brass fillet betwen them as in a lugless frame joint. Thus, the strength of the joint depends upon the strength of the fillet.

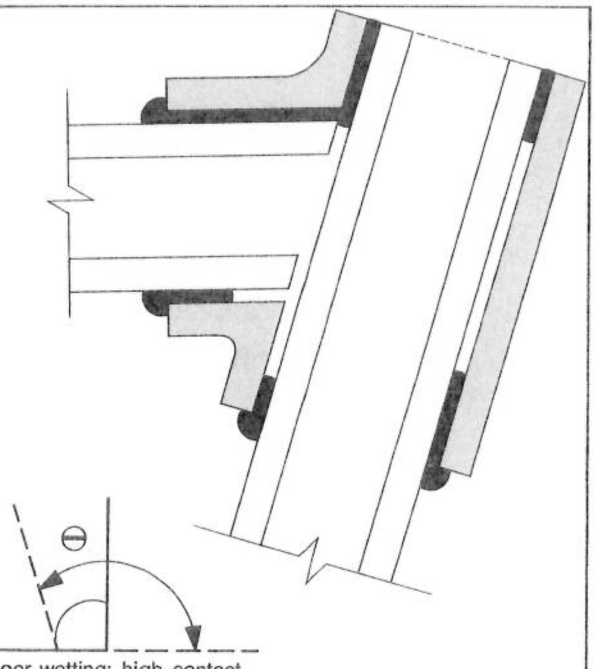
I'm sure you are all well acquainted with the definition of brazing, but I bothered to define soldering, welding, and braze welding, with hopes of giving you a better understanding of what brazing is by comparing it with other joining processes. The differences should help clarify the important factors in brazing.

*See *Bike Tech, June 1982 for definitions of liquidus, solidus, etc.*

Adhesion Theory

What makes one substance stick to another? Nobody really knows. It's not even known why ordinary Scotch® Tape sticks to paper, plastics, glass, or anything else it sticks to. The whole science of adhesives is very complex. Chemists and chemical engineers spend a great deal of time experimenting with thousands of substances, to see if any of them have uses as adhesives. It's very time-consuming trial-and-error work, but the payoff can be enormous — just look at all the adhesives in the hardware store.

Since nobody knows how adhesives work, there are a number of theories around to explain it. The trick in making things stick together is to develop very intimate contact between mating surfaces. The sticky stuff on Scotch® Tape is a very viscous liquid which bonds readily to a cellophane backing. When the tape is pressed onto a favorable surface, the air is squeezed out and the viscous liquid fills up all the microscopic gaps on the surface. This creates such intimate contact that atomic forces between the viscous liquid and the surface (and between the liquid and the cellophane) can form a "mechanical bond." This bond accounts for most of the tape's holding ability.



B: poor wetting; high contact angle; incomplete penetration.

The strength of the bond depends largely upon the degree to which the viscous liquid displaces air, and fills up the gaps on the surface. The more gaps that are filled, the stronger the bond is going to be. To illustrate the difference the amount of contact makes, lightly attach one end of a piece of tape to a relatively smooth, flat surface, and pull the tape parallel to the surface. It should take only a light tug to shear the tape from the surface. Now attach another piece of tape to the surface, rub the contact area with your fingernail, and pull. You'll agree, it takes a much larger force to shear the tape off. In fact, if you do the experiment on a rigid surface (like a desk top), the tape will fail, not the joint. This is an example of mechanical bonding.

In mechanical bonding, secondary atomic forces called Van der Waals bonds are what give a joint its strength. These bonds are due to the electrostatic attraction between the nuclei of one molecule and the electrons of another. The forces generated by Van der Waals bonds vary according to the distance between molecules and the type of molecules. So depending on these factors, anything from extremely weak joints to relatively strong joints can be made.

Another type of bonding is chemical bonding. In this case, much stronger primary atomic forces form the bonds, enabling joints of very high strength to be made. Brazing, soldering, and welding are joining operations which form primary bonds of a type called metallic bonds.

Metallic bonds, as the name implies, are characteristic of metals. These are the bonds which hold metals together and give them their unique properties; i.e., high electrical and thermal conductivity, ductility, shiny appearance when polished, etc. These bonds form when atoms with easily detachable electrons come so close together that their electrons can circulate freely between the atoms. Thus the negatively charged "sea" of electrons in a metal crystal holds the positively charged metal ions securely in place.

During brazing, metallic bonds are formed due to extremely intimate contact between the filler metal and base metals. In addition, there is always some degree of alloying between constituents of the base metals and filler metal. This action also forms metallic bonds. While it is these bonds that allow the filler metal to adhere strongly to the base metals, the strength of the joint depends upon several other factors as well. These factors will be discussed here and in subsequent articles.

There are several other theories of adhesion, but chemical adhesion is the one most likely to explain how liquid metals bond to solid metals. For this reason, I will not attempt to explain the other theories.

Wetting

A factor critical to whether or not an adhesive sticks is its ability to wet the material to which it's applied (called the "adherend"). Wetting is a substance's ability to spread, and consequently become intimate with a

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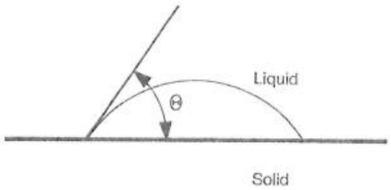

Figure 1: The angle θ is one way of measuring how well a liquid wets a solid surface. As the angle decreases, wetting increases.

surface. Wetting ability depends on the relative magnitude of two variables: the adhesion between the liquid and the surface (due to attraction between molecules of the liquid and molecules of the solid); and the cohesion of the liquid (due to its molecules' attraction for each other). In other words, wetting depends on whether a liquid sticks more tightly to itself or to something else.

A measure of how well a liquid wets a solid is the contact angle θ. Figure 1 shows a drop of liquid that has come to rest on a surface. The angle formed between the surface and a line tangent to the drop is high, which means the liquid isn't wetting the surface very well. The lower the contact angle, the better the wetting. For example, if Figure 1 shows how a water drop acts on the surface of a newly waxed car, then the wax is a barrier to wetting.

All metals are covered by oxide films, which form when a metal is exposed to an environment containing oxygen. You can clean oxides off, mechanically or chemically, but immediately a new oxide layer will start to form. The thickness and tenacity of the oxide layer depends upon the metal and the environmental conditions. Oxides are barriers to wetting because their atoms are bonded ionically. A characteristic of ionic bonding is that there aren't any free (or easily detachable) electrons, which are a prerequisite for forming metallic bonds. Thus, all oxides must be removed if wetting is to take place.

The easiest way to remove oxides is to chemically treat either the liquid or the surface. For brazing either mineral fluxes or gaseous atmospheres are used. Since most bicycle frames are brazed with mineral fluxes, I will forego discussion of protective gaseous atmospheres (although the principle is the same).

It was once believed that molten fluxes increased the wetting ability of brazing alloys (and therefore the bonding) by reducing the alloys' surface tension. All liquids have a surface tension, measured as the force per unit length on a surface, which opposes expansion of the surface area. Surface tension results from the cohesive forces between adjacent molecules of the liquid. Its origin can be visualized in the following way: Imagine a molecule in the middle of a stationary drop of water. This molecule's relation to its nearest neighbors is symmetrical in all directions, and therefore the forces acting on the molecule are the same on all sides. Now imagine a molecule at the free surface of the drop. This molecule isn't being pulled equally from all sides; there's a force pulling it inward that isn't opposed by any force pulling it outward. This means that every molecule on the surface is under a constant force tending to pull it inside the drop (Figure 2). As a result the surface exhibits a tension, and will contract at any opportunity.

But to affect the alloy's surface tension, the flux would have to change the cohesion, and to do this it must dissolve in the brazing alloy. Experiments were done to verify this, and it turned out that fluxes were virtually insoluble in molten brazing alloys. So how does flux enable the brazing alloy to flow better?

Flux is a chemical consisting mostly of fluorides, chlorides, and borates. When applied to a metal in paste form and heated, the water boils off leaving the flux crystals attached to the metal's surface. Upon further heating, the flux melts, wets the metal, and becomes chemically active. During this "active" period, the flux dissolves and absorbs contaminants (primarily oxides) on the metal's surface, and prevents further oxidation of the metal by coating it.

So now the surface of the base metal is essentially free of all oxides, and the brazing alloy has no difficulty in wetting it when introduced. After a while, the flux becomes saturated with oxides from the base metals, brazing rod, oxygen in the air, and torch flame (if an oxidizing flame is used), and is no longer effective. One should complete the brazing operation before this happens (or add more flux).

So the flux doesn't reduce the surface tension or cohesion of the molten brazing alloy; instead it enables good adhesion by cleaning the metal's surface.

Wetting agents can be added to improve wetting, but they must be soluble in the liquid. About 28 years ago, a program was undertaken to develop brazing alloys that didn't require flux, either mineral or gaseous. The brazing alloys were to be made self-fluxing and airproof by alloying them with small amounts of powerful deoxidizers. In addition, it was hoped that the deoxidizers would reduce the surface tension of the brazing alloy.

The results were that the brazing alloys

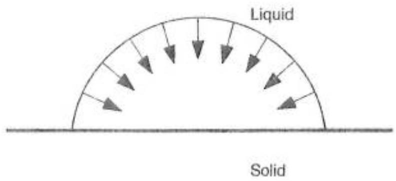

Figure 2: The arrows indicate the direction of surface tension forces. These forces tend to make the droplet contract, thus assuming a spherical shape.

containing the strongest deoxidizers (lithium, magnesium, aluminum, etc.) exhibited improved wetting on steel, but diffusion of atmospheric oxygen through the molten brazing alloy was fast enough to oxidize the base metal. Thus the molten brazing alloy wet the base metal well, but only if the brazing time was very short. Some elements, such as copper, actually increased the diffusion rate of oxygen.

It was also discovered that small additions of nickel or tin might make the molten brazing alloy impervious to diffusion of atmospheric oxygen. As it turns out, this endeavor found use, but only in specialty applications such as jet engine components. Bicycle builders still have to use fluxes.

In view of all this, the importance of using the proper mineral flux is obvious. A good mineral flux should have the following properties:

1. Melt at temperatures below the melting point of the brazing alloy.
2. Completely wet the base and filler metals.
3. Reduce, dissolve, and absorb oxides.
4. Protect the base metal from oxidation during heating.
5. Be displaced by the molten brazing alloy when liquid.
6. It must not run off the base metal, leaving areas open to oxidation.

All six points are important, but the fifth one has special significance. Molten brazing alloy pushes flux out of its way as it's sucked into the joint. If the flux is too viscous, flow of the molten brazing alloy will be impeded. Thus, there will be many areas where the brazing alloy didn't wet the base metals, and the joint strength will be reduced significantly. Fortunately, most, if not all, commercial fluxes meet the six requirements.

It's commonly believed that the criterion for a liquid metal to wet a solid metal free of oxides is that the two metals must be soluble and form an alloy at their interface. This isn't true; wetting will occur if the surface tensions are favorable. Specifically, the surface tension of the solid must be greater than the sum of the liquid and solid/liquid interface surface tensions.

While alloying between the filler and base metals occurs to some degree in most cases, it isn't essential to the forming of a good metallic bond. A good flux will clean the metals so well that metallic bonds are easily formed (provided the surface tensions are favorable). Alloying at the interface usually occurs because at brazing temperatures, it's easy for filler metal atoms to diffuse into the base metal (and vice versa). (Alloying is different from the phenomenon called "brass inclusion" sometimes found in bicycle frames, which is generally harmful, as will be discussed in a subsequent part of this series.)

The degree of alloying at the interface depends upon the brazing temperature, brazing time, composition of the base and filler metals, and how well the flux removes oxides. Sometimes it's possible to have alloying at the interface which has a deleterious effect on the joint strength. For example, when steels are brazed with silicon-bearing brazing alloys, the iron and silicon form FeSi (iron silicide). This is a strong but brittle compound called an "intermetallic." If sufficient silicon is present in the brazing alloy (greater than about 0.2%), the FeSi formed at the interface will greatly reduce the strength of the joint.

Other alloy combinations can also produce similar reductions in joint strength. Fortunately it's been determined, by experimentation and/or trial and error, which combinations of metals produce efficient bonding without extensive formation of intermetallics. These favorable combinations are what eventually end up on the market. But remember, not all brazing alloys are compatible with all base metals. I'll say more about that in Part 2.

Topics of succeeding installments will include:

Compatibility of fluxes and filler metals with bicycle tubing and lugs — capillarity; formation of intermetallic compounds; brass inclusion.

Strength of joints (tensile, yield, impact, fatigue); the role of defects in joints.

Strength of steel tubes after brazing — annealing and hardening; temperature gradients versus length of butt in tube.

Proper framebuilding procedures.

DESIGN CRITERIA

Tubing Rigidity

Its Relation to Size Is Dramatic – But Often Misunderstood

Crispin Mount Miller

Many mechanical effects follow equations that contain power functions — something in the equation will vary with the square or cube, for instance, of something else. These variations are often dramatic, so people quote them a lot. Unfortunately, though, the quotes are often taken out of context.

A prime example is the effect of diameter on frame tubing. Some people will say that a tube's rigidity is proportional to the square of its diameter; some will say to the cube; and some will say to the fourth power (and, of course, some people will shun the dramatic exponents and say it's a simple direct proportion). To design any new kind of frame in a rational way, you need to know: which is it?

Some people also will quote the same relationships not for a tube's rigidity, but for its strength. Are strength and rigidity the same thing?

As it happens, most of these assertions can be correct (or approximately so), depending on the assumptions you make. Strength and rigidity are different properties though and they often vary in different ways.[1] I'll start with examples in which some of the proportions quoted above *are* correct, and then I'll go into more detail about why rigidity and strength work as they do. Finally I'll discuss a few of the implications for frame design.

Bending and Twisting

The first step of the description is to specify the kind of strength or rigidity in question. There are four common types, corresponding to the four common ways of applying a load: axial, flexural, torsional, and shear. Axial loading is lengthwise tension or compression; flexural is bending; torsional is twisting; and shear loading tends to move portions of the object crosswise past one another, similar to a stack of cards pushed sideways.

For bicycle frames the important types of rigidity are flexural and torsional.[2] Strength is rarely a problem in normal use but could be of greater concern in modified designs; again the important types would probably be flexural and torsional. (Strength does affect a frame's ability to survive an accident, of course, and the loading to be withstood in accidents seems to be mostly of a bending type.)

The following examples, then, are for flexural and torsional loading. (Conveniently, the rigidities against these two types of loading always change by equal ratios for tubing; and the strengths also change by equal ratios, but not by the same ones used for the rigidities. For example, if a change in tube design increases the flexural rigidity by 10 percent, the torsional rigidity also increases by 10 percent.) The hidden variable that lets all the different exponents be correct is, of course, the tubing wall thickness. Here are four possible permutations:

A. If both the diameter and wall thickness are multiplied by some number — call it k — then the rigidity increases by a factor of k^4 and the strength by a factor of k^3. (Meanwhile the weight for a given length increases by a factor of k^2.)

B. If the diameter is multiplied by k but the wall thickness is not changed, the rigidity increases by a factor of approximately[3] k^3 and the strength by a factor of approximately k^2. (Weight increases by a factor of approximately k.)

C. If the diameter is multiplied by k but the wall thickness is *divided* by k (so that the weight remains approximately the same) the rigidity increases by a factor of approximately k^2 and the strength by a factor of approximately k.

A fourth example is worth mentioning, even though (or because) it doesn't involve a diameter change:

D. If the diameter stays constant and the wall thickness is multiplied by k, the flexural and torsional rigidities increase by approximately the simple factor of k, and so does the strength (and the weight). For any frame design that uses standard lugs and fittings, of course, this is the only change available.

Deducing from these examples, an approximate rule of thumb would appear to be that rigidity depends on the product of the wall thickness and the cube of the diameter; and strength depends on the product of wall thickness and square of diameter.

As we'll see, this rule is a useful approximation, reasonably accurate for thin-walled tubing, but it leaves the reasons (and the exact magnitudes of change) a mystery. Also, common sense dictates that examples *B* and *C* must encounter some sort of limit.

The reasons do take some careful thought, but they aren't very complex (and they equip you to find the limits and the exact values). I'll discuss rigidity first, and then add one more consideration that will explain strength.

Moment of Inertia

The rigidity of an object can be mathematically defined as the ratio between the load applied (of some specific type) and the amount of deformation that results.[4] This ratio depends on three things:

- The stiffness (i.e., modulus of elasticity) of the material itself;
- The amount of material present to resist the deforming load; and
- The shape in which the material is arranged, which determines its mechanical advantage against the load applied.

The modulus of elasticity will depend only on the material itself. The effect of the other two factors — amount of material, and shape — is customarily expressed as a single term called the "moment of inertia" (not to be confused with bending or twisting moments; related to them only through its mathematical ancestry). In the foregoing examples, since they all assume the same kind of material, all the changes in rigidity depend on changes in this term.[5]

Strain Patterns

The moment of inertia is calculated by considering the cross section of an object as the sum of many infinitesimal portions or "elements" (Figure 1). When the object is loaded so as to deform a given amount, each element will suffer a predictable amount of deformation ("strain")[6] determined by its position within the cross section. For example, consider a tube under a bending load (Figure 2): elements on the outside of the bend are stretched, those on the inside are compressed, and the material along the "neutral axis" (located at the tube's midplane) is not deformed at all. Each element is strained by an amount proportional to its distance from the neutral axis.

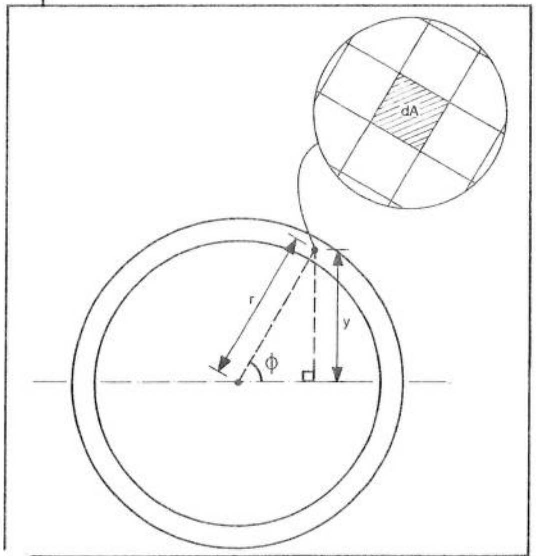

Figure 1: Element of Cross-Section Area

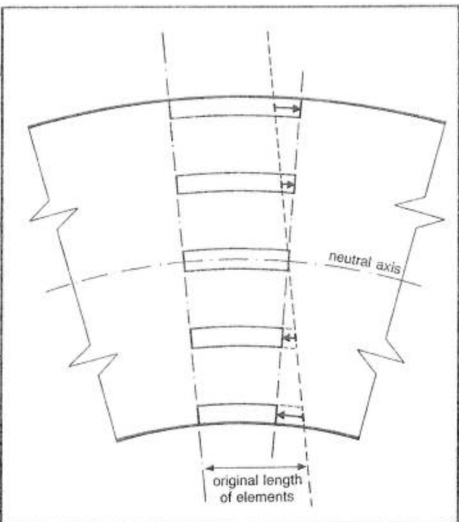

Figure 2: Relationship of Deformation to Position in Bending (exaggerated)

A torsional load creates a quite different deformation pattern. Each element is deformed in shear, as adjacent segments of the tube rotate past one another (Figure 3); and instead of a plane, the neutral axis is a single line along the cylindrical axis of the tube. As with the pattern for bending, though, the strain in each element is proportional to its distance from the neutral axis. (In torsion this proportionality holds true for cylindrical tubing only.)

In either of these loading situations, each element will resist its own deformation on the microscopic level, and, on the large-scale level, contribute a related amount of resistance to the deformation of the tube as a whole. What will be the magnitude of these resistances?

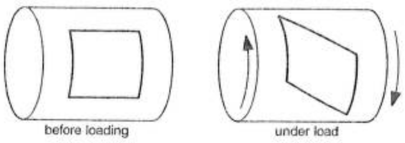

Figure 3: Shearing Deformation under Torsional Load (exaggerated)

Moment Arm — Twice

Both the small-scale and large-scale types of resistance are determined by the distance from the neutral axis (in Figure 1, distance y for bending and distance r for torsion), through the principle of moment-arm (lever) length, but somewhat differently on the two different scales:

The simpler of the two — the resistance of the element to its own deformation — is just the characteristic stress[7] that a material exerts in proportion to the amount of strain in it.[8] Since the strain is proportional to the distance from the neutral axis, so is the stress.

In the resistance to overall deformation, the distance of the element from the neutral axis comes into play an additional time. The force in each element acts to straighten (or untwist) the tube by exerting a *moment* (rather than a simple force), which tends to rotate adjacent portions of the tube back into their original positions. The *force* will be given by the element's area multiplied by its stress. The *moment* will be given by this force *multiplied by the distance from the neutral axis*.

For a given deformation and modulus of elasticity, then, the resistance offered by each element will depend on

- the area of the element; and
- the *square* of the distance from the element to the neutral axis.

The product of these terms defines the contribution of each element to the tube's moment of inertia, and the sum of them (computed as a calculus integral) is the moment of inertia itself. The value of the moment of inertia turns out to be proportional to the fourth power of the radius (or diameter) for any shape that stays the same in all its proportions while its size changes.

(Actually, it isn't necessary to perform the calculus integration to determine this proportionality. If a shape is enlarged by a factor of k and its pattern of division into elements is considered to grow with it — like words printed on an expanding balloon — then:

- Each element's linear dimensions and distance from the neutral axis will increase by a factor of k.
- Each element's area will increase by a factor of k^2.
- Each element's contribution to the moment of inertia will increase by a factor of k^4.)

In direct application, this result is only useful for tubes that retain their proportions during a change in size. But with one more step it becomes applicable to any tube at all:

One shape for which the fourth-power relationship is directly applicable is a solid cylindrical rod — if it just stays round, it retains all its proportions.

Any tube can be regarded as a large rod with a smaller one removed from its middle. Since all elements' contributions to the moment of inertia are additive, the tube's moment of inertia is the difference between that of the large rod and that of the smaller rod.

Any tube's moment of inertia, then, is proportional to the *difference in fourth powers of its outside and inside radii.*

And Strength?

A load exceeds the strength of a tube, and causes permanent deformation, when it becomes sufficient to strain some part of the tube beyond the material's elastic range, so that the stress exceeds the yield strength.

Initially you might think that strength would increase in proportion to rigidity, since rigidity is a measure of the load required to deform the object. But there's a catch:

Rigidity does correspond to the load required to deform the shape of the tube *as a whole*. But strength depends on the maximum value of *strain produced somewhere within the cross section.* For flexure and torsion the relation between this maximum strain and the overall deformation will vary when the diameter changes, because the strain in an element is proportional to the distance of the element from the neutral axis.

As a result, the change of strength with diameter suffers a "penalty," proportional to the first power of the diameter, when compared to the change of rigidity with diameter. The mathematical term that predicts strength (as moment of inertia predicts rigidity) is called the "section modulus,"[9] and is equal to the moment of inertia divided by the distance from the neutral axis to the farthest element of the cross section. Thus, in examples *A*, *B*, and *C*, strength lags behind rigidity by one power of *k;* but in example *D*, where the diameter stays the same, strength keeps pace with rigidity.

Equipped with this knowledge, we can compute some exact values for the examples listed earlier. Table 1 gives values for moment of inertia, and Table 2 for section modulus, which would result if we started with a straight-gauge top tube with an outside diameter of 25.4 millimeters (1 inch) and a wall thickness of 0.8 millimeter, and enlarged it by a factor *(k)* of 1.5 . [10] (The identical bottom lines are not a misprint, but a predictable result of the arithmetic.)

The power-function "rules of thumb," then, turn out to be pretty good for tubes with thickness/diameter ratios in this ballpark — the worst discrepancy between the exact value (given by the difference in fourth powers of radii) and the rule of thumb value is 5.4 percent in these examples. This accuracy is actually almost as good as it is possible to try for, since variations in manufacture will produce errors approaching these: a 0.3 percent deviation from nominal diameter — commonly encountered — will produce a 1.2 percent variation in moment of inertia. (The tables carry results out to three places, but only so as to give a bit of precision in the size of the theoretical error. This three-place precision would be completely spurious to apply to actual pieces of tubing unless it were based on actual measurements of the specific pieces in question.)

If the diameter is doubled (conceivably for tandem frames or recumbents) instead of multiplied by 1.5, the error is greater, but not twice as great — for case *C* with the tube described, the error for k=2 is 9.3 percent. The error also worsens, though, for tubes with greater thickness/diameter ratios.

How do the rules of thumb result from the exact definitions? The approximations can be approached in either of two ways: mathematically or diagramatically.

Mathematically, as shown in the appendix, if the inner radius is expressed as the difference of outside radius and wall thickness, then the difference of fourth powers can be expressed as a series of terms. The first term of this series is the product of the wall thickness and the cube of the radius. When the thickness is small enough (compared to the radius), all the other terms become insignificant, and this first term becomes the rule of thumb.

The diagram approach depends on the division of the cross section into the array of tiny elements. If the pattern of elements is considered to keep its original arrangement while the diameter and wall thickness are changed in various ways, then the individual elements will get stretched and squashed in various ways (Figure 4), and their areas will change. The change in each element's area, when combined with the square of the change in its distance from the neutral axis, will give the "rule of thumb" result.

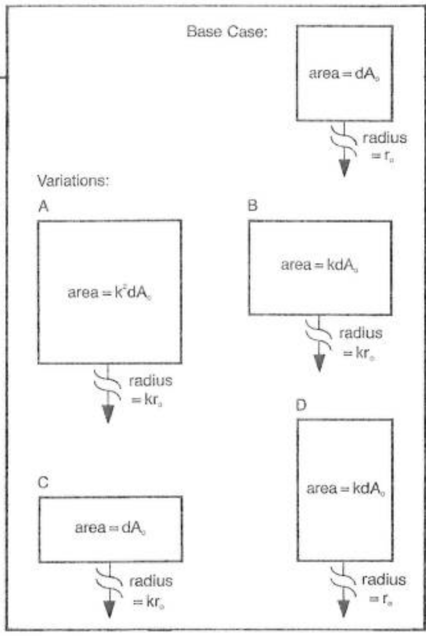

Figure 4: Change in Area of Cross-Section Elements

Implications for Bicycles

When setting out to apply all this to bicycles, one should first question whether rigidity is always desirable. It's nice for the bottom bracket to stay put and for the steering to be definite, but it's also nice, for touring at least, if the bike doesn't ride like a brick. To some extent, a frame design should strive for rigidity under lateral loads but resiliency

Table 1: Moment of Inertia

Example	0 (base case)	A	B	C	D
Diameter (mm)	25.4	38.1	38.1	38.1	25.4
Wall Thickness (mm)	0.8	1.2	0.8	0.533	1.2
Moment of Inertia I ($mm^4 \times 1000$)	4.682	23.702	16.311	11.106	6.695
Factor of Change $\frac{I}{I_o}$	—	5.063	3.484	2.372	1.430
Factor of Change predicted by simplifed power function	—	$(1.5)^4$ = 5.063	$(1.5)^3$ = 3.375	$(1.5)^2$ = 2.250	1.5
Error of simplified power function: $\frac{\text{(simpl.)} - \text{(exact)}}{\text{(exact)}}$	—	0	+ 3.2%	+ 5.4%	− 4.7%

Table 2: Section Modulus

Example	0 (base case)	A	B	C	D
Diameter (mm)	25.4	38.1	38.1	38.1	25.4
Wall Thickness (mm)	0.8	1.2	0.8	0.533	1.2
Section Modulus S $= \frac{I}{r}$ (mm^3)	368.6	1244.2	856.2	583.0	527.2
Factor of Change $\frac{S}{S_o}$	—	3.375	2.323	1.581	1.430
Factor of change predicted by simplified power function	—	$(1.5)^3$ = 3.375	$(1.5)^2$ = 2.250	1.5	1.5
Error of simplified power function: $\frac{\text{(simpl.)} - \text{(exact)}}{\text{(exact)}}$	—	0	+ 3.2%	+ 5.4%	−4.7%

under vertical ones. That's a tall order, since the geometry of the frame tends to create exactly the opposite combination. As a result, most of the vertical resiliency in a frame usually comes from the fork. Some of it, though, may come from the main "triangle," at least with certain frame shapes.

A frame's rigidity can be tailored to respond differently to different loads — such as pedaling, hard steering, and road shock — if the designer can discover the extent to which the deflection for each type of load depends on the rigidity of specific frame tubes (and on the shape of the frame). The designer can then choose an appropriate frame shape and set of tubes. This has long been an art and is gradually becoming a science. We're working on it ourselves but I won't go into it here.

The additional variable of strength can also be tailored, somewhat separately from rigidity, if a design should require it. The easiest way, of course, is to change both strength and rigidity at once by changing the wall thickness (as in example *D*), but in an extreme case one could actually strengthen a tube without changing the rigidity (but with a penalty in weight), by reducing the diameter and markedly thickening the wall (by a factor of roughly k^2, if the diameter were divided by k).

Down Tubes and Beer Cans

Standard present-day bicycle frames do show recognition of the effect of diameter in one aspect: the down tube and seat tube, which typically bear greater loads than the top tube — torsion in the down tube, and lateral bending in the seat tube — have a diameter 12½ percent greater than the top tube (1⅛ inch instead of 1 inch). For a given wall thickness, this extra diameter gives the larger tubes more than 40 percent more rigidity, with only about 12½ percent more weight.

Obviously, though, there's a limit to the amount you can enlarge the diameter without thickening the wall — especially if you choose the even more tempting, constant-weight course (example *C*) of increasing the diameter while *thinning* the wall. Eventually the wall fails by crumpling, a failure mode called "local buckling" (also known as "the beer-can effect").

A customary rule of thumb used by engineers to avoid local buckling is that a tube's wall thickness should be no less than 1/50 of its diameter. Most high-quality steel bicycle tubing turns out to be fairly close to this limit, at least in the midsection: for 1-inch tubing the rule gives a minimum thickness of 0.51 millimeter and for 1⅛-inch tubing, 0.57 millimeter. In lightweight steel tubing, then, the main frame tubes are proportioned to have about as much strength and rigidity as possible, and to be prudent any modification to increase these qualities must include an increase in thickness and therefore in weight. A few builders consider this tradeoff worthwhile for single-rider frames. For tandems, of course, it can be emphatically worthwhile.

Room for Growth

If the tubing is not steel but aluminum (or titanium), the optimum diameter changes dramatically, because for a given weight and diameter, an aluminum tube (for instance) has roughly three times as thick a wall as a steel tube has. As Gary Klein has pointed out,[11] this means the aluminum tube can be enlarged and thinned (as in case *C*) by a factor of roughly 1.7 before it reaches the same thickness/diameter ratio as the steel tube of the same weight. This enlargement will produce multiplications of approximately (by the power-function rule of thumb) 3 in the section modulus and 5.2 in the moments of inertia. While aluminum is neither as strong nor as stiff as steel, these increases more than overcome the differences.

One part of a standard steel frame that isn't close to the 1:50 ratio, and probably could use more rigidity, is the chainstays. These members have a wall thickness comparable to that of down tubes, but their maximum diameter is rarely more than ⅘ as great. Framebuilder Tom Kellogg points out that torsion and bending occur in chainstays when the bottom bracket tilts under a pedaling load. He contends that chainstays should be made in large diameters so that their additional rigidity would help hold the bottom bracket still. I agree. (Some details that might need watching, though, would be the rigidity of the rear dropouts and axle and of the chainstay-bridge attachments, so that none of these becomes fatigued by being a "weak link" attached to chainstays that are more rigid than before.)

One more issue comes to mind: aerodynamic frame tubes. For lateral rigidity, these tubes are a disaster. When a tube is squashed into a vertical oval, bringing all its material closer to the vertical plane which is the neutral axis for lateral bending, the moment of inertia about this axis takes a beating, unless the wall is thickened considerably. (For instance, if the wall is thickened uniformly, it must be thickened by the square of the ratio between original and "squashed" diameters. Nonuniform thickening — more on the sides and less on the top and bottom — can do the job with a little less weight, but not much, and it's a considerably more specialized job.)

An oval-tubed frame, then, will either be heavier than a round-tubed one (if its walls are thicker) or whippier (if it's the same gauge). Such a frame may win a coasting contest, but a road race may be a different story. (Or it may not. I'm still looking for someone who knows what frame rigidity is worth ergonomically. Anyhow, for the sake of science, I for one would like to see a race include a bike built out of oval tubing turned *sideways*.) We haven't gotten our hands on an aero frame since we got our frame-rigidity testing machine, but when we do, we'll let you know.

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[1] *Rigidity is the ratio between a loading force or moment (torque) applied and the deformation that results. Strength is the magnitude of the greatest load that the object can bear without suffering permanent deformation.*

[2] *Within the plane of its frame, a bicycle has a good degree of triangulation, so that it can bear most loads applied from within this plane — vertical and forward or backward loads — as axial (lengthwise) loads on the tubes, with magnitudes and resulting deflections that are fairly small. Against lateral forces, however, the frame is barely braced at all, and it must resist these forces as flexural, torsional, and shearing loads. Shearing deflections do occur, but, like axial ones, they are fairly small. The bending and twisting loadings, however, create relatively large stresses (because, unlike axial and shear loadings, they involve* rotational *deflections, and therefore allow the lengths of the frame tubes to come into play as levers). Consequently they cause large deflections. Most of the deflection of a bicycle frame results from bending and twisting of the tubes in response to lateral forces.*

[3] *These results would apply exactly for tubing whose inside and outside radii both changed by the same proportion. However, all these examples except* A *involve changes in the thickness/diameter ratio, and therefore violate this requirement. There is one loophole: the hypothetical case where the wall has no thickness, so that the inside and outside radii are equal and must therefore change at the same rate. (The only reason these examples are even approximately accurate, in fact, is that bicycle tubing happens to be fairly close to this condition.)*

[4] *I've attached the mathematical expressions involved at the end, for those who want to see them; but this discussion can be read without them.*

[5] *Actually a cylinder has two different moments of inertia — one for flexure and one for torsion — but they always change simultaneously by the same proportion, so the magnitude of change can be discussed for both at once. (Numerically, the moment of inertia for torsion — called the "polar moment of inertia" — is exactly twice the moment of inertia for bending.)*

[6] *Strain is the ratio of elongation (or shear displacement) to the original length (or, for shear, to the width across which the shearing takes place).*

[7] *Stress is the concentration of force in an element — the ratio between the force borne by the element and the cross-sectional area of that element.*

[8] *The ratio between stress and strain is the modulus of elasticity.*

[9] *The name "section modulus" is traditionally used only for the expression that applies to bending; but for cylindrical tubing the analogous expression for torsion applies. As with "moments," "section modulus" is related to "modulus of elasticity" only by their shared mathematical origin.*

[10]*The lines of the tables that give actual numerical values for moment of inertia* I *and section modulus* S *depend on an equation from the appendix:*

$$I = \frac{\pi}{4}(r_o^4 - r_i^4)$$

but all the other columns could be calculated even without knowing the initial numerical value, since the coefficient $\frac{\pi}{4}$ *cancels out.*

[11]*See Gary Klein's "A Hundred Years of Monopoly: Is Steel The Ultimate Frame Material?",* Bicycling, *September/October 1981.*

Appendix: Equations

Rigidity

Bending:

$$rigidity = \frac{M}{K} = E\,I$$

M = bending moment applied

K = curvature = $\frac{1}{R}$

where R is radius of the curvature caused by load

E = material's modulus of elasticity

I = moment of inertia (for bending)

Torsion:

$$rigidity = \frac{M_t}{\Theta} = G\,J$$

M_t = torsion moment applied

Θ = angle of twist (rotational displacement) per unit length

G = material's shear modulus of elasticity

J = polar moment of inertia

Moment of Inertia

Bending:

$$I = \int y^2\,dA$$

$$= \int (r \sin\phi)^2\,dA$$

$$= \int_{r_i}^{r_o}\int_0^{2\pi} r^3 (\sin\phi)^2\,d\phi\,dr$$

$$= \frac{\pi}{4}(r_o^4 - r_i^4)$$

Torsion:

$$J = \int r^2\,dA$$

$$= \int_{r_i}^{r_o}\int_0^{2\pi} r^3\,d\phi\,dr$$

$$= 2\pi\int_{r_i}^{r_o} r^3\,dr$$

$$= \frac{\pi}{2}(r_o^4 - r_i^4)$$

I = moment of inertia

J = polar moment of inertia

$\int_a^b$ = integral (calculus summation of the value of the expression following it, for all values of its variable in the range from a to b);

$$\int_0^{2\pi} (\sin\phi)^2\,d\phi = \pi$$

$$\int_{r_i}^{r_o} r^3\,dr = 1/4(r_o^4 - r_i^4)$$

$$\int_0^{2\pi} d\phi = 2\pi$$

y = distance from neutral axis (see Figure 1)

dA = area element = r dr dφ

dr = element of radial length

rdφ = element of circumference

φ = angular position of element (see Figure 1)

r_i = inside radius of tube

r_o = outside radius of tube

Section Modulus

$$S = \frac{I}{c} = \frac{I}{r_o} = \frac{\pi}{4}\,\frac{(r_o^4 - r_i^4)}{r_o}$$

S = section modulus

I = moment of inertia

c = maximum distance of any element from the neutral axis = r_o for tubing

Rule of Thumb Approximation

Let $T = r_o - r_i$

then $r_i = r_o - T$

and $r_i^4 = r_o^4 - 4r_o^3T + 6r_o^2T^2 - 4r_oT^3 + T^4$

Then $r_o^4 - r_i^4 = 4r_o^3T - 6r_o^2T^2 + 4r_oT^3 - T^4$

If $T << r_o$

then $6r_o^2T^2 << 4r_o^3T$

so $r_o^4 - r_i^4 \doteq 4r_o^3T$

and remaining terms are even smaller;
Substituting this value into the expressions for I, J, and S gives

$$I = \pi r_o^3 T$$

$$J = 2\pi r_o^3 T$$

$$S = \pi r_o^2 T$$

(It should be noted that these rules of thumb are usually more accurate to predict ratios of change than to predict actual magnitude; e.g., "$\pi r_o^3 T$" gives a figure 10 percent high for moment of inertia in case *0* of the examples.)

RESEARCH

Getting The Numbers Right

Human Power Research Methodology, Part 2: Drag Measurement

Paul Van Valkenburgh

In this issue we present the second of three installments of Van Valkenburgh's paper from the IHPVA Scientific Symposium of November 1981. This portion covers the area of drag measurements on human-powered vehicles. (Part 1, in Bike Tech *June 1982, contained the sections of ergonometry and computer simulations; part 3 will cover the topics of stability and safety evaluation.) Proceedings of the entire IHPVA Symposium are available for $16.60 postpaid from: IHPVA, c/o Dr. Allan Abbott, P.O. Box AA, Idyllwild, CA 92349.*

Ground vehicle air drag has gained enormous public importance in just the past few years. Drag coefficient, or Cd, has almost reached the status of mpg. However, the believability of *reported* Cd is no better than *reported* mpg (or *reported* Hp of the previous decade). There are more than a dozen ways of measuring air drag, as shown in Table 1, each with its own unique problems and advantages, and all of them difficult to correlate with each other on identical bodies.

Tunnel Problems

Wind tunnel testing is either too imprecise in available low-cost college tunnels, or too expensive in the necessary large-scale, high-speed aircraft tunnels. Low-cost tunnels suffer from several problems of precision or realism which make them useful only for crude A-B tests on the same model.

First, HPVs have a surprisingly large Reynolds number ("Rn," a function of length times air speed), and unless this is closely duplicated, model data can be significantly inaccurate.

Second, sophisticated laminar-flow bodies are affected by airstream turbulence, and it is very expensive to reduce this turbulence to an acceptable level.

Third, HPVs operate very close to the ground, which has no boundary layer in still air. All wind tunnels without a moving ground plane have varying degrees of boundary layer regardless of exotic methods used to shave or suck it off ahead of the model.

Finally, on many HPVs, the spinning wheels can contribute a large proportion of aerodynamic skin friction drag, so that for the simulation to be realistic — especially around the wheel openings in the body — the wheels should be spun on the model also.

The size and quality of tunnel necessary to overcome these problems is represented best by one of Northrop Corp.'s larger tunnels, which costs around $600 per hour with a minimum setup and shutdown cost of about $10,000 (without a moving ground plane), or perhaps the Caltech tunnel at about a third the cost. Of course, these costs do not include the scale models, which if accurately detailed, can cost thousands of dollars themselves.

Strain Gauges

Strain gauging of full-size vehicles is one way of measuring aerodynamic forces under real world conditions. In the case of automobiles, attempts have been made to strain gauge body mounts and drive axles with limited success, and the problems are worse with lightweight HPVs which may have aerodynamic forces of less than five pounds at 55 mph.

Measurement of such small forces requires precise, expensive, and heavy equipment for on-board recording. In addition, normal roadway vibration forces are comparatively significant and difficult to filter out of the desired signal. Also, to get a meaningful drag figure, speed has to be *absolutely* constant, to prevent longitudinal g-force effects, and this is difficult to control in any vehicle.

Finally, strain gauging a tow rope or push bar is impractical not only because of the constant speed problems, but because of interference effects. For laminar-flow bodies, a tow vehicle would have to be hundreds of feet ahead to eliminate turbulence, or many vehicle lengths to the rear to avoid "bow pressure."

Coasting

Coastdown tests have become very popular simply because they are practical and not because they are simple. The first requirement is a very long, very smooth, very flat piece of pavement — unless terminal speed on a slope is used, which is inaccurate because the true instantaneous slope is seldom known.

Even on an apparently flat road, slight grade variations of one-tenth of one percent can be a problem. For example, if total drag is five pounds, and vehicle weight is 200 pounds, a 0.1% grade ripple in the road will affect the data by 0.2 pounds, or plus or minus 4% of the total, which could be greater than the modification being evaluated. There are many other variables which must be carefully controlled, such as ambient winds, weight, and tire conditions. For example, automobile tire drag can vary more than 10% unless first stabilized by a warmup of about 20 miles.

Given a suitable surface and controlled conditions, there is still the problem of coastdown data recording and interpretation. Recording time over distance, or time for a speed drop, is simple and easy, using either a manual or automatically triggered stopwatch. However, such data only gives discrete point averages, and many runs must be made to get a good plot of deceleration versus speed.

A better method is the continuous analog speed recording, using a tach-generator output from a road wheel. With this method, the strip chart plot may be analyzed by taking any number of time-speed intervals, or slopes of the curve, on a single coastdown run. A more sophisticated technique is to electronically differentiate the speed-time signal and calibrate the output in g's versus speed.

An example of a good data recorder and speed transducer is shown in Figure 1. This is a self-contained four-channel cassette recorder which weighs only about 10 pounds and can be worn by the rider, and a tach-generator which has immeasurable drag or ripple. If the drive is taken off the circumference of the tire, calibration does not change from vehicle to vehicle.

A final complication of coastdown tests is that conversion of deceleration into air drag force requires a good estimation of rotational inertia and factoring-out of the rolling tire drag. Rotational inertia of the wheels and driveline can increase the apparent mass by 3% to 6%, which has a significant effect on drag calculations. And unless computerized curve-fitting methods are used, there can be large errors in the attribution of drag to weight, rolling friction, and aerodynamics.

Figure 1: Four-channel cassette data recorder and tach-generator pickup by rider's leg. Marc Cole of Honda taking data.

Wake Survey

Air pressure integration, either over the body surface or in the body wake (Figure 2), may seem to be unnecessarily complex and imprecise, but it is the method that NACA (now NASA) uses to evaluate laminar-flow shapes. An example of the wake survey being used to evaluate laminar flow on an HPV

Table 1: Air and Rolling Drag Measurement

	Cost	Precision	Realism	Roll Drag
Wind Tunnel				
Small scale/Low speed	$ __00.	0.01	Poor	No
Small scale/High speed	__000.	0.001	Good	No
Full scale	__0000.	0.001	Very good	No
Moving ground planes	double	0.001	Excellent	No
Full Scale Strain Gauge				
Gauges on body	__00.	0.001	Good	No
Gauges on driveline	__00.	0.01	Poor	No
Gauges on push/tow bar	__00.	0.01	Poor	Yes
Coastdown Methods				
Continuous speed record	__00.	0.01	Very Good	Yes
Electronic differentiation	__000.	0.01	Very Good	Yes
Time over distance	__00.	0.01	Good	Yes
Time over speed drop	__0.	0.01	Good	Yes
Speed down slope		0.1	Poor	Yes
Air Pressure Integration				
Over body surface	__0.	0.01	Poor	No
Wake survey	__0.	0.01	Good	No

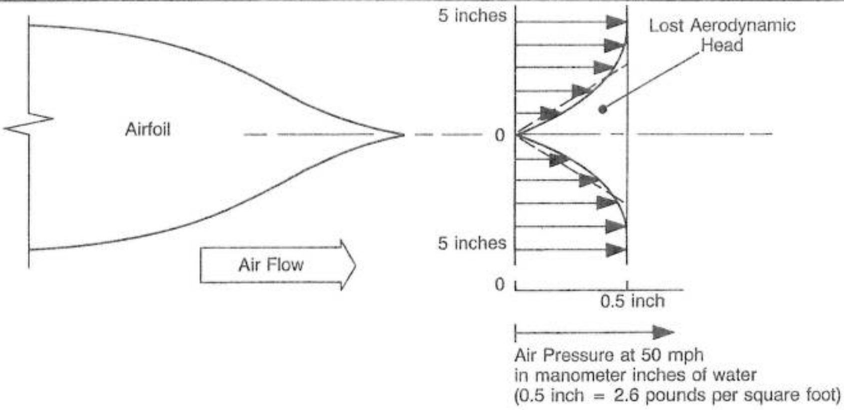


Figure 2: Method of measuring air drag by a wake survey on a simple airfoil. Drag per foot of airfoil span is calculated by integrating lost pressure over thickness of airfoil — done here by approximating curve with dotted triangle:

$$\text{drag per foot} = \text{area of triangle} = \frac{2.6\ \text{lbs/ft}^2 \times 0.5\ \text{ft}}{2} = 0.65\ \text{lbs/ft}$$

or for an airfoil 3 ft. wide (e.g., an HPV 3 ft. high) a total drag of 1.95 pounds.

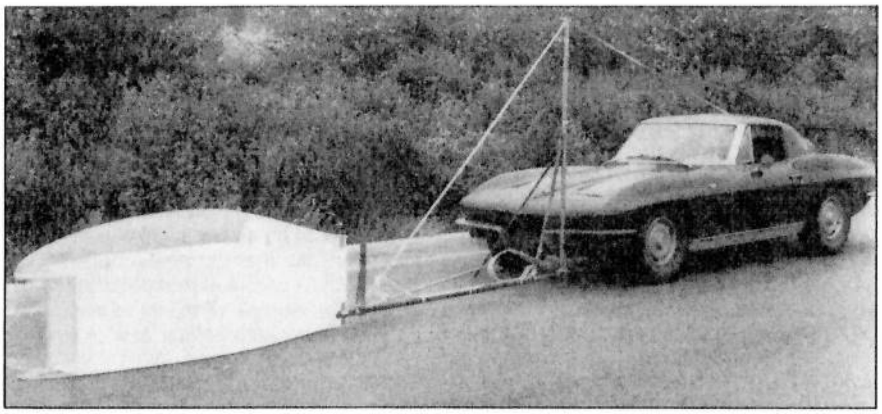


Figure 3: Aerodynamic body mounted on push bar ahead of low-drag instrumentation/push vehicle.

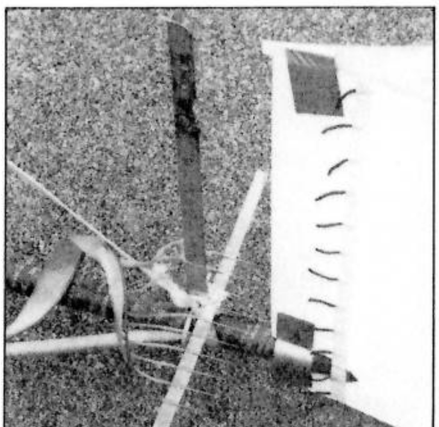


Figure 4: Aerodynamic pressure pickup rake (of fine open-ended tubes) mounted at the trailing edge of airfoil body.

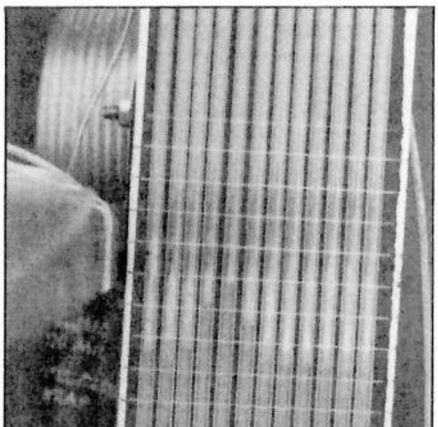


Figure 5: Multitube manometer used to read out pressure rake data.

is shown in Figures 3-5. A mockup of the body is mounted one chord length ahead of a low-drag instrumentation/push vehicle (Figure 3). While this is not enough spacing to avoid interference effects for accurate drag measurement, the minimal bow pressure has little effect on flow conditions. A pressure rake (Figure 4) and multitube manometer (Figure 5) are mounted to the push bar.

With the rake in various positions in the wake, photos are taken of the manometer at 50 mph. The "inches of water" are converted to average psi and multiplied by wake area to give approximate drag in pounds. But of greater interest is the profile of the boundary layer at the trailing edge. Figure 6 indicates that it is only about three inches thick, and definitely not separated; and within the precision of the instrumentation, the indication is that it has good laminar flow.

Conclusion

The object of this discussion was not to describe in detail how to conduct aerodynamic tests, but to show some of the considerations involved in selecting the test to fit the problem.

I should point out that this article was written primarily about aerodynamic testing of fully enclosed HPVs. Actually, right now I am involved in some wind tunnel testing of conventional components, with fairly good results because each component is small and the slow speed, turbulence, and ground effects are not a factor. Therefore we will discuss wind tunnel testing in more detail in a later article.

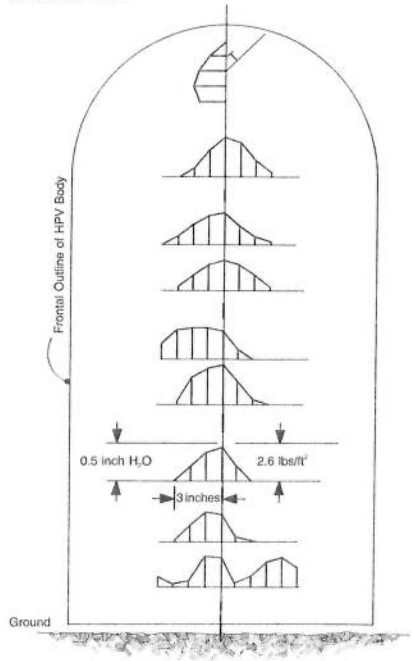


Figure 6: Wake survey air pressure plots at the tail of a NACA 66021 airfoil HPV body.

SHOP TALK

Too Far Gone?

Recovering Damaged, Lightweight Wheels

Eric Hjertberg

When is a wheel too far gone? When is it irreparable? The answer is, NEVER! Famed adventurer Ian Hibell tells the story of fixing his broken and battered rear wheel while crossing Africa. He splinted the rim at the point of each fracture and sometimes wrapped the rim with hemp cord. Finally, he fashioned stout wood sticks to brace between the hub center and the rim, like the spokes of a wagon wheel. The whole thing looked more like a decorative garden trellis than a bicycle wheel.

Hibell did everything except attach a python in place of his tire (think of the traction . . .). The results? He successfully continued his trek, village to village, until he found a bike shop. When it is a matter of necessity the wheel can always be repaired.

The big question we face daily in the shop is not, then, whether wheels *can* be repaired, but *which* bent rims to save and how. The two parts of the decision are: one, what is the mechanical feasibility of a given repair; and two, what are the economic costs to our shop and the customer, both immediate and long range.

Before exploring the mechanical side of wheel repair let us restrict our attention to spokes and rims. Hubs, tires, and tubes are often in need of work but for the sake of this discussion, think of wheel repair from the wheelbuilding point of view. In addition, my comments pertain mainly to lightweight, high-performance bicycles since they are the bulk of machines needing our help.

Disc Brakes

Wheels do more than support the weight of bicycle and rider and provide a rotating center to allow forward movement. They are a fundamental part of the modern brake system. When one thinks of bicycle brakes, normally handles, cables, and calipers come to mind. But the wheel rims are an integral part of the system. Most bicycles use a form of disc brake with the rim serving as disc, which has much to do with the way they are shaped, sized, and maintained. In fact, most wheel repair pertains to the rim's ability to serve as a steady, smooth brake "disc" since it is this ability that is most easily damaged. Think about it — the majority of wheel complaints that reach a shop are provoked by the rim hitting the brake blocks through wobbling or because of a dent or "blip." If it were not for caliper brakes our wheel standards would be vastly different.

Dent Epidemic

This brings us to a most common wheel service problem — the minor rim dent. The epidemic rate at which riders now dent their rims is a reflection of:

1) The inexperience and lack of concentration many cyclists bring to their riding.
2) The vulnerability of modern, narrow clincher rims to point impacts.
3) The popularity of "too small" tires which offer little protection from bumps and potholes.
4) Deteriorating road surfaces and the desirability of riding on secondary, less well paved roads to escape traffic.
5) The unsafe, high speeds which many riders maintain in a useless race with autos and their own schedules.
6) Underinflated tires, particularly on novice riders' bicycles.

Regardless of these causes, and others, most minor rim dents can be simply corrected.

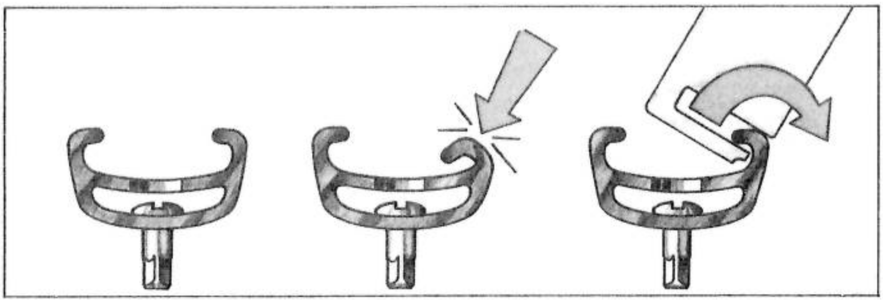
The small lips that "one-inch" clincher rims employ to hold the tire are easily crushed inward. In many cases the main body of the rim is undeflected and the bent edge can be lifted up and out, restoring the rim to new condition (see diagram). A small Crescent wrench adjusted as narrow as possible or the Bicycle Research "La Jeunesse" chainwheel straightener can be used to lever the folded edge back to normal. Only minor retruing is required to recover the rim.

If a narrow clincher is dented so that the main body of the rim is deformed ½-inch or less, loosen the nearby spokes, support the rim on a wood block, and try pounding out the dent — either with a soft mallet or by placing another block on top of the rim and hammering on the block[1] — but be prepared to find that even after a half-hour of careful work you may produce a wheel you would not want to own and use.

Sew-up rims usually make stronger wheels than similar weight clincher rims because they form larger, more rigid tubes in cross section. But they also have thinner sidewalls more prone to deformation. What good is a round, strong wheel that throbs mercilessly every time the brakes are applied? The brake-bothering deformation may be small, but any inconsistency will quickly cause a buildup of brake shoe material, and the superficial scar will feel like a full-sized dent.

A lot of riders have trouble with brakes throbbing on perfectly healthy rim joints, let alone real deformations. High-performance cycling depends on precise control of road speed. Throbbing brakes are more than a nuisance, they are downright dangerous.

Dents to steel and heavier aluminum rims can be squeezed with rim pliers or a smooth-jawed vise. It takes only a few short seconds to transform a rim with multiple dents to tolerable performance. If the rim is dented inward from overcorrection, though, it will be a constant disappointment to its owner. So squeeze gently and repeatedly until you get the desired results. The larger the tire, the less likely that the remaining bend in the rim will be noticeable to the rider.



Lubricate Nipples

Even without dents or sideways blows, wheels often develop minor wobbles (under ¼-inch). Rim wobbles are usually due to settling of the materials. Better-built wheels, by virtue of high tension and prestressing, settle less, but even these can change shape. When touching up a minor wobble, lubricate the nipples so adjustments will be easier. If a wheel is loose, simply truing it may solve the immediate wobble, but the wheel will be more stable in the future if it is tightened as well. Just a half-turn on each nipple can double the tension in some wheels. You may find the firmer wheel easier to straighten since the tighter spokes possess more corrective power over the rim.

[1]*A more detailed description of this technique can be found on pages 123-124 of Jobst Brandt's book* The Bicycle Wheel *(Avocet, Inc., 1981).*

Another common wheel ailment is the broken spoke. Spokes break usually without warning from fatigue. Before undertaking replacement consider how many have broken in the past. If the answer is five or more it is only fair to explain that all the wheel's spokes are probably tired and that it is much wiser and cheaper to rebuild or replace the wheel at once. If few spokes have failed then the replacement has a good chance of being worthwhile.

With lighter wheels the broken spoke may produce a nasty major kink in the rim. If the sideways deflection of the rim is as much as one inch the chances are that the rim is wrecked. Once the new spoke is installed, loosen the opposing pair, lay the wheel flat with the kink against the ground, and push down on the rim to either side of the kink.[2] In about half the cases an acceptable repair can be made.

If the chain is overshifted past low gear the nine nearest spokes can be damaged. Replace the nine, one at a time, tightening each until the wheel is true before the next is undertaken.

"W" Bend

One curious result of a single broken spoke is the multiple bend that usually appears in the rim. Do not let it scare you. When a spoke fails the two neighboring, opposing spokes are suddenly allowed to pull the rim in their direction. But as they do so their tension decreases and they seem looser than their adjacent, outboard neighbors. This forces the rim into two smaller bends opposite to the first major deflection. The resultant shape is a "W" all caused by the failure of only one spoke. When the new spoke is tightened the bends will disappear.

Whenever a wheel is subjected to an excessive side force, a kink in the rim appears. The most common form for this bend is the classic "W" as with a broken spoke. If the bend is less than 1/4-inch, spoke retensioning is often enough to erase the damage. When the deflection is greater than 1/2-inch, some extra levering or bending of the rim is required.

A typical correction for a local kink begins with loosening spokes several turns on the side of the problem and inserting the bent section into a narrow slot, like a partly opened heavy drawer. With caution, the wheel can be pressed down twisting the errant section toward the correct direction.

Another popular strategy is to lay the rim on its side, supported next to the kink by wood blocks. With opposing spokes loosened the kink can be gently struck with a mallet forcing it straight.

[2]*Again, Brandt gives a detailed description of the technique, under "soft wheel failure" on page 122.*

The goal of each of these methods is a small overcorrection, allowing the loosened spokes to be tensioned once again, bringing the rim back to normal. The biggest danger with any such repair is a residual discontinuity in the rim that the brakes will notice. Brake irregularities are an unacceptable result of an otherwise successful repair, and should only be tolerated when budget or time leave no option.

Growing Cracks

You must keep a careful watch for cracks in the rim material, especially with heat-treated rims. Although heat treatment and surface anodizing (coloring) render rims more resistant to dents, the metal becomes more brittle and less able to withstand bends without cracking. The rule with thin metals (this includes frame tubes) is that cracks migrate and grow. Those near a nipple hole will advance until the entire nipple pulls clear through the rim. Little can be done to slow this disintegration so cracked rims are best replaced as soon as possible.

Thunk

The most dramatic wheel damage of all is the classic potato chip shape that represents complete failure of the wheel as a tensioned structure. There is a small chance that it can be sprung back to roundness (and tension) by slamming the rim sideways onto the floor with a single well-placed thunk. This "miracle" repair is more likely with a heavy rim and low initial tension.

Every time someone brings a potato-chipped wheel into our shop, we try this one-thunk repair. However, our success rate is low, especially with thin-walled aluminum sew-up rims and narrow aluminum clincher rims.

Even at its best, the "repaired" wheel usually falls short of desirable. A rim so bent may be predisposed to return to its deformed state and ought not to be trusted.

Warped

Well-built, lightweight wheels which are badly warped should not be repaired. If ever the supply of new rims is completely interrupted we will perfect methods of recovering these lost rims. Such a technique would start with complete unbuilding and then gentle bending of the rim until it is completely flat. Only then could spokes be reconnected and some moderate tension introduced.

The job the rim has of supporting competing spoke tensions is so demanding that it must begin the task in a very straight condition. Spokes are mainly supporting the weight of the bike and rider, and only secondarily can they hold a crooked rim straight.

So much for advice on some mechanical aspects of wheel repair. How can a professional mechanic decide what feasible repair is also economical to perform? With U.S. wheelbuilding rates varying from $12 to $30 per wheel this decision is bound to be unique to each business. Still, there are some considerations common to us all.

Take the rider's point of view. Many cyclists have spent hundreds of dollars to obtain a high-quality ride. They have difficulty finding time to ride and often will not appreciate a small savings if they are reminded of the damage every time the brakes are applied. So, as tempted as you may be to produce a "miracle" repair, put yourself in the user's position. Many riders spend five times as much on rent each month as an entire new wheel would cost.

Even though your repair department may be crowded with damaged wheels, such cases are not just a "fact of life" like punctures. It is often better to recommend a sound rebuilding or an outstanding replacement wheel than to struggle with a repair you would not want to live with yourself.

Then help the rider avoid the damage in the future with larger tires, a heavier rim, and safer riding habits. Without accidents and traumas a well-built wheel should last many tens of thousands of miles.

Try Unlikely Repairs

From time to time it is wise for builders to try unlikely repairs just to see what happens. These experiments are best performed on wheels destined for rebuilding. This is a good way to learn but it is slow. The most efficient way to learn is for a novice to watch an experienced builder work. As long as the teacher is not reluctant to explain, the learner can gather techniques and strategies much faster than by experimentation.

If you are stranded in an African jungle, experimentation may be the only way out. For the rest of us, wheel repair is more of a nuisance than a matter of survival.

The simplest jobs are small blips, especially in one-inch clincher rims; single broken spokes; and minor wobbles (under 1/4-inch). Tougher, less certain repairs are sideways bends and dents of 1/2-inch or more, requiring levering of entire rim sections.

Wheel damage with little chance of successful repair includes cracked rim material, the classic potato chip collapse, and deformation of the rim brake surface (beyond the simple squeeze remedy) causing dangerous throbbing when the brakes are applied. Remember to take the user's point of view. Frequent riding near your personal limit of speed and endurance requires like-new wheel performance for pleasure and safety. With these points in mind I hope your wheel repair decisions will be easier.

DYNAMICS

"Balancing and Steering"

From *Bicycling Science*

Frank Rowland Whitt and David Gordon Wilson

The second edition of Bicycling Science *by Frank Rowland Whitt and David Gordon Wilson is due to be issued this September. The new edition contains a large portion of new material, much of which reflects new research in several areas, especially the explosion of activity in experimental human powered vehicles. There is also a new chapter on the history of human powered machines and vehicles, written with interesting technical insights about the significance of the advances described.*

One new portion that applies to both new and traditional machines is the analysis of steering geometry. We print a portion of that discussion here.

Copies of the book will be available for $19.95 (hardcover) and $9.95 (paperback) from bookstores or from MIT Press, 28 Carleton Street, Cambridge, MA 02142.

Balancing and Steering

The balancing and steering of bicycles is an extremely complex subject on which there is a great deal of experience and rather little science. We will report what we believe to be the best of each. Both approaches — experience and science — have attempted to answer [the following] questions:

- What are the geometrical relationships that can give the single-track vehicle, considered as a rigid body, "good" steering characteristics? This question concerns riding at any speed, but particularly at low speed, when steering angles (the angle through which the handlebar is moved by the rider) can be large. Above 2-3 m/sec (9-13 mph) the handlebar cannot be turned by more than a few degrees to either side of the straight-ahead (neutral) position without the rider being thrown out of balance and, usually, off the bicycle.
- What other factors, when combined with geometrical relationships, can avoid the steering instabilities known as shimmy (a rapid oscillation of the front wheel about the neutral position, usually occurring rather suddenly at fairly high speed)?

A vibration or an oscillation is usually similar to the bouncing of a weight hung on an elastic thread. The occurrence of steering oscillations implies, therefore, that the elasticity of the structure and possibly of the rider is involved. Let us defer discussion of this complex question until after we have considered the still-complicated question of the steering characteristics of a bicycle considered as a rigid body.

Steering Characteristics of Nonflexing Bicycles

The geometry is complicated because of the many angles involved and because of the offset of the front fork (Figures 1 and 2). Three important angles are:

- the steering-head angle (the most important of the angles defining the frame), which is usually between 68° and 75°;
- the steering angle, or the angle of the handlebars from the straight-ahead, neutral position; and
- the angle of lean of the bicycle frame to the horizontal.

If the fork had no offset, the front wheel would sweep out a sphere as it was turned. With the offset, the wheel sweeps out a "doughnut." It is the combination of bicycle lean with the front wheel position as a slice of the doughnut that makes bicycle geometry so complex. An additional problem when we account for the gravitational and centripetal forces occurs if we wish to account for a nonhorizontal ground surface.

There is no real disagreement about how a rider steers and balances a bicycle. One steers into or under a fall, just as one balances a broomstick on a finger. The following questions have intrigued many people, including some famous mathematicians and applied mechanicians such as Timoshenko (ref. 1):

Why are some bicycles easier to steer than others?

Why do some bicycles steer themselves easily, whereas others do not?

What are the effects of steering angle, fork offset or trail, height of center of gravity, and so forth (Figure 1)?

However, members of the lay public who have perused the scientific literature (refs. 1-6) have been intrigued to find often complete disagreement among the experts, even about fundamentals. One advocates a high center of gravity for stability; another concludes from the equations that a low mass center is desirable. One finds that gyroscopic action is important; another the exact opposite. We have found that the most useful and relevant information about bicycle steering and stability is that given by David Jones, a chemist who looked into bicycle stability as a diversionary project (ref. 7).

Jones set out to build an unridable bicycle (URB). In his URB I, he canceled out the gyroscopic action of the front wheel by mounting near it another similar wheel which

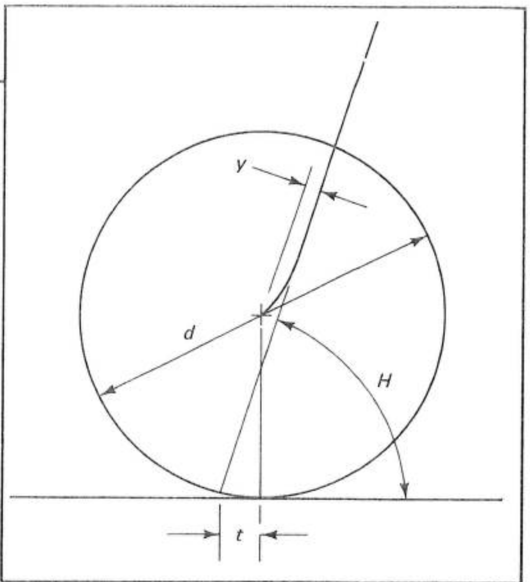


Figure 1: Front fork geometry. H = head angle; y = fork offset; d = wheel diameter; t = trail.

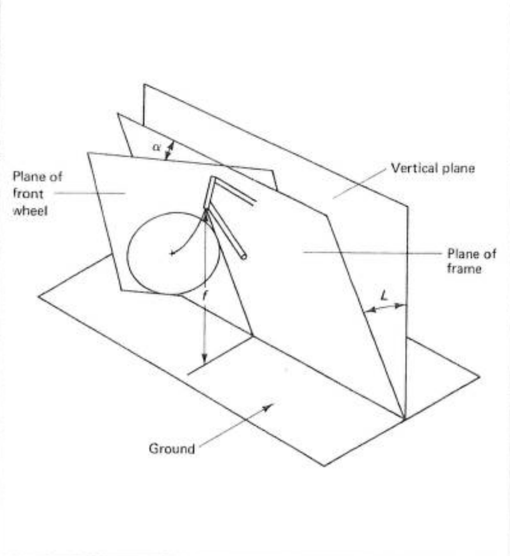


Figure 2: Steering geometry. α = steering angle; f = frame height; L = lean angle. Adapted from ref. 7.

he could rotate backward. He found that this made little difference to normal handling, and concluded that gyroscopic action has little influence on bicycle stability. He did find, however, that URB I would not travel riderless. Gyroscopic action was important for the lightweight bicycle alone, but not for the bicycle plus rider, when the rider was controlling the bicycle with the handlebars. When Jones attempted to ride URB I "no hands," he could only just maintain his seat. The bicycle seemed to lack balance and responsiveness. This confirmed Den Hartog's analysis (ref. 2).

We will not relate in equal detail Jones's several other URBs, with large, small, and reversed fork offset and with a tiny front wheel. Suffice it to say that he was able to ride all of them, although URB IV, with a very large fork offset, was unstable and very difficult to ride. URB III had reversed fork offset and therefore a very large trail, and was extremely stable. When pushed, riderless, it would steer itself for an astonishingly long time, negotiating depressions and bumps in the road and continuing until it was almost stationary before falling over. It was, however, sluggish and heavy to steer on any path other than that dictated in some way by its interaction with the roadway. (D.G.W. had an old car with some of these characteristics; every bump or hollow in the road would change the direction without any movement of the steering wheel, and to maintain a straight course required continuous anticipatory steering. A high degree of stability is not always desirable in a vehicle.)

Jones quantified a stability function after making the following observations: When the bicycle is wheeled by, for instance, holding the saddle, it is caster action that makes the front wheel go straight ahead when the bicycle is moved with the frame vertical on a horizontal roadway. The front wheel "trails" the frame. The rear wheel trails along, too. One can immediately see the importance of trail by pulling the bicycle backward. If the steering-head bearings are free, the front wheel (now in the rear) will immediately flop around to some large steering angle. This movement of the front wheel is not at first assisted by gravity. The wheel sweeps out a doughnut, and regardless of the head angle (so long as it is less than 90°) or the fork offset (so long as it is finite) the front wheel is at a point of unstable equilibrium. That is, it requires a small disturbance to make it flop over. When the frame of the bicycle is tilted, the wheel is no longer in equilibrium in the straight-ahead position. There is a force or turning moment that increases the frame tilt and acts to turn the handlebars. The equilibrium angle taken up by the handlebars is, however, an inverse function of frame tilt. In other words, with a small tilt the handlebars turn a long way, and with a large tilt the handlebars go only a few degrees from the neutral position. The reason the handlebars turn is that this allows the frame to fall; the frame and the weight carried on it seek the minimum-potential-energy position.

The computer program Jones wrote to solve the steering geometry of a single-track vehicle with a rigid body and thin wheels (no allowance was made for tire cross-section shape) produced graphs like Figure 3, which is for one steering-head angle and one fork offset (as a proportion of wheel diameter). He decided to concentrate on the steering characteristics when the steering angle was near to zero, as in normal riding. Small changes in steering angle would produce a rise or fall in the frame height *(df/dα)* (Figure 2). Jones reasoned that, for stable steering, the steering should "want" to turn into the curve as the frame leans around a bend. The reason for the steering wanting to turn had to be the fall of the frame in these circumstances. This criterion is expressed mathematically as the requirement that

$$\left(\frac{\delta^2(f/d)}{\delta\alpha\delta L}\right)_{\alpha=0}$$

be negative.* From his computer program Jones produced Figure 4, which covers all likely combinations of head angle and what he calls "front projection," identified on Figure 4. He found that the graph agreed with experience. URB IV was indeed in the unstable region, while URB III was far into the stable region. We have added some other bicycles to those Jones considered.

The information in Figure 4 can be expressed in more familiar terms as

$$(y/d) = 0.00917[(90° - H)\ (\sin H) + 4u], \quad (1)$$

where y is the fork offset, d is the wheel diameter, H is the head angle (in degrees), and u is the stability criterion

$$\left(\frac{\delta^2(f/d)}{\delta\alpha\delta L}\right)_{\alpha=0}$$

Experience indicates that bicycles have good steering characteristics when u is between -1 and -3.

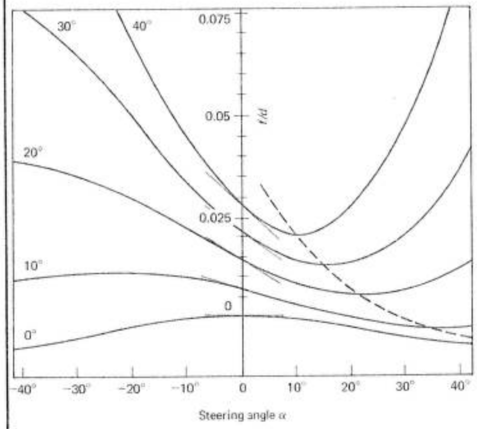


Figure 3: Typical results of stability calculations. Degree values on curves represent lean angle L; axis at middle represents relative frame height f/d; slopes indicated near middle are $[d(f/d)/d\alpha]_{\alpha=0}$. Head angle H is 70°; fork offset y/d is 0.094. Successive curves are displaced vertically for clarity.

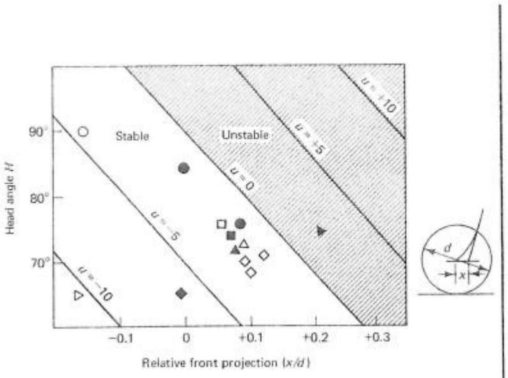


Figure 4: Jones's stability criteria. Diagonal lines represent constant stability, u = $[a^2(f/d)/a\alpha aL]_{\alpha=0}$ (0) 1887 Rudge; (•) high-wheelers; (□) modern track bike; (■) modern road bike; (△) modern tourist bike; (▲) Raleigh "chopper"; (◇) Ross child's bike; (◆) 1879 Lawson safety; (▷) URB III; (▶) URB IV. Note: Because high-wheelers differ from safeties in that the rider pedals and straddles the front wheel, the points for high-wheelers are included for interest only.

Range of Practicable Configurations for Standard Bicycles

The standard diamond-frame safety bicycle has resulted in the universal selection of steering angles from a very small range. This range has emerged from the following considerations. The crank length has generally been chosen at 170 millimeters to suit the majority of adult riders. The height of the bottom bracket above the ground has then been fixed so that the pedals clear the ground in at least low-speed cornering. The rear wheel is brought as close to the bottom bracket as can reasonably be arranged, with the seat tube angle positioning the saddle so that the rider's center of gravity is reasonably forward of the rear-wheel center even when an upright riding position is used. (Otherwise the front wheel would lift off the ground every time one attempted rapid acceleration.)

Then, in touring or commuting bicycles, the front wheel plus a possible fender or mudguard is brought as close to the bottom bracket as possible without there being a possibility of the feet or toe clips catching on the fender or fender stays during a turn.

If a large steering-head angle is used with this proviso, the top tube or crossbar becomes long, requiring a long reach to the handlebars. Accordingly, a relatively small head angle is used for touring and commuting bicycles. Racing and track bicycles do not use fenders, and the designers allow the possibility of interference between the toe clips and the tire because the skill of the rider can be relied upon to avoid it.

Therefore, a relatively large steering-head angle can be used on a racing bicycle, giving a smaller and more rigid frame. The consequence is that touring bicycles generally have head angles of 72°-73°, road racing bicycles have angles of 73°-74°, and track bicycles have angles of 74°-75°.

We have calculated the stability index u for some of the bicycles listed in reference 8 (see Table 1). It is surprising, and gratifying, to see the small range of u values used by designers. This seems to confirm the value of Jones's work. As might be expected, there is a tendency for the high-speed road racing and track machines to have u values in the more stable range (from -2.0 to -2.65) and for the touring machines to use u values from -1.85 to -2.3, which give somewhat lighter, more responsive steering but still give plenty of stability according to Jones's criterion.

The overlap between the touring and the racing machines is notable. It would seem that one could specify u = -2.0 for any type of bicycle and simply specify the head angle at 72°-73° for touring or 73°-75° for racing. Any variations of u from -2.0 would be for personal taste rather than because of any safety considerations. However, we have given a wider range of u (from -1.0 to -3.0) in Table 2, together with a range of head angle from 70° to 76°, so that the fork offset may be specified by interpolation if desired.

Equation 1 may also be used. For example, if we wish to specify the fork offset y for a track bicycle with wheels 680 millimeters in diameter and a head angle of 74°, and if we choose u = -2.25, the equation gives

$$y = 680 \times 0.00917 \; [(90 - 74) \sin 74 - 4 \times 2.25] = 39.78 \text{ mm } (1.565 \text{ in.}).$$

The trail, *t*, is also given in Table 1. It may also be calculated from the formula

$$\frac{t}{d} = \frac{1}{\sin H}\left(\frac{\cos H}{2} - \frac{y}{d}\right).$$

However, trail is a dependent variable, and not of primary importance. The framebuilder works to a fork offset, *y*, and this should be specified from the head angle *H* and the desired stability value *u*, using Table 2 or the formula upon which it was based.

Table 2: Ratio of fork offset to wheel diameter for various stability indices.

Head angle	Stability index *u* −1.0	−2.0	−3.0
70°	0.135	0.099	0.062
72°	0.120	0.083	0.047
74°	0.104	0.067	0.031
76°	0.088	0.051	0.015

Shimmy

This phenomenon is well but trivially illustrated by many small carts, such as those often used in food markets, whose castered wheels oscillate through a large angle when the cart is pushed above a certain critical speed. Shimmy is dangerous when it occurs in vehicles carrying people. When airplanes switched from having a single trailing tail wheel to a single leading nose wheel (which, of course, was mounted with a degree of trail), many lives and planes were lost when a nose-wheel would suddenly shimmy to the point where control was lost or some part of the structure failed (ref. 9).

A shimmy-type oscillation occurs in a system with mass, structural springiness, and damping if a mechanism arises to reinforce a random initial oscillation and if the damping is small. By "damping" we mean some form of frictional dissipation. A ball bouncing on the end of a spring will have some damping in the spring material and more damping in the air resistance; if the ball is lowered into a pool of water, the additional damping stops the oscillation very quickly.

Bicycle front-wheel shimmy probably happens as follows: something, a bump in the road perhaps, causes a sudden change in steering angle when the bicycle is going straight ahead. The machine is going too fast to respond by turning in the direction of steer. Rather, the inertia of the bicycle and rider carries them forward, and the caster action of the front wheel produces a very large restoring moment. Because of the mass and gyroscopic inertia of the front wheel, it does not respond exactly in phase with the restoring moment. Rather, some energy is stored in flexing of the forks, of the handlebar stem, and perhaps of the wheel itself. Most of this energy is converted to kinetic energy when the wheel passes through its neutral position, causing it to overshoot and to repeat the process. There is not much damping in this system so long as the oscillations are small. When they build up to the point where the handlebars move appreciably, much of the additional energy will be lost in the friction between the hands and the grips. There will also be increased losses in the tire "scrubbing" on the road. But the oscillations may still be large enough to cause loss of control.

Shimmy may be influenced by loads carried over the front wheel, or by looseness in various joints and bearings. There is no universal cure for shimmy. It should be helpful to increase the stiffness of all components, especially the front fork, the handlebars and stem, and perhaps especially the wheel. Spokes should be pulled up to produce a high stress. Paradoxically, this usually results in longer spoke life than if the stress is low enough to allow considerable spoke flexing. Increased stiffness will increase the natural frequency, which will increase the speed at which shimmy could occur and will reduce the amount of energy stored in the vibrations. The inherent damping (friction) in the tire-road contact and in the fork-frame-handlebar structure may then be sufficient to suppress shimmy altogether.

(Editor: The slopes $\frac{d(f/d)}{d\alpha}$ in Figure 3 (the tangent lines drawn at center)

Table 1: Steering geometries and stability indices of high-quality bicycles.

Bicycle type	Head angle	Fork-offset ratio[a]	Stability index u[b]
Touring	72°	0.0736	−2.27
	72°	0.0740	−1.99
	72°	0.0692	−1.86
	73°	0.0845	−2.28
Road-racing	73°	0.0837	−2.26
	74°	0.0729	−2.00
	74°	0.0976	−2.64
	74.5°	0.0804	−2.20
Track	75°	0.0759	−2.09
	75°	0.0953	−2.60

a. Fork offset/Wheel diameter.
b. $u = [\delta^2(f/d)/\delta\alpha\delta L]_{\alpha=0}$.

represent the rate of change of frame height due to change in steering angle. The tendency of the front wheel to turn sideways will be proportional to this rate.

$$\left(\frac{\delta^2 (f/d)}{\delta\alpha \, \delta L} \right)$$

is the rate of change in these slopes *due to increased frame tilt L, and can be seen as the progressive steepening of slope from the bottom line to the top line. The physical counterpart of this term is the increase in turning tendency with increase in frame tilt.*

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LETTERS

Through a Glass, Darkly

I was pleased to receive my first issue (Vol. 1 No. 1) of *Bike Tech* yesterday. It is good to see that some people are indeed interested in "getting the numbers right" after so many years of ideas apparently based on very little engineering data.

I was disappointed, however, to find that this issue was not the one shown in your ad on page 120 of the June 1982 *Bicycling.* Was that the pilot issue referred to by Ralph Hirsch on page 15? Since you apparently went to some effort to create that issue, why didn't you share all of that good information? I got out my magnifier to read those pages shown, but according to the table of contents, some of it is missing. Not a good way to start a new effort.

Milford S. Brown
El Cerrito, California

Editor: The pilot issue was printed for distribution within the industry to gauge reaction to Bike Tech. *We'll be printing updated versions of the better pilot issue articles over the next few months. This issue's installment is the section on ISO standards.*

Rim Squasher

I read the article on frame rigidity in the June issue of *Bike Tech* with interest. I would like to suggest another, related area to research. I operate a small bike shop and like to do custom wheelbuilding. Although tables on spoke strength exist (*Bicycling,* November 1976) I know of no tables on comparative rim strengths.

I have performed crude experiments, loading unspoked rims with weights and measuring the deflection, but I would appreciate more precise data on a greater selection of rims. In addition to testing the strength of the rims in a vertical plane, it would be interesting to test the side rigidity of the rims as well, since this seems to bear on the ability of the spoked wheel to stay true. (I suspect that this is why in my experience the Weinmann A-129 rim tends to stay true longer than the Super Champion #58, even though my tests indicate that the Super Champion rim is nearly as strong as the Weinmann in a vertical plane.)

It would also be interesting to include the weights of the rims, so that the strength-to-weight ratio could be calculated for each rim.

Bob Witter
The Cycle Clinic
Minneapolis, Minnesota

Editor: Wheels are one of the areas of testing that we plan to undertake in the coming year or two (unless someone out there has already learned what we want to know and wants to publish it). (Has anyone? Drop us a line!) Rim behavior is clearly one of the fundamental unknown variables involved, which we (or whoever) will have to deal with.

One caution: Strength and rigidity are different things. (See "Tubing Rigidity," elsewhere in this issue — it's written about frame tubing, but several of the ideas apply to rims as well.) Both qualities are important, but their benefits are different: strength enables a rim to resist the various kinds of permanent *bending (all described in Eric Hjertberg's "Too Far Gone?", also in this issue). Rigidity enables the rim to resist any bending, permanent or not; one consequence is that a rigid rim can distribute a load over a larger number of spokes, "diluting" the load and minimizing spoke fatigue. Lateral rigidity is the main defense against "potato chip" wheel collapse, but strength is the main factor that determines the likelihood of recovery from it.*

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.