Materials • Mechanics • Physiology • Engineering • Aerodynamics

BIKE TECH

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IN THIS ISSUE

AERODYNAMICS

Testing for Aerodynamic Drag: A New Method

Crispin Mount Miller

In pursuit of better testing of the Zzipper fairings and Tailwind aerodynamic panniers he makes and sells, Glen Brown has been developing a new aerodynamic drag testing technique this year. The new technique requires only the addition of one piece of equipment to his existing downhill coasting test rig.

The new method offers an accuracy of one percent (of total drag for a standard bicycle); compensates automatically for variations in slope (a major problem for most coasting tests); and uses self-contained on-board instruments whose total cost is less than $400.

The new instrument is a single-axis accelerometer. Rather than deducing the drag force from terminal speed or rates of change in speed data, Brown's method takes direct measurements of inertial effects on the bicycle and rider.

The automatic correction for slope considerably simplifies the extraction of results. In other coasting tests the slope of the road must be known exactly so that the record of speed change can be corrected for the propelling or retarding effect of gravity. But to the accelerometer the slope is irrelevant: what it measures (see box) is not simply change in speed, but deviation from pure inertial motion.

Pure inertial motion can be thought of as a one-dimensional counterpart of "free fall" (which generally requires three dimensions). An accelerometer in free fall will always read zero, and so will a single-axis accelerometer in free inertial motion along the accelerometer's axis.

Rearward Acceleration

On level ground the accelerometer reads simple change in speed after all: since pure inertial motion would be constant speed, the deviation (which happens because of drag) is the rearward acceleration that occurs as the bike coasts to a stop.

But on a downhill slope, pure inertial motion includes gravitational acceleration (as free fall does) and therefore would give ever-increasing speed. In this case, the deviation is the *loss* of acceleration as drag limits the speed to a terminal speed. In both cases, though, the accelerometer gives a reading proportional to the drag.

(To be rigorous, there is one small difference between the slowing-down case and the downhill case. The rotational inertia of the wheels makes the effective inertial mass, that determines the level-ground slowing-down reading, slightly different from the gravitational mass that determines the downhill terminal-speed reading. Since bicycle designs usually minimize wheel inertia, the resulting error is only one or two percent for the pure level-ground case, and proportionally less for situations approaching a steady terminal speed. The data can be corrected if the approximate rate of change in speed is known, but for tests run near terminal speed, as Brown now does them, the error is negligible.)

Brown uses the technique principally for comparative evaluations of the drag effects of various accessories on a standard bicycle, though he has also tested one human-powered streamliner (Jim Gentes's, sponsored by Blackburn and SunTour) with it.

Brown estimates that the new method's precision — roughly one percent of total drag, for a standard bike — is about the same as that of his previous drag-testing method (reading terminal speed, to the nearest 0.1 mph, for a bike coasting downhill). However,

FROM THE EDITOR

Bike Tech is now four issues old, and we'd like to introduce ourselves to our readers and review our short history. We began as a *Bicycling* Magazine experiment to satisfy the needs of dedicated technical enthusiasts and bicycle industry professionals with a truly technical publication. The reason this was an experiment was that we didn't know whether the world wanted such a publication.

From our reader response, we now realize that the world has wanted such a publication for quite some time. Almost 8,000 of you have subscribed, and this figure will be badly dated by the time this issue reaches your hands. Some of the most prominent bicycle companies in the world have flattered us by ordering multiple subscriptions.

At least one reviewer has been quite pleased: "The first two issues are physically thin, but the content is better than I had dared hope," wrote Jay Baldwin in the fall 1982 issue of *CoEvolution Quarterly*. "The editors hope *Bike Tech* will become an information clearinghouse, something that has been needed for a long time." (*Bike Tech* must stay at its current size unless the prices of paper and production somehow decrease.)

But we aren't yet on Easy Street: indeed we do hope to become a clearinghouse, so we'd like to reach more sources of information. The most fascinating test results come from the labs of private bicycle companies, and we haven't yet managed to persuade the larger companies to share their test data with our readers. We'll keep asking. We have some long-range plans to share testing work with university students, framebuilders, and small custom companies, and cooperatively generate articles. Perhaps such an arrangement would help your work. Send us a proposal. *John Schubert*

BIKE TECH

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the new method produces data much more rapidly than the old; and the new method can be used on hills where the old method couldn't, since the new method is less sensitive to variations in the slope of the road.

Equipment

The equipment used for the method consists of the accelerometer, a digital electronic speedometer, and a miniature cassette voice recorder. The accelerometer (with its readout and supporting circuitry) costs about $200; the speedometer, $120; and the recorder $50.

The accelerometer, an oil-damped pendulum type, is mounted low in the front frame triangle of the test bike, just above the bottom bracket, to minimize its reception of "noise" motion when a wheel hits a bump and the bicycle rocks forward or back. A damped-pendulum accelerometer does have one drawback — significant lag due to the damping — but Brown considers it hard to beat in other respects: it can resolve a slope to the nearest tenth of a degree (or a small acceleration to the nearest 0.0017 g) and it's rugged enough that he can fasten it onto a bicycle frame and not worry about it.

Brown zeroes and calibrates the instrument by setting the bicycle, with its rider (himself), on a level surface (which he checks by facing the bicycle in both directions) and then setting a block of known thickness under each wheel in turn to simulate a specific upslope and downslope.

The speedometer is a custom-built Digitach® model with a precision of 0.1 mph. This instrument's sensor "reads" the passage of every spoke, rather than the passage of spoke-mounted magnets; part of the installation procedure is to magnetize each spoke by "wiping" a magnet against it.

Both the accelerometer and the speedometer have handlebar-mounted digital displays. Brown takes data as he rides by glancing at each of these displays, first the speedometer and then the accelerometer, and reciting the number from each into the cassette recorder, which he wears strapped to his chest (under his jersey to avoid wind noise). He also records remarks about any unusual conditions at the time of each reading.

Brown emphasizes that proper sampling technique requires recording the first number that one sees at each glance, and then mathematically averaging several data points, rather than staring at the display and trying to "eyeball" an average. He takes the speedometer reading first in each pair of data numbers, and then the accelerometer reading, to minimize the error due to the lag in the accelerometer. (The lag results from both mechanical damping in the pendulum itself and electronic damping in the readout circuit. During an approximately steady acceleration, the reading trails the actual value by about two seconds.)

The compensating delay provided by the sequential reading technique is helpful, but not sufficient; Brown would like to develop a better solution. Such a solution could be either a better way to use the output of the existing instrument, or a different accelerometer that shows quicker response without sacrificing the virtues of the present one. Meanwhile, he gets around the problem, at the cost of some versatility, by taking data only when the speed varies gradually (i.e., near terminal speed down a slope) so that errors due to lag are small enough not to be a problem.

Procedure

To run a test, Brown first takes readings at a slow roll (at which air drag is negligible) to obtain a figure for tire rolling friction. This figure, designated "R," will later be subtracted from the high-speed readings to isolate the portion of drag due to air flow.

Brown then rides the bike down a selected hill with a fairly even slope. He pedals the bike to a speed close to terminal speed (learned from previous runs) and then stops pedaling and takes readings as the coasting bike approaches terminal speed.

In the simple terminal-speed tests Brown used before he had the accelerometer, each run yielded one data point — the value of the steady terminal speed — if Brown managed to pedal the bike to an initial speed near enough to the terminal speed so that the speed would stabilize (within the precision of the speedometer) in the distance available. If the bike never settled into a steady speed, the run yielded no data at all.

If the hill had a varying slope, the problem was compounded: for one run to be comparable to another, each run had to stabilize at terminal speed at the same place on the hill, so that they all occurred at the same slope. Sometimes it took Brown five runs to get one data point.

The accelerometer changes the situation dramatically by removing the restriction that the bike must reach terminal speed. If the speed is steady, the accelerometer simply reads the slope, and if it is not, the accelerometer reads a combination of slope and acceleration; but the reading is proportional to the drag in either case. (The reading can also be thought of as indicating the "equivalent slope" at which the present speed would be the steady terminal speed.)

No longer dependent on waiting for the bike to reach a steady terminal speed, Brown typically can read and record 20 data points — pairs of numbers — per run. In addition, since equilibrium is not necessary, minor fluctuations in slope are no longer a problem. (Because of the accelerometer's lag, Brown does try to be within one or two mph of equilibrium speed when he takes readings — this will keep the lag error smaller than the resolution of the instrument. But this is not too stringent a requirement: for a terminal speed of 33 mph, typical on the hill he has used lately, the slope, roughly a six percent grade, can vary either way by about one-eighth of its magnitude without changing terminal speed by more than two mph.)

Interpreting the Results

To compare the "dragginess" of different vehicles, one must express the drag in some standardized form that corrects for irrelevant variables (especially the vehicle's speed at the time of the reading). The form traditionally used is a term called coefficient of drag. This term compensates for differences due to different speed, and evaluates the magnitude of drag in proportion to the vehicle's frontal area. (Mathematically, the coefficient of drag is defined as

$$C_d = \frac{D - R}{1/2\ \rho\ V^2\ A}$$

where D is total drag; R is tire-rolling drag; ρ is air density; V is airspeed; and A is frontal area.)

In comparison tests such as Brown's, though, coefficient of drag is somewhat irrelevant. For each modification, the figure of interest is how that modification's drag compares to the drag of the "base case" bike — not how its drag compares to its own frontal area.

Accordingly, the term Brown uses for a figure of (de)merit is independent of actual frontal area. The term is "equivalent drag area," defined as

$$A_d = \frac{D - R}{1/2\ \rho\ V^2}$$

He derives this parameter from each run of numbers by taking the slope of a linear regression of (D − R) versus V^2, and dividing by $1/2\ \rho$.

Some of the results obtained so far are listed in the table. Particularly interesting are the results for front panniers. These bulky accessories might be expected to increase the drag somewhat, even if they were streamlined. Instead, the streamlined Tailwind panniers reduced the drag six percent, and even the standard Cannondale panniers (presumably not designed for air flow) appear to reduce the drag one percent.

While the Zzipper fairing had the greatest effect of the items tested, Brown finds its result — 12 percent — disappointing (previous tests have shown almost twice this reduction). He qualifies this comment, however, with the note that he was riding straight-armed with his shoulders well above the top of the fairing, where they caught the high-speed airstream that the fairing deflects.

Further Refinements

Two major factors still limit the circumstances in which the method can be used: accelerometer lag and sensitivity to wind. Brown continues to seek ways to overcome these limits, at prices comparable to that of the present equipment.

He keeps reading catalogs, but hasn't yet found a quick accelerometer that's as sensitive and durable as the one he has. Barring that, he's considering better ways to correct data from the existing instrument: perhaps a small portable chart recorder or a tape recorder that directly records the digital signal. From either of these records, he could determine not just the reading at a given instant, but also the rate of change of the reading, and thereby correct for the lag.

Wind is a bigger nuisance; a hill can usually be trusted to stay the same shape, but the air stays perfectly still only before sunrise, if then. To test air drag, as Brown points out, ground speed (the figure measured by a wheel-sensing speedometer) is only approximately what you need to know; what you really need is airspeed, and if there's any wind the two aren't the same. Because air drag varies with the square of velocity, the bike's airspeed must be accurate within 0.5 percent for the drag figure to be accurate within one percent; on a 33 mph run the results can be thrown off by a 0.2 mph breath of breeze.

Brown is thinking of adding some kind of sensor for airspeed or head wind pressure, carried on the bike to detect the same variations the bike encounters. If he can find or build one that can detect airspeed within 0.5 percent, he can replace the speedometer with it and conduct tests all day. (Better yet, if it reads head wind pressure — $1/2\ \rho\ V^2$ — directly, that's actually what he wants to know; he can plug that figure straight into his equations and forget about speed.)

Regardless of the prospect for these refinements, Brown finds the system useful in its present form, and is planning more tests with it. The list of test subjects includes more panniers, more positions of the Zzipper fairing, and combinations of fairing and panniers; a Zzipper fairing augmented with a sewn-on stretch-fabric "girdle" that reaches from one edge of the fairing, around the back of the rider, and back to the other edge, to enclose the rider (recently used by Zzipper-sponsored rider Jim Woodhead to set an IHPVA-certified San Francisco-to-Los Angeles record of 21:20:47); and some aerodynamic wheel modifications selected by Eric Hjertberg.

Results for Some Accessories

Vehicle	Equivalent Drag Area A_d (sq. ft.)
Base case — bare bicycle	4.24
Bicycle with Cannondale front panniers	4.20
Bicycle with Tailwind front panniers	4.02
Bicycle with Zzipper fairing (no panniers)	3.74

Equations of Motion

Accelerometer reading:

$\alpha = a - g \sin \phi$

where α = accelerometer reading
a = actual acceleration
g = gravitational acceleration in free fall
ϕ = slope of road (down)

If $\phi = 0$ (level road)
then $\alpha = a$

If $a = 0$ (steady speed)
then $\alpha = -g \sin \phi$

Reading for free inertial motion:

(correction terms for effect of wheel mass appear at right ends of equations)
(see text)

$$F = m\,a \qquad + m_w\,a$$

where F = net force in direction of motion
m = total true mass
m_w = apparent additional inertial mass due to rotating mass of wheels

also $F = m\,g \sin \phi$

therefore

$$a = g \sin \phi - \frac{m_w}{m + m_w}\,g \sin \phi$$

$$\alpha = a - g \sin \phi$$

$$= 0 \qquad - \frac{m_w}{m + m_w}\,g \sin \phi$$

$$\approx \qquad -0.01\ g \sin \phi$$

for most racing bicycles.

Motion in the presence ot arbitrary forces (thrust and drag):

(corrections for wheel mass at right ends)

$$F = m\,a \qquad + m_w\,a$$

and $F = T - D + m\,g \sin \phi$
where T = thrust (applied by rider)
and D = drag (applied by air and road)

therefore

$$T - D = m\,a - m\,g \sin \phi + m_w\,a$$

$$a = m\,\alpha + m_w$$

so $D = T - m\,\alpha \qquad - m_w\,a$
or if $T = 0$
$D = -m\alpha \qquad - m_w\,a$

and if also
$a \ll \alpha$
then $D \approx -m\,\alpha$

ENGINEERING ANALYSIS

The Performance of Machines and Riders on Hills

David Gordon Wilson

When I was in grammar school near Birmingham, England, as World War II ended, we had a talk from a director of one of the many bicycle companies in that city. He told us of his firm's plans for major advances in bicycle design in the bright new world we all saw ahead.

In a hushed voice which reminded us of the way we civilians were told about new military developments like radar and rockets, he told us that he was going to shorten the wheelbase. This was the clue, he assured us conspiratorially, to significant increases in speed records, particularly up hills.

His assertion puzzled me then, and it puzzles me now. I am equally intrigued by the reports of testers of new machines. They find bikes to be "responsive," "sluggish," "skittish," and so forth, through a range of performance which is rather remarkable when one considers that to the uninitiated observer the machines are identical. Can the configuration of the frame, and its stiffness in response to road and rider loads, have so dominant an influence?

The general equation relating power input to speed, weight, and drag (see box) does not provide much guidance; of the several variables in the equation, only weight *(mg)* will have any more importance on hills than on the level, and while everyone appreciates the importance of lightness, the difference between one good bike and another is usually rather small. In a lightweight bicycle of 9 kg mass, the frame will contribute about 2 kg ± 1/2 kg, and shaving much off this can lead to flexibility, and, worse, fatigue failures.

There is a term in the equation for mechanical efficiency, ηmech, but the possibility for substantial differences among new bicycles using similar chains and sprockets is really negligible.

This equation, though, is written as if the feet and drivetrain apply a steady torque to the rear wheel. It does not address any effects of the way that pedaling force varies during each crank cycle. These effects probably depend on two factors that could be very different in different machines, and that are difficult to quantify: flexibility and ergonomics.

Flexibility

Flexibility will have different effects when it is found in different areas, and when it is found with and without damping. Some flexibility in the tires, or the forks, or perhaps in a sprung frame, can reduce rolling resistance over rough surfaces. Only when the surface is extremely smooth, as it should be on a railroad line, will an inflexible wheel give the lowest rolling resistance.

I used to ride a Moulton bicycle with a spring-suspension frame. For rough surfaces it was a delight. But if I took to jumping on the pedals to get up a particularly steep hill, some of my precious energy went into compressing the springs, and, as they were well damped, this energy was dissipated. The Moulton gave one a lesson in the virtues of a smooth pedaling action.

Gravity

Springiness or flexibility in the total transmission between the rider and the driving wheel is of a rather different nature. One form of flexibility is that of gravity. Instead of pushing the pedal steadily down, the rider can push rapidly, lifting him/herself up on it, and allowing his weight to carry him to the bottom of the stroke. This technique is often used; it is looked upon with disfavor by coaches, but has no identifiable physical losses. If it is a bad technique, it is so because of ergonomic reasons, which I will discuss below.

The interesting question is whether frame and handlebar flexibility is different in kind from gravity flexibility. It seems to me that it is, if only because it reverses its direction of action, which gravity does not do.

When a powerful rider pushes strongly on the right pedal, he pulls on the right handlebar (and applies some controlling forces on the left bar and the saddle). The combination of frame and handlebar must bend to some extent. The rider will get back the energy that he has put into bending the frame only if he pauses or relaxes pedal pressure slowly enough at the end of the stroke for the frame to spring back in a controlled way. But very soon he must push on the other pedal, reversing the direction of frame bending.

At one extreme, this system could be regarded as one in which the rider pedals a rigid bike and at the same time uses a spring-type chest exerciser. There are virtually no mechanical losses in a chest exerciser, but using it is hard work, because muscles do not recover energy from springs. In fact, it takes muscle energy to let a spring relax, or to run downhill.

Again, this is an ergonomic question, and maybe one facet of good riding technique is to permit the small amount of spring energy in the frame to be fed through the chain as the legs taper off in driving force, rather than having the spring energy dissipated as "negative work" in the muscles.

I cannot answer this question, but I have no doubt that, in this respect, a more rigid frame will be better than a flexible frame, even though suitable riding techniques might enable skilled people to avoid all the losses which could be involved in bending the frame.

Major Factor: Gearing

I think, though, that the major ergonomic uncertainty in comparing one bicycle to another is not so much the losses, which in any good bicycle should be small, but instead how one bicycle allows its rider to develop more power than does another bicycle.

The major factor is undoubtedly the gearing, with saddle height or frame size and crank length having a small effect.

The best data I know about on the effect of gearing are still those given in the report of the Japanese Bicycle Production and Technical Institute, 1968 (Figure 1), which we have reproduced in *Bicycling Science* . A typical set of curves for one rider (each rider will have a unique set of curves) shows that for each gear ratio there is a pedaling speed that will give maximum ergonomic efficiency for bicycling on level ground. The maximum efficiencies are not the same for different gear

$$W = \frac{C_V}{\eta\text{mech}}\left\{\Sigma mg\left[C_R + \frac{s}{100} + \frac{a}{g}\left(1 + \frac{m_W}{\Sigma m}\right)\right] + 0.5C_DA\rho(C_V + C_W)^2\right\},$$

W = power required
C_V = ground speed
ηmech = mechanical efficiency
Σm = total mass of cycle and rider
g = acceleration of gravity
C_R = coefficient of rolling resistance of wheels
s = slope (up), in percent
a = forward acceleration
m_W = effective rotational mass of wheels
C_D = coefficient of aerodynamic drag
A = frontal area
ρ = density of air
C_W = speed of head wind

From Whitt and Wilson, Bicycling Science, *second edition (MIT Press, 1982), p. 157.*

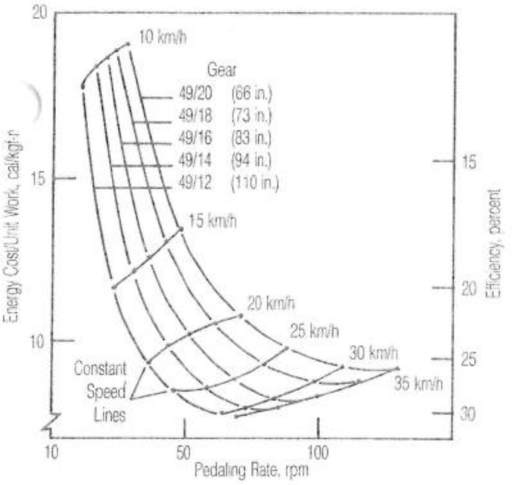


Figure 1: Effect of gearing on net energy efficiency (and, by inference, on power output). (from *Bicycling Science*, 2nd Edition, MIT Press, 1982, p. 66).

ratios, so that there is also one gear ratio which gives the overall best ergonomic efficiency.

Maximum Output Conditions

What is not shown on this graph are the conditions for maximum output power. If these were given, we should be able to plot a curve of the type shown in Figure 2 which would be the "envelope" curve for the best conditions for any one person.

And this curve would not be tied to bicycling on the level, but would be for pedaling at any power output. From it one could find out the best pedaling rate for all power levels up to the maximum that the rider could produce. This information could be applied to hill climbing as well as for pedaling on the level or against a head wind.

I cannot see why a curve of this sort would be different if obtained on one bicycle rather than another of the same general type, so long as the frame-and-handlebar rigidity was similar. Therefore I confess that I am puzzled as to why one bicycle should be better than another in hill climbing.

Large Assumption

I am making a rather large assumption. The enormous difference between bicycling and most other athletic pursuits, such as running, is that whereas some runners do relatively better on the flat while others are physically more suited to hill climbs, bicyclists have gearing systems which, for races of equal duration, should make powerful riders equally good on flat runs or on hills.

I assume, then, since gearing can allow the rider to use an optimum force and foot speed on any grade, that the principal difference between riding up a hill or on the level is, or should be, only the greatly reduced wind cooling that runners and bicyclists experience on uphills — but this is not a difference that applies to one bicycle and not to another.

So we come back to the question: what makes one bicycle better than another on hills? For machines that have been judged equally good on the flat, so that their frame-and-handlebar flexibilities are similar, I believe that there is only one answer. The difference has to be that one has more suitable gears for hills.

It is an obvious fact of bicycling life, after all, that the gears on most bicycles enable riders to work at their optimum pedaling rates only over a rather restricted range of speeds.

Riders are often very fussy about gears, using very close-ratio clusters to get exactly the best ratio for level running in various wind conditions and for gentle inclines; but when a moderate hill appears, these riders are totally unable to pedal at anywhere near their optimum rates. So they resort to non-optimum methods of delivering power, pulling on their toe straps (tests of champion bicyclists have shown that they never exert an upward pull on their toe straps when pedaling in their normal high-efficiency range) and standing up off the saddle to dance on the pedals.

Standing up to pedal can be a relief at any time to a body that has been constrained in one position, but it obviously is not an efficient way to pedal, or racers would use it all the time.

I'm closing, therefore, with a plea to exercise physiologists and to bicycle testers. Physiologists, we would like human-performance data presented in the form of Figure 2. We presume that the data have been taken, but I have not been able to find them all gathered in a form I can easily use. And testers, when you report on the performance of various bicycles, please make it clear whether you are criticizing the heart of the bicycle — the frame — or just one of the components, the gearing system.

More Questions About Hills

Although he regards gearing as more important, David Gordon Wilson makes several interesting observations about a bicycle's flexibility under the rider's pedaling forces.

I have one observation to add: whatever effect the bicycle's elastic flexibility does have on efficiency, that effect is not a constant percentage. It will be a greater percentage when the rider pushes hard on the pedals, because the spring energy stored in the frame will be proportional to the square of the force applied.

So *if* the rider pushes harder on the pedals to climb hills (instead of gearing down), frame flexibility will affect performance on hills more than in most level riding (though this is a loose distinction, since riders also push very hard in sprints on level ground).

Is this important? It depends on whether it makes sense to climb a hill that way. I don't know. But the data in Wilson's Figure 1, and the idea of Figure 2, raise some interesting questions on the subject:

Why Spin?

A given riding speed on the graph in Figure 1 will presumably correspond to a given constant power requirement. A striking feature of the graph is that for every plotted speed (or power level) the greatest efficiency occurs at the *lowest* cadence, or pedaling speed. (Efficiency is given on an inverted scale at the right side of the graph). Spinning the cranks rapidly may have virtues, but apparently efficiency isn't one of them. What are they, then?

I can think of some possibilities: spinning increases blood flow through the leg muscles; and (for many of us, at least) it helps preserve joints and tendons.

One might think that spinning would also facilitate higher power outputs. For a given foot force, it certainly does; power is the

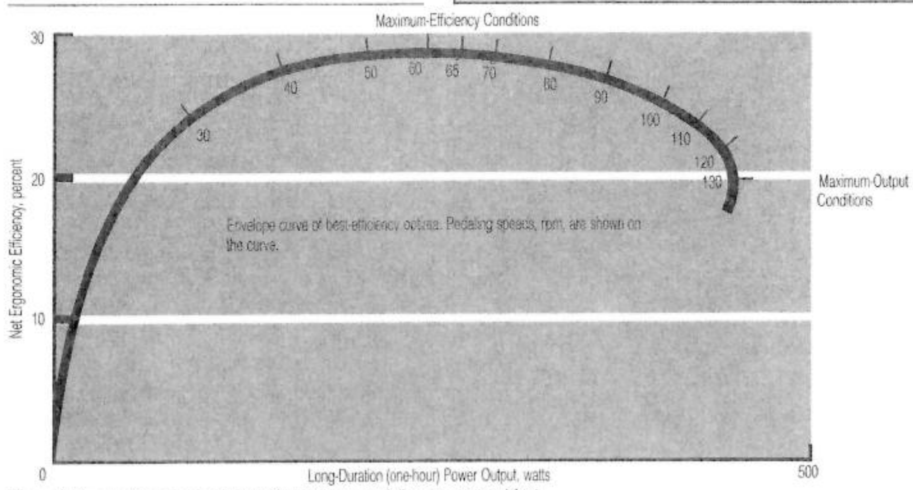


Figure 2: Hypothetical envelope curve for optimum pedaling by one subject

product of foot force and foot speed. But if one is comparing seated spinning to slower pedaling *standing up* (as many riders climb hills), the question is less clear-cut, because a standing rider usually applies much more force to the pedals. (The very highest power outputs, of course — sprints — are done by spinning *and* standing up.)

For any power output sustainable for longer than a minute or two, the rider probably can either sit or stand, so the question really is which is more efficient — which one is less work for a given net output? At moderate, long-term output levels, sitting is presumably more efficient, since, as Wilson points out, riders don't choose to ride standing up all day. But is this also true at higher outputs?

To resolve this question we need *two versions* of Figure 2: one for riders seated and one for riders standing up. How would the "standing" line compare to the "sitting" one? Certainly all the cadence graduations would be shifted to the right (since the greater foot force would place any given cadence at a higher power level); and the right end, representing all-out sprints, would be farther out than the maximum sitting-down power; but where would the "standing" line have its efficiency peak? And — most important — would any of its right end show higher efficiency than the sitting line?

At any rate, most fast riders spin on level roads — so what does happen on a hill?

Some of us install whatever cogs we have to, and keep on spinning. But many racers stay in gears in the fifties and methodically stand up to climb the hill.

They could install lower gears. Why don't they? Is it habit, from having learned to ride on bikes that didn't have them? For successful racers, that seems unlikely. Is it a tactical preference to keep the gear intervals narrow for the sake of high-speed riding on the flat, and just grin and bear the hills?

It might be because of a decision to work harder during hill climbs than on the flat. It's always important to minimize the amount of time you spend below your chosen pace, because going faster (to compensate) costs more energy than you saved by going slow. But hills are an especially emphatic case of this rule.

At level-riding speeds, because air drag is important, going slower does save some energy per mile. But gravity offers no such relief when you climb a hill; in fact, it may even cost you more energy if you spend longer straining at it, since it costs some energy just to spend time with your muscles tense.

If a "Figure 2" for standing riders would show higher efficiency than the sitting one at high power outputs (even the first Figure 2 is still hypothetical, remember) then this — the choice to work harder — would be a good reason to stand up on hills.

Or there may be no good reason, as Wilson suspects; or there may be one that neither of us has thought of. Does anyone know? *Crispin Mount Miller*

MATERIALS

The Metallurgy of Brazing, Part 3

Strength of Joints

Mario Emiliani

In the first two parts of this Series, I've concentrated on the microscopic aspects of brazing. In this part, I'll examine some properties of brazed joints from a large-scale perspective. The topic of this part is the strength of brazed joints, and the factors which affect them.

Tests to Determine Mechanical Properties

Many tests have been devised to measure the properties of materials. For all of them, there is a tradeoff between specific simulation of one particular structure (which enables accurate prediction of that structure's performance) and standardization (which enables comparison of one test with many others). Since relatively few tests have been made (or published, anyway) specifically on bicycle frame joints, this article will report on the results of standardized tests, while noting their limitations.

One of the first tests performed on a material when someone wants to know its basic mechanical properties is the tensile test. However, a tensile test subjects the test specimen to a very unrealistic loading situation: it is simply pulled from both ends until it fails. In reality, most if not all load-supporting members are subjected to a variety of stresses from many different directions; a situation called combined loading.

Combined Loading

Perhaps there is no better example of this than bicycle frames. Frames are subjected to many different types of stresses, which are very difficult to duplicate in the laboratory. While the tensile test gives useful ball-park numbers, the tensile strength, yield strength, and ductility of a material under combined loading may be much lower than in the pure tensile loading applied by the test. Safety factors are always used when designing parts under stress; this is one reason why.

Similarly, the fatigue, impact, and shear strength data of brazed joints presented here are from tests that did not accurately represent the types of stresses a bicycle frame "sees." Furthermore, the test specimens were usually not shaped like bicycle frame joints. While the tests and test specimens are idealized, however, they do give data on the specific alloys used in bicycle frame construction. Thus, they are excellent starting points, and a great deal of information can be gathered from them.

Unfortunately, there seems to be no data even vaguely relevant (to bicycles) on yield strength or ductility. I've never been able to

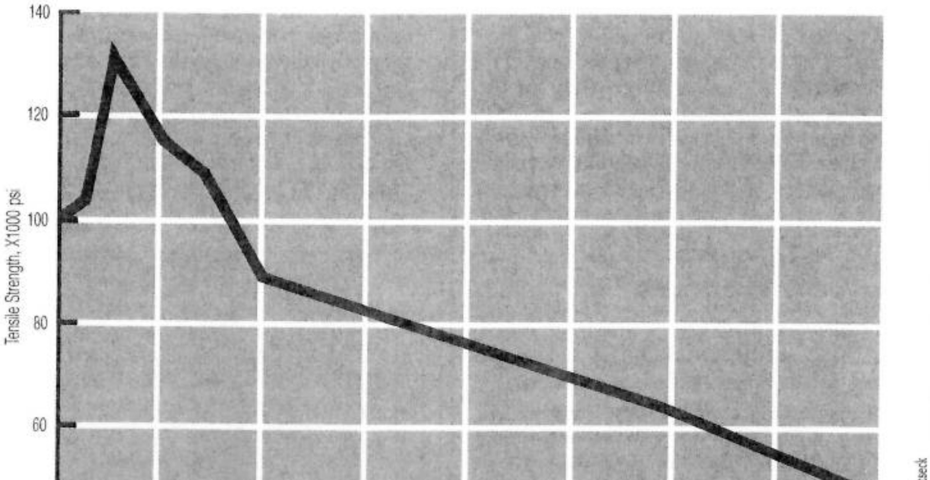


Figure 1: While a joint clearance of about 0.0015 inches produces the strongest possible joint for the conditions in which the joints were brazed, the joint is still very strong for clearances up to 0.005 inches. However, the maximum tensile strength shown here won't be the same for frame joints, since frame tubing isn't as strong as the base metal used in these tests (from C.D. Coxe and A.M. Setapen, *Welding Journal* vol. 28, no. 5, p. 462; by permission of American Welding Society *Welding Journal*).

find anything published on these properties for any brazed joints whose materials (specific alloys used) resemble those of bicycle frame joints. This is quite a pity since a lot of work was done determining the tensile strength of brazed joints, and it would have taken very little effort to collect yield strength and ductility data during those tensile tests. Thus, this discussion will have to be limited to the tensile, fatigue, impact, and shear strength of brazed joints.

Throughout this part, it's assumed that filler metals and base metals are compatible, the surfaces of the base metals are smooth, and the joint is cooled by natural convection. These are reasonable qualifications, since most frames are built under these conditions anyway, and it greatly simplifies analyzing the more relevant factors that affect joint strength.

Factors Affecting the Strength of Brazed Joints

For a given filler metal, there are five factors that determine how strong a brazed joint can be:

1. Joint clearance
2. Strength of the base metals
3. Voids in the filler metal
4. Quality of the metallic bond
5. Geometry of the joint

I include the fifth factor just to remind the designer to avoid stress concentrations in the joint. Although many potentially damaging stress raisers exist in bicycle frames, they don't normally have any effect. Thus, I won't discuss the fifth factor further.

The remaining four factors have different effects on the mechanical properties of the joint, so no broad generalizations can be made. Instead, I will discuss the four factors individually as they pertain to the tensile, fatigue, impact, and shear strengths.

Tensile Strength

To my knowledge, the relation of tensile strength to joint clearance has never been determined for steels brazed with brass filler metals. However, extensive work has been done with steels and silver brazing alloys. While this doesn't exactly mirror the ways all frames are made, it still provides useful information. Furthermore, the mechanical properties of brass-brazed joints probably parallel those of silver-brazed joints fairly closely, because in many cases the type of filler metal isn't a factor.

• Effect of joint clearance — Forty-three years ago, researchers at Handy and Harman (well-known manufacturers of brazing filler metals and fluxes) undertook research to determine the optimum joint clearance for butt-brazed specimens (a butt-brazed joint is made of two bars placed end-to-end and brazed together). This classic work appears in that company's publications *The Brazing Book* and *Brazing Technical Bulletin No. T3.* The data from this research can be plotted as a curve of tensile strength versus joint clearance (see Figure 1).

This research used torch-brazed specimens of 18-8 stainless steel (18 percent chromium, 8 percent nickel), which had a before-brazing tensile strength of 160,000 psi (and about the same after brazing). The specimens were deoxidized with a mineral flux, and the filler metal was BAg-1a.

The curve shows that optimum joint clearance is about 0.0015 inches. This is probably very close to the optimum clearance (though a different strength would result) for brass- or silver-brazed bicycle frame joints too, since the brazing procedures are the same. (The optimum clearance depends on how the joint is brazed. For example, good bonds can be achieved with smaller clearances by using gas fluxes instead of mineral fluxes.)

While it's certainly desirable to braze at the optimum joint clearance, it's just not practical when building frames. It's all but impossible to achieve uniform clearances on frame joints, so why waste the time?

I'm sure someone is thinking that it is worth the time because the joint will be as strong as possible. Well, take another look at Figure 1. You'll notice that the tensile strength of the joint is 100,000 psi or greater for clearances between 0.001-0.005 inch.

Thus, while an optimum clearance exists, it's not necessary — for all practical purposes the joint is strong enough, even with 0.005 inch clearance. So in terms of framebuilding practice, a slip-fit is all that's required to produce very strong joints.

Small and Large Clearances

Figure 1 also shows what happens to the tensile strength of the joint at very small and very large clearances. Joint clearances less than 0.001 inch reduce the tensile strength of the joint because capillary dams begin to prevail, resulting in many unbonded areas.

Beyond 0.005 inches, the tensile strength of the joint again decreases. This is due to an increase in flux inclusions, poor capillary attraction, and a general increase in defects since thick joints have a statistically greater chance of containing more defects.

As the joint thickness approaches 0.024 inches or more, the tensile strength of the joint approaches the tensile strength of the filler metal; about 50,000 psi. Thus, large buildups of filler metal at seatstay clusters, for example, are probably not much stronger than the tensile strength of the filler metal itself (all the filler metals listed in Table 1 of Part 2 of this series, published in the October issue of *Bike Tech,* have tensile strengths of about 50,000 psi in the as-cast condition).

The optimum joint clearance in Figure 1 yields a tensile strength of 135,000 psi. You may have heard that clearances beyond about 0.003 inches in silver-brazed joints result in substantial reductions in the joint's tensile strength, but the same is not true for brass-brazed joints. This is hard to believe since the tensile strengths of both filler metals are about the same. Anyway, Figure 1 shows that the tensile strength does drop off rapidly, but so what? The joint may still be strong enough.

For instance, suppose the seat tube/top tube joint is stressed to a maximum of 25,000 psi. Then the tensile strength of the joint need only be about 40,000 psi. If this were the case, then the joint clearance could be very large. Unfortunately, nobody has ever determined the stress distribution in frames, or a tensile strength versus joint clearance curve applicable to frames.

In the second part of this series, I spoke

Types of Stresses in Frame Joints

This issue's brazing article deals extensively with tensile strength of brazed joints (among other topics) and this raises an important question: why should bicycle framebuilders care about tensile strength? The answer is easily overlooked.

At first, it might appear that the joints of a lugged bicycle frame are stressed only in shear. Since the tubes are enclosed in the sleeves of the lugs, one might assume that a joint would fail only if a tube slides out of it (or twists within it), and such a failure occurs by shearing of the filler metal.

It would follow from this that the tensile strength of the brazed joint would be irrelevant, and so would be the results of fatigue and impact tests that measure strength with tensile loadings (as most of them do, although it's possible to test a specimen for fatigue or impact strength under shear loading).

However, this reasoning overlooks another type of stress that occurs in lugged joints, when they bear bending moments: when a moment begins to spread the angle that two tubes form at their intersection, the spreading of the angle tends to lift the lug away from the surface of each tube, in the region inside the angle. Thus the brazed joint receives a tensile stress.

As it happens, this type of loading is very important, because it occurs in the most heavily loaded joint in the frame: the bottom bracket. (It also occurs in the head tube joints.) Every pedal stroke tends to pull one end (or the other) of the bottom bracket shell away from its side of the seat tube. I once had a cheap handbuilt frame (with poorly bonded joints) that failed there.

Crispin Mount Miller

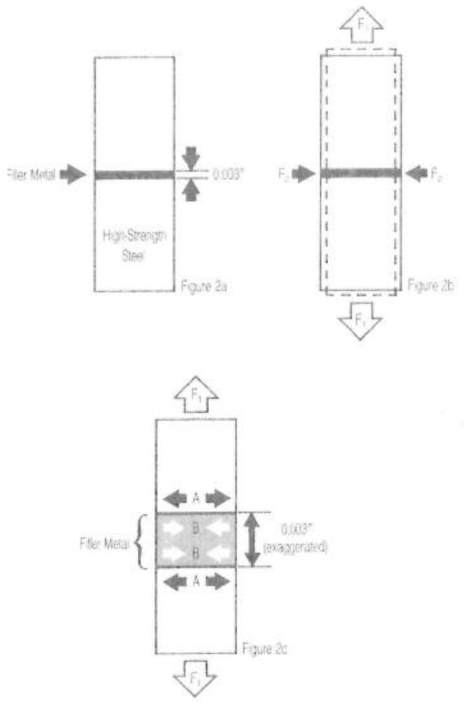


Figure 2: The tensile strength of a butt-brazed joint is two to three times as great as the tensile strength of the filler metal because the steel bars don't allow the filler metal to yield.

about the importance of having joint clearances between 0.002-0.005 inches to ensure adequate capillary attraction, and to help avoid capillary dams. This, simply put, is the overriding factor. The above clearances will virtually guarantee a strong joint (as experience has proved), without having to worry about an optimum clearance. All that's necessary is to take reasonable care when preparing the joint: degrease the tube, clean it well with sandpaper (inside the tube too, if you expect to get a fillet of filler metal in there), wipe the tube off with a clean cloth, and flux the joint prior to brazing. This will also ensure the formation of good metallic bonds.

• Strength of the base metals — A very important factor in determining the tensile strength of the brazed joint is the tensile strength of the base metal. The tests used to produce Figure 1 utilized a stainless steel with a tensile strength of 160,000 psi before and after brazing. With all other things constant (brazing time, brazing temperature, joint clearance, etc.), the tensile strength of the joint increases as the tensile strength of the base metals increases.

Since steels like Reynolds 531, Tange Champion, Columbus SL, etc. have lower tensile strengths (on the order of 110,000 psi before brazing, and roughly the same after brazing), the tensile strength of their brazed joints will be less, at any given clearance, than that shown in Figure 1. Just how much less is hard to say, since no tests have been performed.

As I've shown in the September/October 1981 issue of *Bicycling,*[1] the tensile strength of bicycle steels varies considerably depending on what temperature they are exposed to during brazing. Obviously, this too will affect the tensile strength of the joint.

By now you may be wondering why the tensile strength of brazed joints can be 2-3 times as great as the tensile strength of the filler metal. The reason lies in how the stress is distributed in and around the joint.

Figure 2a shows two high-strength circular steel bars that have been butt-brazed together with a compatible filler metal. The thickness of the joint is small, say 0.003 inches. When the rod is pulled in tension by the force F_1, it must elongate. This also forces the diameter of the rod to decrease proportionally. Thus the forces F_2 act radially inward to make the rod thinner, as shown by the dotted lines in Figure 2b.

Soon the rod is stressed to the yield point of the filler metal, but the steel bars aren't near their yield point yet. And since the steel bars are so close together, they simply prohibit the filler metal from contracting enough to yield. Figure 2c shows the situation: the forces which resist F_2 are greater at "A" than at "B", so the filler metal isn't allowed to yield just yet. As the stress increases, the forces at "A" and "B" become equal, and the filler metal begins to yield. Eventually the joint fails, usually near the yield strength of the base metals.

0.001 Inch

The thinner the joint clearance, the greater the ability of the steel bars to prohibit yielding of the filler metal. Theoretically, as the joint clearance becomes infinitely thin, the tensile strength of the joint approaches the tensile strength of the bars. But as we all know, clearances less than about 0.001 inches result in joints with many voids. Thus, the tensile strength of the bars can't ever be reached (at least by the methods used to build bicycle frames).

Conversely, as the joint clearance gets large, the steel bars aren't as effective at restraining the filler metal. Eventually the steel bars won't have any effect, and the joint will fail at the tensile strength of the filler metal.

• Voids — Small quantities of flux, called flux inclusions, can be trapped in the filler metal upon solidification. Structurally they are the equivalent of voids, and are obviously detrimental because they disrupt the continuity of the filler metal, and can reduce the area bonded.

To a degree, flux inclusions are unavoidable. However, their occurrence can be minimized by brazing within the temperature ranges given in Table 1 in Part 2. This ensures that all the flux is molten and can be readily displaced by the molten brazing alloy.

Flux within the joint that has been saturated with oxides is more difficult to displace, so keeping the brazing time to a minimum will help. Another trick is to make the filler metal flow in the direction of gravity. Since frame joints contain a range of clearances, the filler metal can't possibly flow uniformly. Gravity will help smooth out the flow, and hence help the filler metal displace the molten flux more effectively.

Figure 3 shows porosity in a frame joint caused by overheating the filler metal. This too will reduce the bonded area, as well as produce stress raisers within the filler metal. Unlike flux inclusions, the porosity shown in Figure 3 is avoidable, but it is not uncommon.

• Quality of the bond — It's no surprise that if the base metals aren't properly cleaned and fluxed, the metallic bond isn't going to be good. This will affect not only the tensile strength, but every other mechanical property as well; and it will affect them long before any of the other three factors becomes relevant. In this part, it's assumed the joints are properly prepared, so that the bond has little or no effect (i.e., the joint fails midway between the metals joined).

With this assumption, there are only three things left which might damage the metallic bond: flux inclusions, porosity caused by overheating the filler metal, and de-wetting of the flux (i.e., failure of the flux to stay on all parts of the surface). The third factor is a common occurrence which can be avoided by simply buying a better flux, or by applying more flux.

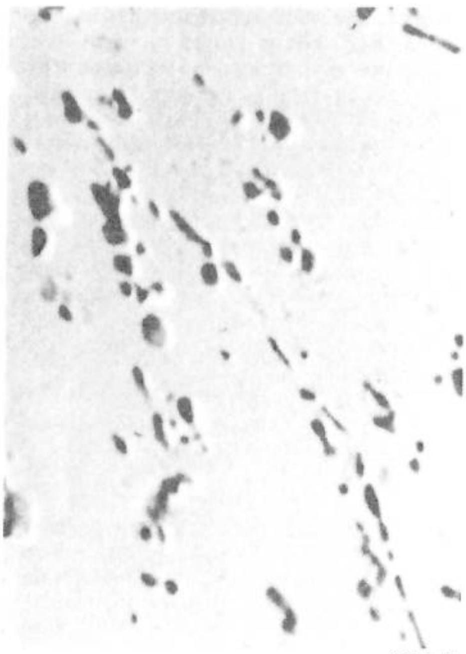
Flux inclusions are usually unavoidable, so some of them are bound to lie at the filler metal/base metal interface. This, of course, will impair the quality of the bond by reducing the area bonded. Porosity has the same deleterious effect.

Fatigue Strength

The fatigue strength of bicycle frames has long been sought-after information. But, sad to say, nobody has ever produced any data that's worthwhile because everyone's tests have been so unrealistic. The loads a frame "sees" are very complicated, and aren't even roughly simulated by standard fatigue test equipment. This being the case, we'll take a look at some simplified tests performed on butt-brazed specimens.

• Effect of joint clearance — Figure 4 shows graphs of applied stress (in bending only) versus the number of cycles (of applied stress) to failure, commonly called S-N curves. In Figures 4a and 4c, the base metal is AISI 1020 steel, while the base metal is AISI 4140 steel in the remaining two figures. The before-brazing tensile strengths of the 1020 and 4140 steels are 64,000 psi and 164,000 psi respectively. Both these steels

[1] *Mario Emiliani, "Reynolds vs. Columbus vs. the Framebuilder's Torch,"* Bicycling, *September/October 1981, pp. 92-97.*



Mario Emiliani

Figure 3: This photo is from the top tube/head tube joint of a well-known Italian racing frame. The porosity in the joint was caused by overheating the filler metal, an (R)BCuZn-type. If enough of these voids are present, the mechanical properties of the joint will be impaired.

are similar or identical to steels used to make bicycle frames.

The joint clearance is 0.001 inches in Figures 4a and 4b, and 0.010 inches in Figures 4c and 4d. The base metals were deoxidized with mineral flux and oxy-acetylene torch-brazed using BAg-1. Thus the materials and procedure used to braze the specimens are the same as those used in constructing many frames.

Notice that in all the figures, the curves become parallel to the "cycles" axis at about 22,000 psi (while they vary from this value by 10 or 15 percent, in the context of fatigue tests these variations are very small). This stress is called the fatigue limit, and it means that the joint can survive stresses at or below this level for an infinite number of cycles. So it appears that within the range of clearances tested the joint clearance has little or no effect on the fatigue limit.

Joint clearance does appear to have a minor effect on fatigue strength.[2] The fatigue strength at one million cycles in Figures 4a and 4b is about 32,000 psi, while the corresponding fatigue strength in Figures 4c and 4d is about 26,000 psi. But it's hard to say whether the difference in fatigue strength is statistically significant, since only nine tests were performed per curve. Furthermore, the joint clearance in Figures 4c and 4d is twice as large as the maximum recommended by the American Welding Society (AWS); if the joint clearance were 0.005 inches, there probably wouldn't be a statistically significant difference. So the bottom line is that if the joint clearance is within that recommended by the AWS, the joints should have close to the maximum fatigue strength.

• Strength of the base metals — The difference in before-brazing tensile strengths of the base metals is quite large, but the four curves all give practically the same fatigue limit. Thus, the before- and after-brazing tensile strength of the base metal doesn't significantly affect the fatigue limit (as long as it's higher than about 21,000 psi).

The fatigue strength does not seem to depend on the strength of the base metals either, because Figure 4a is very similar to Figure 4b, and Figure 4c is very similar to Figure 4d.

• Voids and quality of the bond — The soundness of the joint is the single most important factor determining its fatigue strength. The fewer the voids, the greater the fatigue strength.

Defects such as porosity (Figure 3) and flux inclusions act as stress raisers, which can locally magnify stresses to well beyond the yield strength of the filler metal. Soon cracks begin to form, and because they too act as stress raisers, the cracks continue to grow. Eventually the joint will fail.

It's very difficult if not impossible to produce joints absolutely free of voids by the methods used to join frames. Even so, I've never seen a brazed joint fail by fatigue. Perhaps there is an inherently large safety factor which helps the joints tolerate voids.

Fatigue failures outside the joint do occur, however. These failures usually occur in the heat-affected zone, and are invariably next to a lug point or other obvious stress raiser. While realistic fatigue data would be nice to have, this problem could be avoided through better construction techniques and/or design: thin the lug tips in critical areas, use a different lug design, or use a heavier gauge tube.

Impact Strength

Bicycle frames are often subjected to large

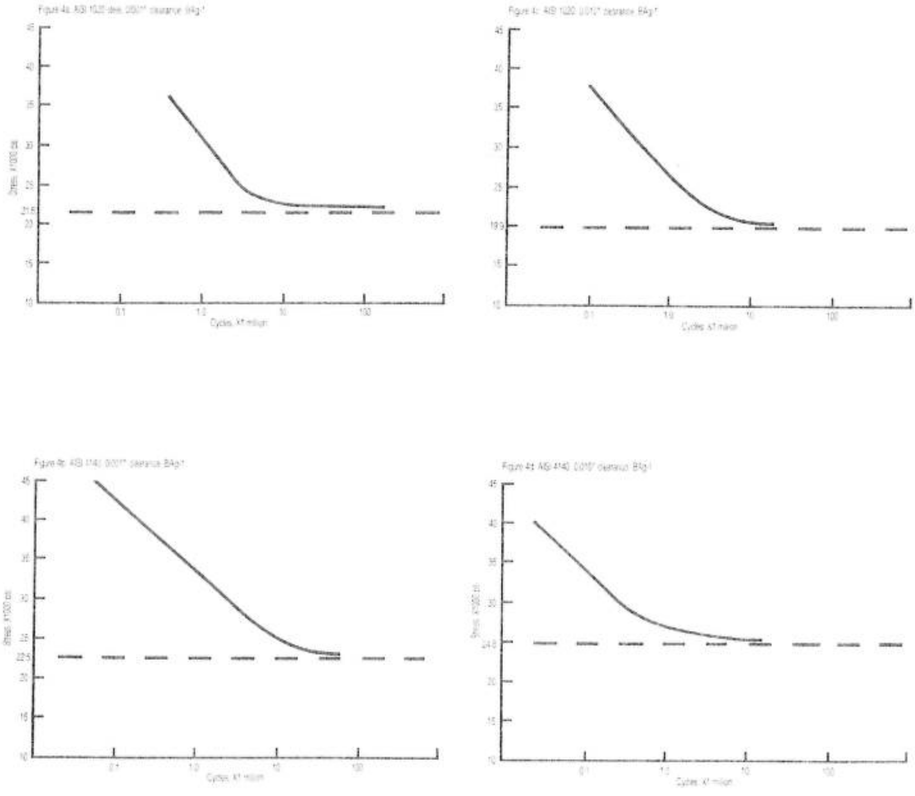


Figure 4: Both the fatigue limit and fatigue strength are not significantly affected by the tensile strength of the base metals. However, the fatigue strength is influenced by the joint clearance, while the fatigue limit isn't. Voids will reduce the fatigue strength of the joint because they act as stress raisers (from C.H. Chatfield and S. Tour, *Welding Journal* vol. 37, no. 1, pp. 37s-40s; by permission of American Welding Society *Welding Journal*).

[2]*The stress which a specimen can sustain for a given number of loading cycles; the number of cycles must be specified.*

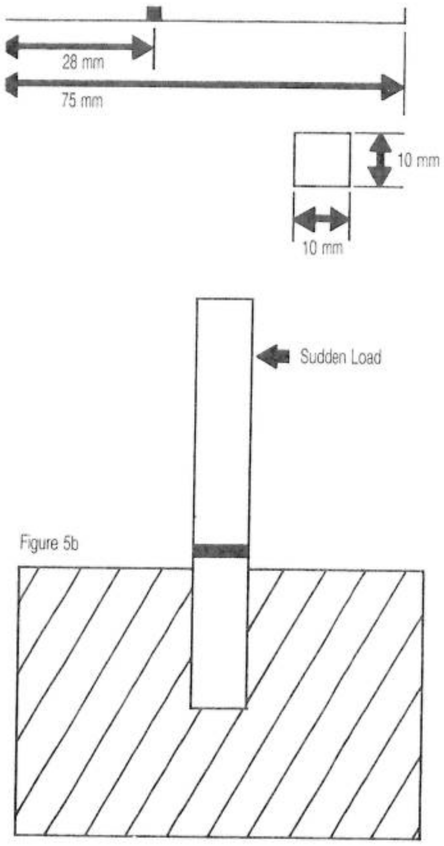


Figure 5: A standard impact specimen is made by butt-brazing two steel bars with dimensions shown in Figure 5a. In Figure 5b the short end of the specimen is held rigidly while a sudden load is applied, usually by a massive swinging pendulum (from "Tentative Methods for Notched-Bar Impact Testing of Metallic Materials," ASTM Designation E23-56T).

forces for very brief periods; riding over large bumps and potholes, for example. These types of forces are called impact loads, and the impact load needed to break a material is called its impact strength. Values of impact strength are usually given in foot-pounds, and indicate the amount of energy that a test specimen of the material can absorb before it breaks.

The impact strength of a material varies depending on the geometry of the test specimen. To ensure that everyone's impact data are comparable, there are standard specifications for test specimen geometries. Standard impact test specimens are notched so that the specimen is guaranteed to fail (in other words, the notch tests a material's ability to withstand the effects of a defect under impact loading). Brazed-joint specimens, however, need not be notched, since the filler metal acts as a weak spot. The data presented here are for un-notched brazed specimens. Figure 5 shows what a brazed impact specimen looks like, and how it's loaded to failure. As a rule of thumb, the minimum acceptable impact strength is about 15 foot-pounds for test specimens of joint types being considered for brazed frameworks.

• Effect of joint clearance — The effect of joint clearance has been investigated, but so few data points were taken that it's not worth discussing. Besides, the following factors have been shown to be more critical.

• Strength of the base metals — A pair of impact tests, with base metals of high-strength steel and soft iron, showed a dramatic difference in a way one might not expect.

The tests were performed on butt-brazed specimens of AISI 4140 steel, whose before-brazing tensile strength was approximately 150,000 psi and ductility approximately 10 percent, and Armco Iron (iron with a maximum of 0.02 percent carbon), whose respective properties were approximately 50,000 psi and 40 percent. The filler metal was BAg-1a, and the joint clearance was 0.002 inches.

The results were that the joints brazed in 4140 steel absorbed an average of 1.7 foot-pounds of energy, while the joints brazed in iron had an average impact strength of 39.9 foot-pounds.[3]

The joints brazed in iron absorbed 23 times more energy because the iron's tensile strength was low, close to that of the filler metal. Thus the filler metal was strong enough to make the iron permanently deform a lot — and large plastic deformation is the name of the game in impact testing because (for a given yield strength) the more a material deforms, the greater its impact strength. The joint in steel, however, had a very low impact strength because the steel was much stronger than the filler metal, and as a result didn't deform much.

Let's assume that the after-brazing tensile strength of frame tubes is 95,000 psi. This is much higher than the tensile strength of any filler metal, so you'd expect the impact strength of frame joints to be fairly low. But most frame joints have large laps (i.e., the portion of the lug which overlaps the tube). These tests didn't take this into account; but another test did.

AISI 4140 steel bars and rings brazed together with BAg-1a, having a radial joint clearance of 0.002 inches, a lap of 0.375 inches, and a bond area of 0.474 square inches, produced average tensile impact strengths of 88 foot-pounds.[4] Since most frame joints have considerably more lap and bonded area than this (a long-point Prugnat seat lug without cutouts provides about 1.75 square inches of bond area to the top tube, for example), their impact strengths must be quite high. This is proved by experience, since we've all ridden over big bumps without any problem (sometimes to our surprise). It's probably the lap on frame joints which produces the "inherently large safety factor" so important to this and other mechanical properties.

• Voids and quality of the bond — Once again, these factors prove to be very important. Voids have a pronounced effect during impact loading because the rate of stressing is so high. In a given length of time, cracks will travel much farther in an impact test than in a tensile test.

Shear Strength

When a bicycle hits a bump, the top tube/head tube joint is stressed in tensile shear (i.e., the top tube wants to pull out of the lug). During a sprint, the down tube "feels" torsional (twisting) shear stresses. Both joints, of course, are under the influence of other types of stresses as well.

Shear tests in tension produce different results from shear tests in torsion, because the mode of deformation in and around the joint is different. Nobody has ever done comprehensive tests using just one type of shear. Consequently, the data given below is a mix of the two.

• Effect of joint clearance — Figure 6 shows plots of shear strength versus joint clearance for circular steel shafts brazed 0.78 inches into steel rings. It's not known whether the specimens were tested in torsional shear or tensile shear. The before-brazing tensile strength of the steel was about 64,000 psi, and the filler metals used are shown on the curves. The brazing procedure is also not specified.

The curves show that the shear strength of the joint depends upon the filler metal, and there is an optimum joint clearance (which I hope you're not too concerned with). These curves are similar to Figure 1 in that there is a drop in strength at either end of the curves.

The shear strengths for RBCuZn-D are quite a bit higher than those for the other two filler metals, possibly because nickel, copper, zinc, and iron form an intermetallic at the interface. This strong intermetallic may affect the stress distribution in a way similar to how the steel bars prohibit yielding of the filler metal in a tensile test. But don't get too excited by what the curve shows; since the brazing procedure wasn't specified, the same high shear strengths may not exist in frame joints.

Since most frame joints have laps, the effect of joint area must also be considered. The joint area influences the shear strength of the joint in a way similar to Figure 6: when the joint area is small, about 0.5 square inches, the shear strength of the joint is

[3] *C.D. Coxe and A.M. Setapen,* Welding Journal, *Vol. 28, No. 5, pp. 462-466.*

[4] *H.A. Smith and P.A. Koerner,* Welding Journal, *Vol. 25, No. 3, p. 190-s.*

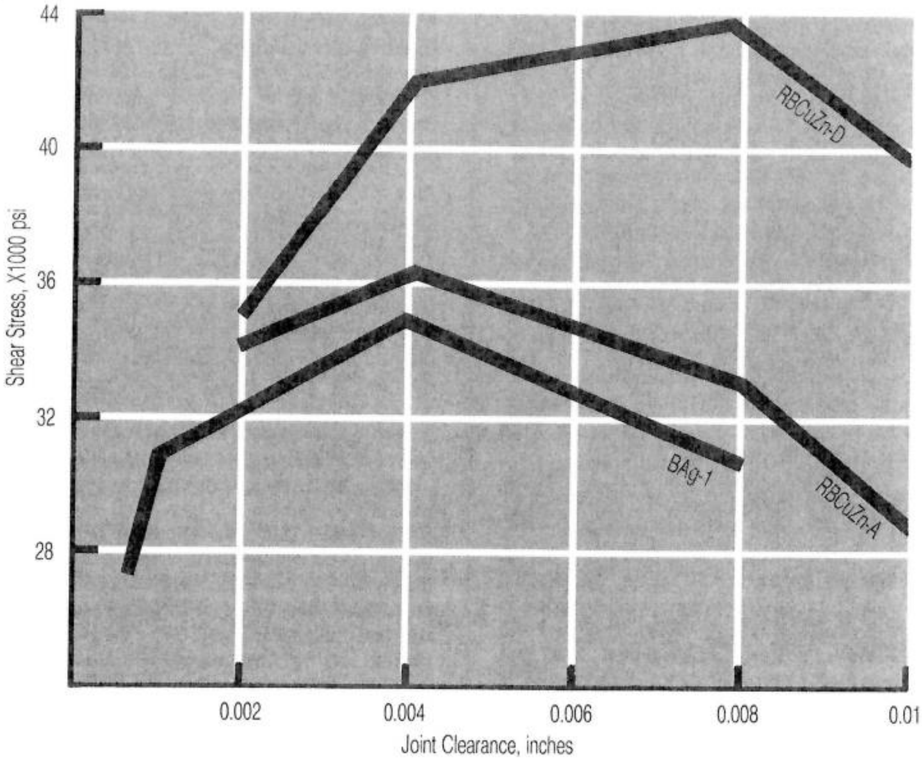


Figure 6: The maximum shear strength occurs at clearances considerably greater than the maximum tensile strength in Figure 1; yet another reason why it's not worthwhile to try and achieve a uniform clearance in frame joints (from J. Colbus et al., *Welding Journal* vol. 41, no. 9, p. 415s; by permission of American Welding Society *Welding Journal*).

around 40,000 psi. As the joint area gets larger, to around two square inches, the shear strength is about 28,000 psi.[5]

When the lap, and consequently the joint area is small, the joint quality is usually very good. Hence the joint strength is high. But as the lap increases, the shear strength of the joint (per unit area) drops partly because the quality of the bond is reduced. The other factor which makes longer laps weaker (for their size) is that beyond a certain length, the stress isn't distributed uniformly along the lap. The shear stress becomes concentrated at the ends of the lap, while the center-portion of the lap doesn't support any stress. Figure 7 shows this situation.

In practice, joint failures due to shear stresses only are extremely rare on lugged frames, although shear stresses are probably a contributing factor in fatigue failures. In any event, it's not necessary to modify current framebuilding practices to improve the shear strength of joints.

[5]*J. Colbus, et al.,* Welding Journal, *Vol. 41, No. 9, p. 415-s.*

• Strength of the base metals — The shear strength of brazed joints is much less dependent on the tensile strength of the base metal when tested in torsion than when tested in a tensile test. Armco Iron and AISI 4140 steel (having tensile strengths of 50,000 psi and 135,000 psi respectively) brazed with BAg-1a produced shear strengths of 36,000 and 43,000 psi respectively.[6] This is not a big difference. Thus, while the after-brazing tensile strength of the tube may vary considerably, the shear strength of the joint (in torsion) doesn't.

• Voids and quality of the bond — There's nothing left to say about this that wouldn't be redundant, except that bicycle framebuilders should use paste fluxes. Paste fluxes minimize voids and improve the bond quality because they provide better coverage of the metal when molten. Furthermore, paste flux isn't blown away by the flame as easily as are powdered fluxes prior to becoming liquid.

[6]*See footnote 3.*

Summary

By now I'm sure you're aware that voids and bond quality are the key factors in determining the mechanical properties of brazed joints. Thus, the framebuilder should always remember these requirements for good joints: use a good paste flux, prepare the joint properly, maintain joint clearances between about 0.002 and 0.005 inches, and employ good brazing technique (i.e., don't overheat the filler metal or spend too long making the joint, etc.).

In the final part of this series, I'll discuss what happens to the tubing during brazing, why tubing manufacturers recommend specific temperature ranges, and why many framebuilders choose to ignore this advice.

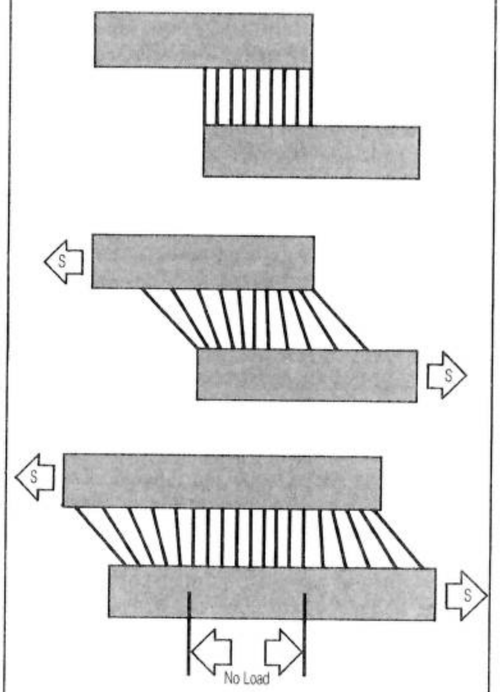


Figure 7: (Top) The parallel lines between the plates represent the filler metal in an unstressed state. (Center) When a shear stress S is applied to a lap whose joint area is small, the stress is distributed evenly throughout the joint. (Bottom) When the joint area increases, a portion of the lap in the center doesn't carry any load. Thus, lengthening the lap doesn't guarantee a stronger joint simply because the bond area is greater. (from H.R. Brooker and E.V. Beatson, *Industrial Brazing,* Butterworth Group, Seven Oaks, England, 1975).

STRUCTURES

The Front Quadrilateral Behaves as a Triangle!

John S. Allen

It's an overwhelmingly common habit to look at a bicycle frame and assume that its front quadrilateral must endure constant bending stress, due to its deviation from the more structurally sound triangle shape.

Surprisingly, this isn't true. A close look at frame geometry will show why.

It is well known that a load on any of the vertices of a triangularly braced structure produces only tension and compression forces in the structure's members. One example is the rear triangle of a bicycle. Since the seatstays and chainstays meet at a point very near the rear hub axle, the forces on them due to rider weight and chain tension are almost purely tension and compression.

That is why the seatstays and chainstays can be relatively thin compared to the other tubes of the bicycle frame. The seatstays and chainstays do have to resist some bending and torsional loads — the tension on the chain pulls them sideways somewhat, as do the forces from leaning the bike when riding out of the saddle.

A hub gear or hub brake also applies bending force by trying to rotate the bicycle's rear triangle around the rear hub. However, the largest force on the rear triangle under ordinary conditions is in compression and tension.

It is less well known that the front "triangle" of a bicycle also is stressed almost purely in compression and tension under ordinary riding conditions. The front "triangle" is actually a four-sided shape: the top tube and down tube do not meet, but are attached to the head tube at two separate places one above the other. Intuitively, one would think that large forces would tend to flex the joints that connect the head tube to the top tube and down tube.

But in fact, the forces pushing the lower end of the head tube back and forth are small, because of a pecularity of bicycle frame geometry. If the centerlines of the down tube and top tube were extended forward to meet at a point, this point would lie almost directly above the front axle.

This design feature is no accident; its advantages of strengthening the frame can be discovered either by mathematical analysis or by framebuilders' trial and error. Which came first is hard to tell, buried in history's record. Archibald Sharp does describe it mathematically in his 1896 book *Bicycles and Tricycles;* but its use preceded the book, going at least as far back as the development of the diamond-frame bicycle around 1890.

Looking at the drawing, we can see how the geometry of the front "triangle" determines the way the down tube and top tube carry the load imposed on the fork. Assume that the front wheel carries a load of 100 pounds, the down tube is at an angle of 45 degrees from horizontal, the top tube is horizontal, and that the two tubes, if extended, would meet directly over the front axle.

Basic physics tells us that a force points in a certain direction, along a line. Any equal force anywhere on the same line and pointing in the same direction will have the same effect. Point F at the intersection of the top tube and down tube centerlines is on the line of the actual force at point E, the axle. Therefore, we can substitute a force at point F for the one at point E.

The top tube and down tube are both much stiffer against tension and compression than against bending. For purposes of our discussion, then, let us pretend that they are hinged at their joints with the head tube and seat tube.

No Bending Force

If we were to attach a structure to the head tube to transmit the force from point F, we would see that this force would pull the down tube in pure tension and push the top tube in pure compression. There would be no bending force trying to fold up either hinge. In reality, the hinge in compression between the top tube and head tube would be unstable and tend to fold up — but there would be no bending force on it if it were held perfectly straight.

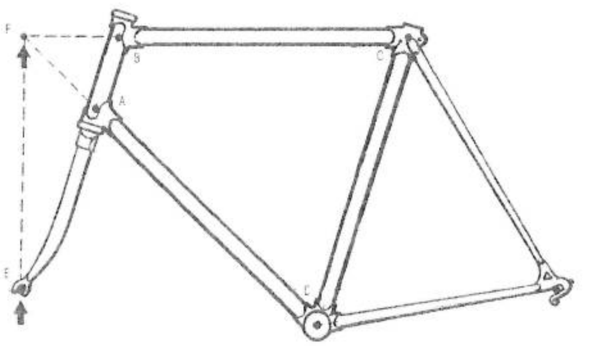
Now — remember that the force at point F is equivalent to the force at point E. This means we can move the force back to E, and replace the structure reaching to F with one that reaches to E, without changing the forces felt by the frame. Conveniently enough, a structure reaching to E already exists — it's the fork. Thus the fork loads the frame in the same way that our structure reaching to F would. *The forces on the top tube and down tube, then, are parallel to the tubes.*

Furthermore, then, ***the down tube must support the entire weight load,*** since the top tube is horizontal. With the down tube at 45 degrees from horizontal, as in our example, a 100-pound weight load on the down tube requires that there be a 100-pound horizontal component as well, in order for the total force to be parallel to the tube. The two forces are added according to the Pythagorean theorem (vector addition) to make a total tension of 141 pounds. Compression on the top tube is 100 pounds, balancing the horizontal component of the force on the down tube.

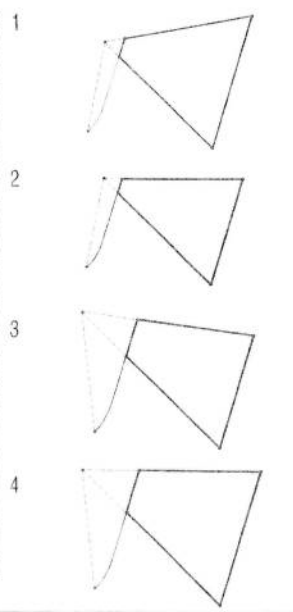
One interesting conclusion arises from this analysis: most of the flexibility of a bicycle frame against front wheel road shock is in the fork and handlebars, not in the frame. The fork absorbs shock in bending, but the frame does not do this as much, as it is in effect triangulated.

Also, relatively small variations in the dimensions of the front quadrilateral can have a major effect on its stiffness under weight load and road shock. If a frame has a horizontal top tube, the intersection of the top tube and down tube will be ahead of the front axle in a tall frame and behind the axle in a short frame. Further variations can be introduced by using a sloping top tube.

When the front wheel rides up over a bump, the fork blades flex *forward* into the bump. To some degree, this increases the harshness of the bump. The bump, and the



"Ideal" frame shape: point F directly above point E.



Variations in Frame Shape

1,2: Downward top tube slope or shorter frame: harsher resistance to road impacts, greater strength against head-on collisions.

3,4: Upward top tube slope or taller frame: smoother ride, but more vulnerability to head-on impacts.

flexing of the fork, tend to push the bike's frame backward as well as upward. In principle, a frame with the intersection of the top tube and down tube centerlines behind the front axle will give a harsher ride than one with this intersection ahead of the axle. In the former case the frame will resist the backward force from the fork, and in the latter case it will flex more easily.

This hypothesis tends to be confirmed by the harsher rider of small frames, and the behavior of the few frames that are built with sloping top tubes. Polo bike frames, frames for motorpaced racing, and the newest generation of European track racing frames have a top tube that slopes downwards toward the head tube. These frames are intended to be stiff. Often, the downwards-sloping top tube is used in conjunction with a front wheel smaller than the rear wheel.

Unusually Comfortable

Some frames built for small riders, and some custom touring frames, have a top tube that slopes upwards toward the head tube. Contributing Editor Fred DeLong owns such a frame, and reports that it is unusually comfortable.

However, such a frame is more likely to be damaged by a head-on impact. The triangulated strength and stiffness of the frame apply only to vertical forces — forces that point straight up from the axle, and more or less toward the intersection of the top and down tubes. Horizontal forces, the kind you get when you run into a tree, are borne almost entirely as bending by the top and down tubes. That's why a bike frame is vulnerable to a head-on impact, but is almost never damaged when the front wheel lands hard on the pavement, although the fork may be damaged.

Braking forces are an interesting case. Rear wheel braking — in any case, not very powerful — increases the weight loading on the front wheel but puts no horizontal force on the fork. Front wheel braking increases the weight loading and does produce a horizontal rearward force on the fork. But note that the rider resists this force mostly through the handlebars! There is actually little horizontal loading on the frame, except in a few unusual cases: a tandem, in which the rear rider's deceleration is transferred through the frame; a cargo tricycle; or a bicycle pulling a heavy trailer.

Although it is possible to draw a few conclusions from the analysis in this article, the behavior of the front quadrilateral deserves further study. We at *Bike Tech* plan to investigate it further using ride tests as well as our new frame testing machine, and we welcome contributions from our readers. We would be especially pleased to see a computer simulation of the behavior of the bicycle frame over bumps of varying steepness and height for various loads and with differing frame dimensions.

SHOP TALK

Internal Hub Gear Interchangeability Tricks

Here are some tricks that can get internally-geared hubs repaired when the "right" parts aren't available. These tricks can avoid the need to replace a hub or an entire wheel. Often you can save a wheel with an obsolete hub by installing the internals of a newer-model hub. And there is wider interchangeability of parts inside hubs than you might imagine.

Shimano cartridge-type coaster brake three-speed hubs will fit into the shells of the older Shimano coaster brake three-speed hubs. (The cartridge-type hubs are those which can be removed from the shell without using a special tool to unscrew the right ball cup.) The only problem in putting the new hub in the old shell is that the left dustcap is different.

On the older hub, the left dustcap is a separate part. On the new model, the left dustcap is part of a pressed-together assembly with the brake arm and brake cone. You can carefully separate the parts of this assembly and install an old-type dustcap in place of the new one; then assemble the cartridge-type hub into the older shell.

The new cartridge-type Shimano coaster brake will fit even more easily into the old-type Shimano coaster-brake shell. In this case, there are no special tricks needed.

The Sturmey-Archer S3C Tricoaster hub will fit into the shell of the earlier TCW Mark III, and vice versa. The appearance of the ratchet inside the shell for the left pawls is different: the TCW shell has slots and the S3C has neater-looking ramped cutouts. But the basic dimensioning is the same. The only Sturmey-Archer three-speed coaster hub that cannot be salvaged by using newer internals is the very early TCW, which had a ball ring with 24 loose balls on the left side. The more recent TCW Mark III and the S3C use a large-diameter ball retainer on the left side, with 18 balls.

Narrower Brake Band

Many parts are interchangeable between the S3C and the TCW Mark III, but be careful of those that are not. Especially, note that the brass brake band (brake shoe) for the S3C is very slightly narrower, and that the S3C uses a special, extra-wide right ball cup. Failure to observe these cautions will result in a hub which will not go together right.

It has recently become difficult to obtain parts for Sturmey-Archur hubs. Mechanics would do well to keep a stock of older Sturmey-Archer hubs and parts until Sturmey-Archer gets its act together. Also, note that SunTour and NK in Japan make hubs that are identical copies of the Sturmey-Archer AW three-speed. It is worth investigating these companies as sources of parts.

Sturmey-Archer FW four-speed and S5 five-speed hubs are of essentially the same construction. The FW is easily upgraded to a five-speed, if you can get the parts. Since the FW is no longer made, this is the best way to keep an FW in service. Note that the larger of the two axle gears of the FW has its lugs (the parts that engage the axle to keep the gear from turning) ramped on one side. This gear must be replaced with one whose lugs are square on both sides for use in a five-speed hub.

Three Little Pigs

The five-speed hub has gone through three versions. The earliest, marked S5, used a bellcrank and pushrod at the left end of the axle. Three models of bellcranks were made, of varying durability like the homes of the three little pigs: one bellcrank of plastic, one of stamped steel, and a final version of machined steel. The machined steel bellcrank is the only reliable one, but it requires a pushrod with a head like a nail.

The pushrod for the earlier bellcranks has no head, and the paddle of the machined steel bellcrank will slip past it. Solve the problem by making a pushrod out of a tenpenny nail. You'll have to grind down the head to fit inside the bellcrank, and cut the nail to a length of $2^7/_8$ inches for the $6^1/_4$-inch axle, or $2^5/_8$ inches for the 6-inch axle. The end of the pushrod should protrude $^1/_2$-inch from the end of the axle when in its outer position.

If you can't get a bellcrank for the earlier model S5 hub, you might make one by installing the mechanism of a Shimano bellcrank in a Sturmey-Archer right axle nut. Braze the two together. This may seem like a lot of trouble, but people who have experienced the five-speed hub in urban traffic usually get hooked on it. There are quite a few old-model five-speed hubs still around. I think that there are one or two sitting on a shelf in a bike shop in almost every town or lurking in the garage on Junior's old Raleigh Chopper.

The old model S5 has proven more reliable than the newer S5.1 and S5.2, which use an indicator spindle and pullchain at the left side. These use a spring to hold the axle gears in low gear position, and once these begin to slip they get constantly worse. John Forester reports that the S5.2 has a stronger spring, which seems to solve the problem. For older hubs, another cure is to grind the lugs on the axle and the gear that engages it, tapering them slightly to pull them together under power.

"Downwards Compatible"

In the S5.1 and S5.2, it is in any case necessary to use a primary sun gear (the gear that engages the axle) that has square (not beveled) corners on its lugs. The gear used in the earlier S5 had moderate bevels; the gear used on the FW, as mentioned previously, had an outright ramp on one side of each lug. These gears are all "downwards compatible" — later versions of the gears can be used with earlier versions of the hub. The same is true with axles. The S5.1 and S5.2 use an axle with a longer left slot, which works perfectly well in the S5 and FW.

When rebuilding the S5 or FW, note that the locknut that holds the gears and dog ring on the left end of the axle looks much like the usual Sturmey-Archer axle end locknut. It is *not* the same! The special dog-ring locknut has the threads cut away on the inside. A regular locknut will not screw down tight. To remove the gears, flatten the tabs on the lockwasher, unscrew the locknut, remove the lockwasher and dog ring, then push both gears all the way to the right to remove the axle key that holds them in place.

All models of the S5 should be controlled with a Sturmey-Archer trigger for the right side. For the left side, use another trigger shifting between second and third gear positions, or a derailleur control. As Sheldon Brown points out, the left cable is either completely tight or completely loose, so the derailleur control is acceptable — and it eliminates the need for cable adjustments.

Both controls should be on the handlebars. The greatest appeal of an internal hub is its rapid shifting, and this is lost with controls at the stem or down tube. Most problems with the Sturmey-Archer five-speed hub are traceable to the controls sold with it. These regularly slip out of gear, because they have only an unreliable click detent, not the positive escapement detent of the Sturmey-Archer trigger control.

If you've given up on finding parts to restore a four- or five-speed hub, note that an AW three-speed hub will fit into the same shell. But make sure the left ball cup in the shell has a flange around the central hole to center the three-speed hub's planet cage. If the flange is missing, as it is in some four- and five-speed hubs, you must replace the left ball cup. Even a splined ball cup can be replaced, by pounding it out with a hammer and wooden block. Install a new ball cup by pounding it in, also using a wooden block to prevent damage.

Interchangeable Germans

There is considerable interchangeability of parts among Fichtel und Sachs (Torpedo) internally geared hubs as well. This is mostly between hubs of similar design that are shifted differently (one two-speed Fichtel und Sachs hub comes in two versions, largely identical except that one shifts by backpedaling and the other shifts automatically in response to rotary speed) or that are provided with or without a coaster brake. This interchangeability is covered well in *Sutherland's Handbook for Bicycle Mechanics*.

Sturmey-Archer, Shimano, and Fichtel und Sachs sprockets for hub gears and coaster brakes are all interchangeable. Sturmey-Archer does not make a 21-tooth sprocket, though the other two manufacturers do. The 21-tooth is a useful size; with the usual 46-tooth chainwheel, it puts the top gear on a three-speed bike somewhere near 75 inches. This is where it should be; most three-speed bikes are seriously overgeared. The same sprocket works well with a five-speed hub, too. The fourth gear will be around 70 inches for upwind riding and the top gear around 85 inches for downwind riding.

This article is intended to supplement the information in Sutherland's handbook, to which I contributed. Readers should refer to that book for further information, including complete lists of parts numbers. Where this article indicates that parts are interchangeable and Sutherland's does not, it is due to Sutherland's very cautious approach in questions of interchangeability.

JSA

In Search of Parts ...

Ask your bike shop to query the companies listed below about local distributors. (These companies are not retailers.)

Sachs Motor Corporation, Ltd.
9615 Cote de Liesse Road
Dorval, Quebec
Canada H9P 1A3

Shimano American Corporation
205 Jefferson Road
Parsippany, NJ 07054

SunTour USA Inc.
10 Madison Road
Fairfield, NJ 07006

TI Sturmey-Archer of America
1014 Carolina Drive
West Chicago, IL 60185

For the record, N.K.'s address is:

Nankai Tekko Co.
9-8 Kohamanishi 1-Chome
Suminoeku
Osaka
Japan

Chainwheel Interchangeability Quirks

Even though two chainwheels have the same bolt circle diameter and hole size, they may not be interchangeable. Watch out for these problems:

• Chamfer on one face/both faces. Older Stronglight 93 chainwheels were chamfered on only one face. The chamfer was made to face the outside of the outer chainwheel. The teeth of both chainwheels were flush with the faces that attached to the spider, and the spider was unusually thick (about 4.7 millimeters). Newer Stronglight 93, 104, and 105 chainwheels have the same bolt circle, but are chamfered on both sides, with the teeth centered on the thickness of the chainwheels. The spider is thinner, about 3 millimeters. Newer chainwheels cannot be used on the old cranks unless the spider is thinned.

• T. A. Cyclotouriste inner chainwheels thinned/not thinned at bolt circle. Newer T. A. Cyclotouriste inner chainwheels in the larger sizes are thinned at the bolt circle and use spacer washers 3 millimeters thick. The purpose of thinning the chainwheels seems to be to adjust chainwheel spacing to the difference in size between chainwheels. You may have to use new spacer washers when replacing chainwheels, so it's a good idea to order a new bolt set.

• Fit of chainwheel inner lip to flange of spider. An example: Sugino Mighty Tour and Super Maxy 5 chainwheels have the same bolt circle and hole diameter, but the Mighty Tour spider has a flange which interferes with some Maxy 5 chainwheels. Some metal must be filed off the Maxy 5 chainwheels if they are to fit the Mighty Tour cranks.

• Spider thickness as it relates to chainwheel size difference. The Stronglight 99 has an unusually thick spider, about 4 millimeters. The chain tends to fall between the chainwheels when the chainwheels are only a few teeth different in size. Yet as a wide-range double, this crankset shifts beautifully. The greater the difference in the number of teeth between chainwheels, the farther the chain deflects to the side during shifting, and the farther apart the chainwheels should be. Select your crankset, add washers, or thin the spider as necessary for optimum shifting with the tooth difference you have chosen. T. A., as mentioned before, seems to have addressed this problem. The new SR Apex crankset which is a copy of the Stronglight 99 (28 teeth minimum, chainwheels interchangeable) has the same thick-spider design.

• Good news: the new Sugino AT inner chainwheel (minimum 24 teeth) is interchangeable with Avocet; and bad news: the new Shimano Deore inner chainwheel (minimum 26 teeth) is *not* interchangeable with Stronglight 99 and SR Apex. The bolt circle is a silly 1.5 millimeters smaller.

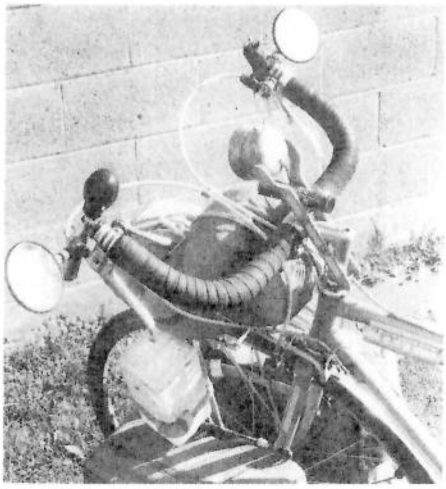
JSA

INVENTIONS

Projects and Prototypes

With this issue we inaugurate a new department. Lots of people are cooking up interesting things on wheels, and we think other people should hear about them. If you have an interesting project to show off, send us a letter! Please include a good, sharp glossy photograph that shows the interesting features well.

Please note that we have not tested any of these designs and make no endorsement or certification of their safety or sanity.



Phil Reynolds, who lives on his bicycle for much of the year, came by our office to show us his touring handlebars. They're made from standard dropped handlebars by cutting off the downward bend and tig-welding on a section of straight tubing instead (for straight tubing he used the bottom ends of the same handlebar). He finds them more comfortable than any other bars, and also likes to have all his controls in hand at all times (note gear lever boss brazed onto clamp band of brake lever). Phil gets his mail at Box 81, Newfoundland, NJ 07435.

Most recumbents have either a very long wheelbase or a very short one, to avoid interference between and the pedals the steering motion of the front wheel. Tom Traylor of Lomita, California, didn't want to settle for that, so he built a front-wheel-drive model whose drivetrain moves with the steering assembly. He says it handles very well. He sells plans for $12 from P.O. Box 326, Lomita, CA 90717.

From the Cutting Room Floor . . .

Last issue's mammoth section of ISO standards was to have included the short-but-important article below. Sadly, it didn't fit. We feel it is important enough to call your attention to it here and now.

No Standards, Standards Under Deliberation . . .

The ISO has no standards for:

bearing surface dimensions of spindles and bottom bracket cups
oversize bottom bracket cups for use in frames with stripped threading
chainwheel bolt circle diameters
chainwheel fixing bolt dimensions (and other parameters of chainwheel interchangeability)
bearing cone and cup dimensions for hubs
freewheel removers
dropout derailleur tab dimensions
toe clip size marking (the markings "large," "medium," and "small" do not currently correspond to the same size for different manufacturers)
toe clip to pedal mounting dimensions

Standardization of any of these parts would eliminate many headaches for distributors, retailers, and mechanics. Standards *are* being developed for:

one-piece crank dimensions
handlebar stem to handlebar clamping bolts
front fork inside diameter
bottom bracket spindle lengths
bottom bracket widths
chainlines
over-locknut distances
hub flange thickness and spoke hole dimensions
headset press-fit dimensions
fender eyelet threading
derailleur lever braze-on mounting bosses
handlebar stem wedge expander dimensions

but they are still in an early stage at which committee members are not free to make their deliberations public. *Bike Tech* suggests that the ISO examine the items listed above, if it has not begun discussion of them. You, our readers, may have additional suggestions. You may forward them to Joan Gardulski, Acting TAG Administrator, American National Standards Institute, 1430 Broadway, New York, NY 10018.

LETTERS

Wheel Repair

I have read with interest the first two issues of *Bike Tech*. It is good to see that a periodical with an emphasis on the technical aspects of cycling has arisen. The "Shop Talk" section especially shows great potential.

Eric Hjertberg's article on saving wheels was an excellent analysis and discussion of the most common problems seen in a shop. The section entitled "Thunk," however, may well lead some to try a violent method of repair when a technique somewhat more precise and less prone to overcorrection exists.

While working at framebuilder Matt Assenmacher's shop in Swartz Creek, Michigan, I was introduced to the "Drawer" technique of wheel repair for potato chip and severely bent wheels.

Rather than thunking the wheel on the floor, place that portion of the wheel which is most severely out of line in a partially opened desk or chest drawer. Then, apply pressure to the wheel until the major portion of the bend is gone. One can then attempt to bring the wheel further into true by more conventional methods. As was stated in the essay, this technique works best with low tensioned, heavy wheels.

By using the drawer, we have also had success straightening out lightweight clincher and sew-up wheels. This raises another question: should a wheel that has failed once already be ridden in a twice-weakened condition? On this issue, the worker can only make a suggestion, the final decision lies with the customer.

David Stanley
Flint, Michigan

And a Phone Call

Milton Raymond telephoned to report a successful preliminary test of his prototype tandem recumbent with both-wheel steering (see "This Bike Steers With Both Wheels," *Bike Tech* June 1982).

Ultimately this vehicle is to have two steering modes: simultaneous, parallel steering of both wheels for balancing; and steering of the front wheel alone to change the heading of the vehicle. For this initial test, Raymond set up the steering linkage to provide only the parallel motion (since that function was the major unknown feature of the design). In addition, since the transmission is not completed, this test was done by coasting.

In short trials at low speeds the bike was very easy to balance, even at slow walking speeds, despite its 12-foot wheelbase. Encouraged, Raymond and associate David Keppel took it down a nearby hill that afforded a half-mile run at approximately 25 mph (the brakes, judged to need more work, could only hold the speed down to that value). At this speed the bike was still easy to balance, though it showed extremely quick response ("like a fighter plane") to balancing control motions.

The hill has a 30-degree bend in the road halfway down, which posed a unique requirement since the steering linkage could not change the bike's heading: the bike had to finish the run 30 degrees crabwise, proceeding straight down the road while pointing into the opposite lane (to the consternation of oncoming drivers). This it did, without problems. (Raymond wants to test that again, though, with better brakes, since a strong brake, applied suddenly, may cause balancing difficulties in this orientation.)

Raymond is now proceeding with further steering design and testing, and refining his design for the transmission.

Feet First Federation

That enthusiastic band of folks who have bought Avatar 2000 recumbent bicycles has started an owners' club. This means a club newsletter with articles of special interest to Avatar owners, such as where to put the water bottles and how to carry panniers so they don't wobble, and a serialized article on the origins of the Avatar 2000 by co-inventor David Gordon Wilson. Anybody with ten bucks can join, but only actual owners (or former owners) can hold office in the club. The newsletter will undoubtedly be indispensable to true recumbent lovers; send your $10 to Avatar Owners' Club, c/o Chris Paulhus, #83, Old Mill Road, Harvard, MA 01451.

Bike Tech Involved in Bicycle Industry Seminars

Many of the authors and editors of *Bike Tech* will be involved in a *Bicycling* Magazine-sponsored International Cycling Conference for manufacturers, distributors, and dealers in conjunction with New York's International Cycle Show next February.

The conference, which will take place February 17 through 19 at the New York Hilton, will focus on technical and business topics of vital interest to the bicycle industry. Among the *Bike Tech* authors who will be represented in their field:

MIT's David Gordon Wilson, Zzip Design's Glen Brown, and exercise physiologist Ed Burke will be among the panelists in a seminar on aerodynamic design.

Bike Tech's John Schubert and John Allen and *Bicycling* Magazine's Frank Berto will offer a seminar on improving existing bicycle product lines.

Ritchey MountainBikes's Gary Fisher and *Bike Tech's* John Schubert will offer a seminar on designing lightweight off-road "klunkers."

Other seminars offered will include "Targeting the Entry Level Cyclist," "The Retail Marketplace," "What the Industry Can Do to Get More Women into the Sport," "The Nature, Direction and Future of the Domestic Bicycle Industry," "Marketing Seminar: Cycling Apparel — Clothes Make the Cyclist," and "Overcoming Distribution System Bottlenecks."

For more information and complete course descriptions, please indicate which seminars you are interested in and write to International Cycling Conference, c/o *Bike Tech*, 33 E. Minor St., Emmaus, PA 18049.