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BIKE TECH

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IN THE LAB

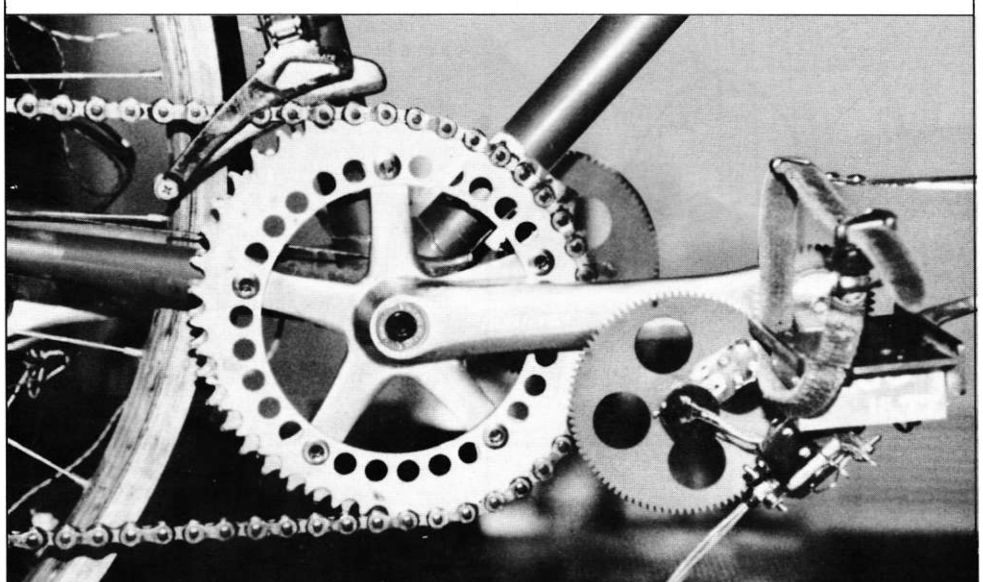
Designing Chainwheels to Optimize the Human Engine

Shinpei Okajima

Recently Honda introduced a 50 cc motorbike which can cover 200 kilometers on one liter of gasoline. Ten years ago its ancestor could do only 50 kilometers: the efficiency has been improved by a factor of four. Meanwhile the riders of the Tour de France eat about the same as the riders ten years ago, and cover a little less ground in their 24 days than did their predecessors. Has there been no improvement at all?

It could be said that the human bicycle engine has already reached its ultimate mechanical efficiency; but I do not think so. In the automobile industry, the facilities working just on gasoline mileage number in the thousands. In the bicycle industry we can see, from inventors' drawings, that many people worked to improve the bicycle — un-

Shinpei Okajima is Assistant Manager of the Development Section in Shimano's Technical Division in Osaka, Japan. One of his major projects this year has been the development of Shimano's new Biopace chainwheels. (A noted racer, he served not only as project engineer but also as one of the test riders.)



The instrumented pedal and crank, with two experimental chainwheels mounted. Pedal contains strain gauges for force measurement; small gear on crank end (visible through toe strap) turns large gear to operate potentiometer (black disc at center of large gear) for record of pedal tilt. Other large gear (beyond chainwheel) registers rotation of crank spindle.

IN THIS ISSUE

til the 1930s. Then people seemed to forget that bikes were useful tools of locomotion, worthy of development, and the inventions stopped.

At Shimano we are trying again to improve human engine efficiency, this time using the same facilities and procedures which would be used in the laboratories of the automobile companies. With this highly integrated equipment, and with the background provided by the many achievements of researchers in biomechanics and muscle physiology, we have found that there is still a lot of room for improvement in the application of human beings as bicycle engines.

Can a Tool Improve Your Performance?

Except in the case of top athletes, daily training can improve performance a lot more than equipment can — you would do better to spend your money at a grocery store than in a bike shop. But there is another reason we seek to discover the "golden axe":

Most sports require specific techniques and a kind of sacrifice before you can enjoy them — it can take years of practice to run hard without hurting your knees, or to spin skillfully on a bicycle, for instance.

One of the best qualities a tool can have is to be easy to learn and easy to use. More than improving the rider's performance a little, what we hope to do is offer riders the joy of the sport with less sweat and strain.

Impedance Matching

A vigorous bicycle rider seeks two types of power output: maximum power for sprinting, and sustained moderate power for long-distance events. (The word "power" in this discussion will be used in a strict sense: the rate of energy transfer; the amount of work done per unit of time. It can also be expressed as the product of force and velocity, or, for rotary devices such as cranks, as the product of torque and rate of rotation.) A ten-speed bicycle rider is very familiar with the relation between torque and pedaling speed and uses the derailleur to make the best use of it; it is nearly impossible to compete in most road races without a multiple freewheel.

In a technical phrase widely used in electronics but less-known in mechanics, the benefit obtained by selecting the right gear is called *impedance matching* (Figure 1): the speed and torque at which the muscles work best is matched (through use of a gear ratio) to the speed and torque needed at the wheel.

But the best speed for the muscles varies. Good sprinters spin at 160 rpm. Pursuiters spin at 120 rpm.

Road riders vary the spinning speed from 70 to 130 rpm, depending on the situation, to maximize efficiency. When he can enjoy cruising, a racer spins relatively slowly to match the lower power output requirement; since rapid leg motion increases friction in the muscles and loss of kinetic energy, in this case it is pointless to maintain a high cadence.

Then at a critical point in the race, say to join a break, he will shift to a higher gear to go faster, but the higher power for this speed does not come from using the higher gear — the rider produces the higher power by *raising the cadence*, to 110 rpm or so. The gear increases less than the speed does, and simply allows the feet to keep up. If the rider cannot spin with the heavy ratio, his power will drop and he will fall behind the pack.

So the ideal spinning speed seems to reflect the power required, as is common for the efficient operation of any power unit: the spinning (or muscle contraction rate) of racers is rapid, to produce a high power output, while slower spinning is economical at lower power outputs.

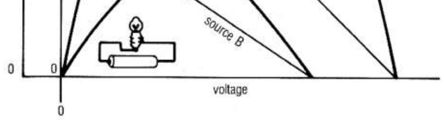


Figure 1: Impedance-matching concept — voltage-to-current relationships, and power curves, for two electric power sources with different internal resistances and voltages. Maximum power output occurs when load is selected to draw a specific amount of current which is different for the two sources.

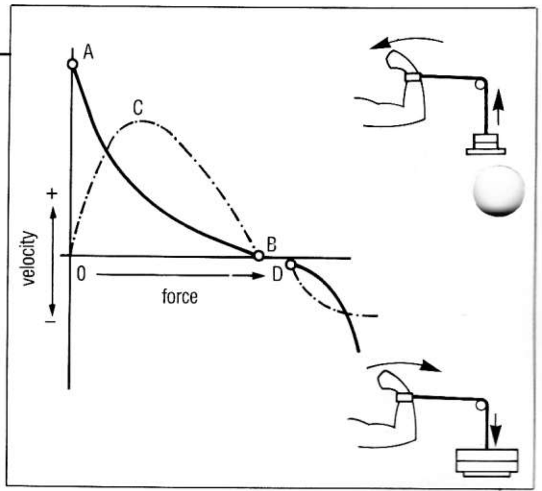


Figure 2: Force-to-velocity relationship of muscle. Positive velocity corresponds to active contraction, or "concentric work," ranging from A (maximum speed at no load, or "shadow boxing") to B (high force but no motion, or "isometric" effort). Maximum power (best impedance match) occurs at C. At still greater loads, D and beyond, the velocity becomes negative (i.e., the muscle is forced to extend), and the work it consumes, acting as a brake, is called "eccentric work."

Slow Contraction

The physiologist A.V. Hill, in a classic experiment, demonstrated the force-to-velocity relationship of muscle, with an electrically stimulated frog leg muscle (Figure 2). Cyclists have rarely thought about such

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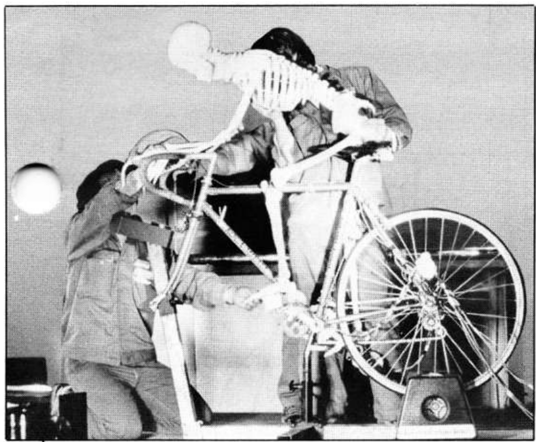


Shimano researchers examine anthropometric parameters.

questions in specific numerical terms. But what if we did — what would this relationship imply for cycling motions?

The muscle contraction speed during pedaling is relatively slow. It is about one-third as fast as that of running at a similar level of effort: since muscle contraction is roughly proportional to foot speed, the ratio (with the typical values shown here) is

$$\frac{170 \text{ mm crank length}}{340 \text{ mm wheel radius}} \times \frac{16 \text{ teeth on cog}}{48 \text{ teeth on chainwheel}} \times \frac{40 \text{ km/h cycling speed}}{20 \text{ km/h running speed}} = 1/3$$

This lower speed range that the human engine runs in when pedaling means that the opportunity for power output is also relatively low. The engine runs three times as fast, allowing a higher power output, when running, because there is less restriction of the way the work is transmitted. On a bicycle we get some freedom from gravity by using wheels to suspend body weight, and we get higher speed by using a stepped-up transmission. But at the same time we have to transmit our work through pedals, and thus lose the speed of muscle contraction because pedaling requires a circular path for the foot, and requires the direction of the useful force on the pedals to be perpendicular to the crank. These conditions are not easy nor natural. The mechanism of the bicycle is effective for locomotion, but not made for muscle to display its potential.

Mashers All

An anthropometric study shows that all human bodies share certain features (trained cyclists can obtain massive thighs, but not three legs). Crank torque data for various cyclists show that they tend to push, not to spin or pull (Figure 3). They show the peak force when the crank reaches its forward horizontal position, and the minimum a little before the crank's upper vertical position.

From beginners to champions, the improvement of the ratio between pedal force and crank torque (which depends on applying the force in a useful direction) is less than 10 percent at 60 rpm.

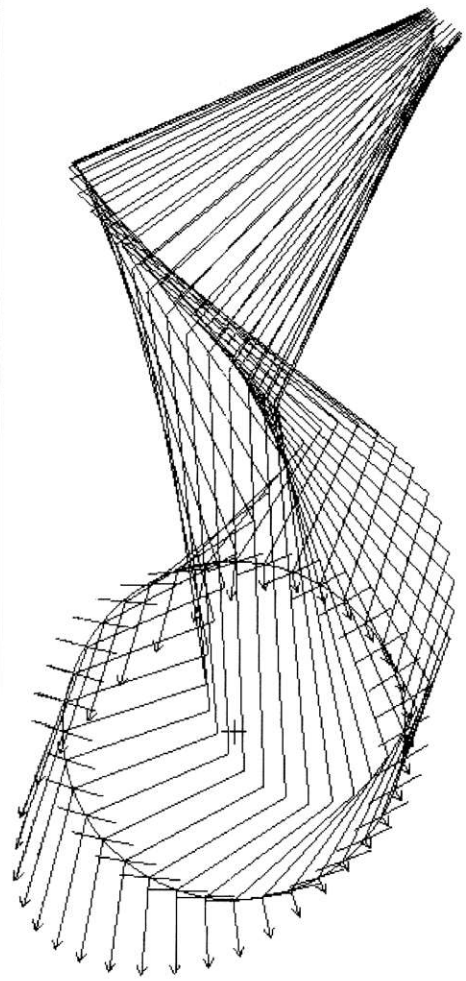


Figure 3: Diagram of typical pedaling-force pattern. Short slanted lines around circle represent positions of pedal, and arrows attached to them represent force exerted on pedal. Lines above circle represent leg in various positions, swinging from hip joint at top. (Note: lower, or "foot," segment represents a line from pedal spindle to *ankle*, not to heel.)

The pedaling-force patterns of trained cyclists are the results of optimization of physical capability with the existing bicycle mechanism. These cyclists cannot spin three times as fast as untrained cyclists, but they do prefer to spin 50 percent faster.

We concluded that a higher muscle contraction speed with less difficulty in spinning the pedals would increase the efficiency.

So how could we relax the restrictions?

We adopted three approaches:

- an improvement of impedance matching between the "output resistance" of the muscles and the "input resistance," represented by the workload;
- a reduction of the loss of the leg's kinetic energy;
- a reduction of the requirement for muscle coordination skills the rider must learn.

Legs

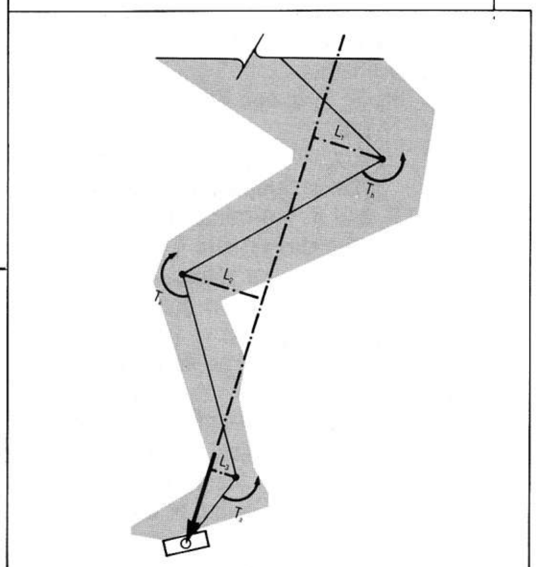
Human legs can be described as multiple lever-actuators connected at three joints (Figure 4a). The ends of each leg transmit force to the pedal at the foot, and to the upper body at the hip. Since the force on the pedal is the result of the torques at the three joints, the following equation applies:

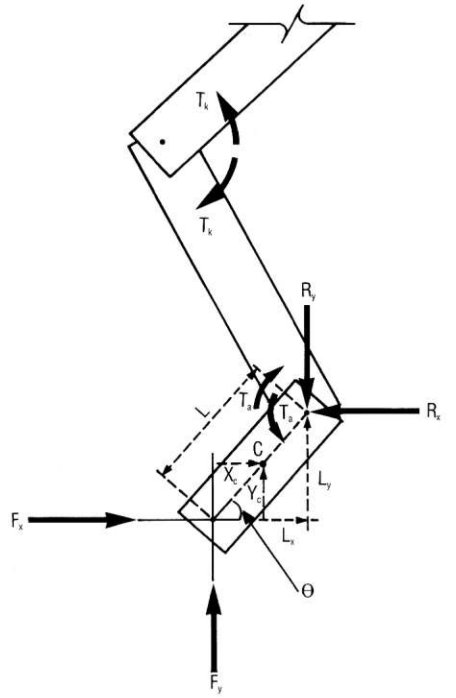
$$T_h/L_1 = T_k/L_2 = T_a/L_3$$

where T_h is hip torque, T_k is knee torque, T_a is ankle torque, and L is the effective lever

Figure 4: Legs as multiple lever-actuators.

a. Quasi-static model (inertial forces not significant) — relationship of joint torques: $T_h/L_1 = T_k/L_2 = T_a/L_3$





b. Dynamic model — calculation of joint torques:

Equations:

Balance of forces:

$$F_x - R_x = m\ddot{x} \quad (1)$$

$$F_y - R_y = m\ddot{y} \quad (2)$$

Balance of torques (around center of gravity C):

$$T_a + F_x y_c + R_x (L_y - y_c) - F_y x_c - R_y (L_x - x_c) = I\ddot{\theta} \quad (3)$$

where

$\ddot{x} = d^2x/dx^2$, i.e., the x-component of acceleration; and similarly $\ddot{y} = d^2y/dy^2$ and $\ddot{\theta} = d^2\theta/d\theta^2$;

and m is mass, I is moment of inertia, and other terms are as shown in the diagram.

Procedure:

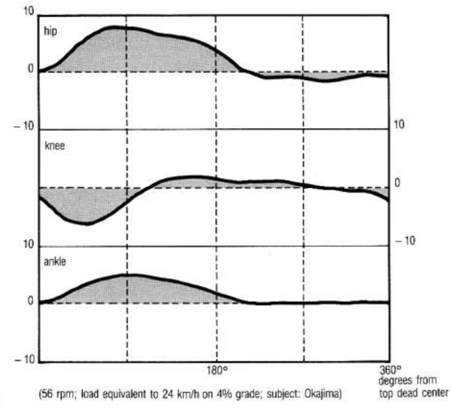
F_x, F_y, $\ddot{x}$, $\ddot{y}$, and $\ddot{\theta}$ are measured data.

m, I, x_c, y_c, and L are deduced from anthropometric statistics.

R_x and R_y are calculated from equations (1) and (2).

Given all these values, equation (3) can be solved to give joint torque T_a.

The procedure can then be repeated with values appropriate for the knee-to-ankle segment (including equal and opposite reactions to the T_a, R_x, and R_y just found, acting at the ankle) to find knee torque; and again to find hip torque.



c. History of leg joint torques during one pedal revolution.

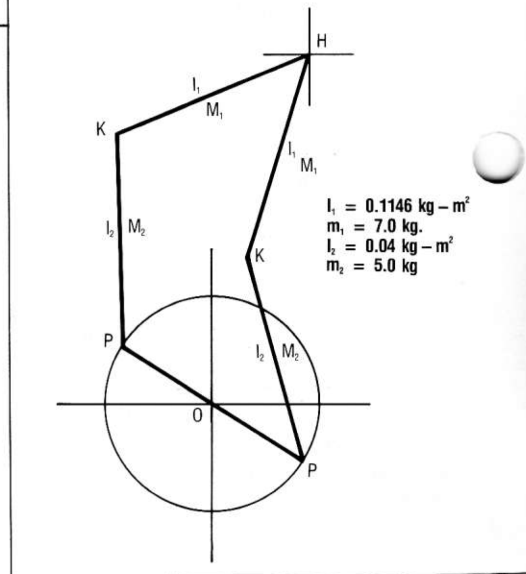


Figure 5: Simplified pin-jointed model of leg. O, H, K, and P are friction-free pin joints.

length for each joint — the distance of the joint from the line of the pedal force vector. (To be exact, this equation applies only to slower cadences, at which the momentum of the leg is small. The more general solution, which takes mass into account, is described in Figure 4b.) Figure 4c shows a typical history of one leg's joint torques during one pedal revolution. (Note in particular that the knee joint's torque must reverse during the power stroke.)

Weakest Link

Since the joints are connected in series, only one joint — the weakest — can exert its maximum torque. This weakest joint is therefore the restricting factor for a rider's performance.

Sometimes multiple muscle groups (actuators) cooperate to make the resultant force perpendicular to the crank. These muscle groups must cooperate three-dimensionally to keep the joint in a stable position. In this case the weakest *muscle group* around the weakest joint is the real restricting factor. The restricting muscle varies during the pedaling stroke, because lever lengths for joints and participation of muscle groups vary with crank angle, and varying the hip and upper body placement according to the output also has some influence.

In general the limiting "link" is a muscle group around the knee. We are built to walk using mainly hip joint torque in a pendulum motion, not to pedal using knee joint torque for back-and-forth motion and hip joint torque for up-and-down motion. Cramp and overuse syndrome can be observed most often in muscle groups driving the knee.

We saw two specific restrictions to be solved: 1) the difficulty of spinning, both in the motion and in the direction the force must be applied, restricts the speed of muscle contraction during pedaling to a rather slow rate, and requires the force to be on the high side. 2) The knee joint is overused, while the hip joint is underused. (The ankle joint plays a passive role; rather than extending it, its muscles simply act as "brakes" to limit the amount it flexes under the forces applied by the upper leg.) Since pedaling involves the interaction of many variables, solving these restricting factors should create an overall improvement that will allow hidden capabilities to be developed.

Velocity — An Angle

A high-speed film analysis of open-air bike riding shows very small acceleration in each stroke (about a 2 centimeter gain of distance relative to an object moving at the bike's average speed), because the inertia of the total mass of bike and rider is relatively large compared to the force variation given by pedaling. Thus the angular velocity (rate of rotation) of the crank is almost even throughout each pedal stroke.

But of the major ergonomically important variables on a bicycle — type of power motion, magnitude of motion, velocity of motion, and basic rider position — velocity is relatively easy to modify, within limits, on existing bikes, simply by altering the drivetrain. This modification offers an important opportunity — varying the velocity can extend the impedance-matching concept to provide different matches at different times *during each stroke* as the leg's output varies.

We decided that an appropriately uneven angular velocity pattern would reduce the loss of kinetic energy, and also make it easier for the rider to switch between the firing of different muscle groups at appropriate times

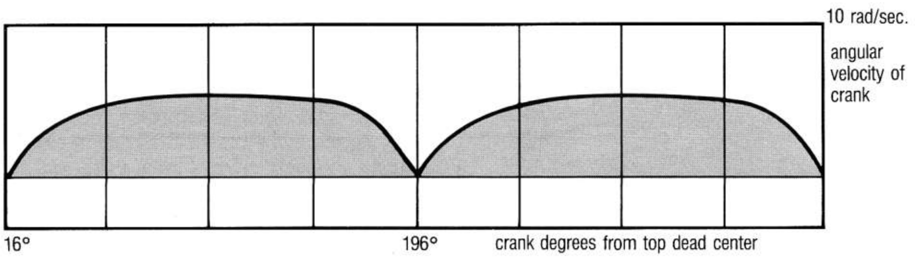


Figure 6: Crank-velocity pattern for optimum power transfer with link model of Figure 5.

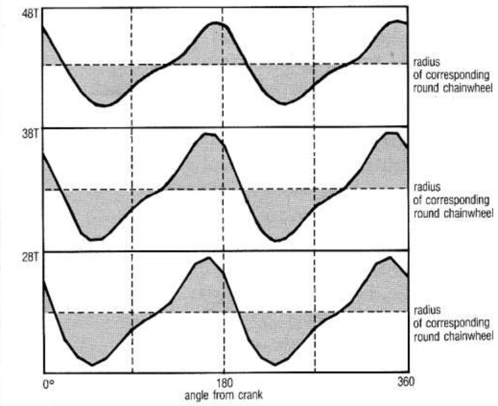


Figure 7: Radius of Biopace chainwheels as a function of angle from crank.

(to be specific, at the reversal of knee torque). These two benefits, we thought, could offer a substantial solution to spinning difficulties.

Ideal Variation

We started with a computer simulation for a crank driven by a simplified leg — a solid, pin-jointed model with appropriate masses, lengths, and moments of inertia (Figure 5). For this model, Figure 6 shows the ideal angular velocity variation for an optimum impedance match. As the joints transmit the kinetic energy of the various segments, the crank's angular velocity varies irregularly. The assumptions made here might in principle have made the result inaccurate, but we later obtained similar results with the actual mechanism. We estimate that this information is applicable except in the case when the hip socket is not kept stationary.

Back to Reality

With these insights we set out to develop a chainwheel that would provide an appropriate pattern of varying crank velocity. Using a computer-aided design and manufacturing system, we designed and made many different prototypes, sometimes differing by less than a millimeter in various radii tested. (This fabrication method was vital not only because of the complex shape, but because the shape of each tooth is different from that of its neighbors.)

We tested each version by mounting it on an instrumented bicycle and riding on a treadmill with it, while several types of sensors fed data on riding peformance into our computer (see appendix for details). Meanwhile the rider formed subjective judgments about the way each version worked.

The chainwheel that eventually resulted, which we call the Biopace®, bears only a distant resemblance to Figure 6, for a number of reasons. To begin with, actual chainwheels cannot provide motions with such sharp fluctuations. In addition, the model was for a frictionless and ankle-less "limb," whose behavior is somewhat different from that of a human leg.

Among mechanically practical velocity patterns, it turns out that the simple approximate sine-curve provided by the familiar elliptical chainwheel can come about as close to the Figure 6 pattern as does any other. For some of our initial experiments we used

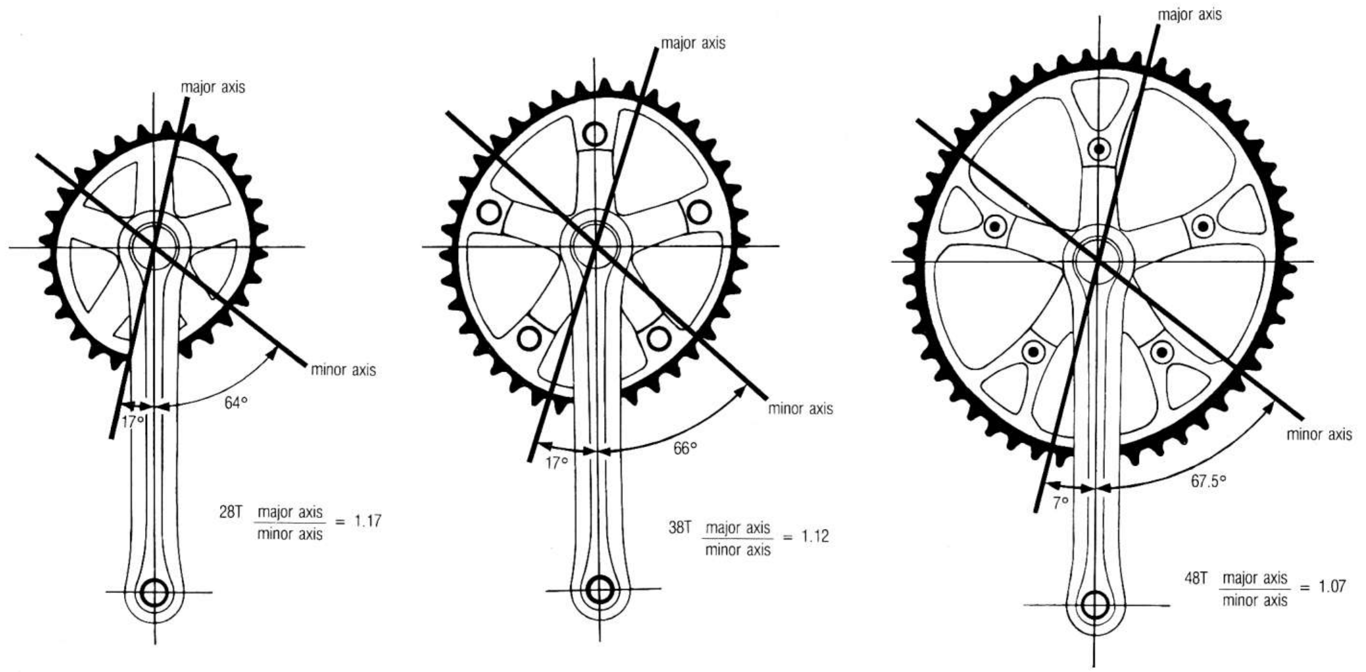


Figure 8: Biopace chainwheels.

an elliptical chainwheel, drilled for crank bolts every 12 degrees, so it could be mounted at a variety of phase angles. However, there is a fundamental difference between our application and all the traditional installations of elliptical chainwheels: the speed variations are reversed! The traditional installation gives the crank its *maximum* speed at the top and bottom positions, where Figure 6 gives a minimum speed.

We later departed from the elliptical shape, in order to deal with the muscle-switching aspect of our agenda: at each switching point, the two muscles may both pull at once for a moment, opposing each other, and wasting effort. A small pause will make the switching easier and avoid such conflict, so we added a slight break in the velocity curve at the knee-torque switching point. (This feature may be a bit confusing — it is not a period of slower velocity, but a period of slower *change* in velocity, so that the inertial deceleration force on the foot lets up for a moment while the muscles are switched.) The final shape is plotted in Figure 7 and drawn in Figure 8.

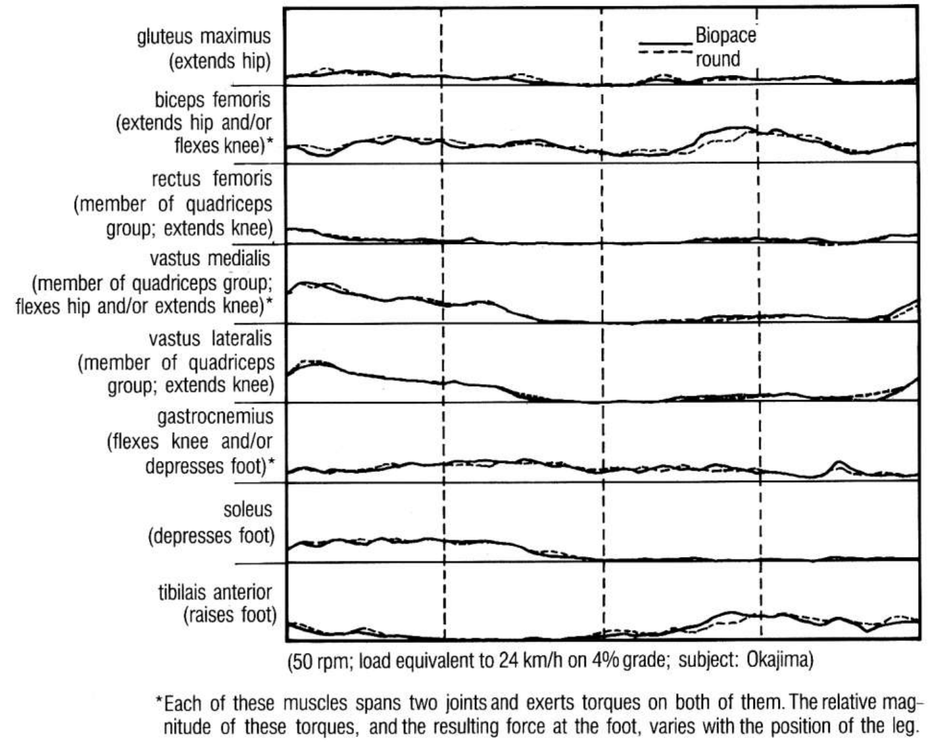


Figure 10: Electromyograms for round and Biopace chainwheels at equal cadences and loads.

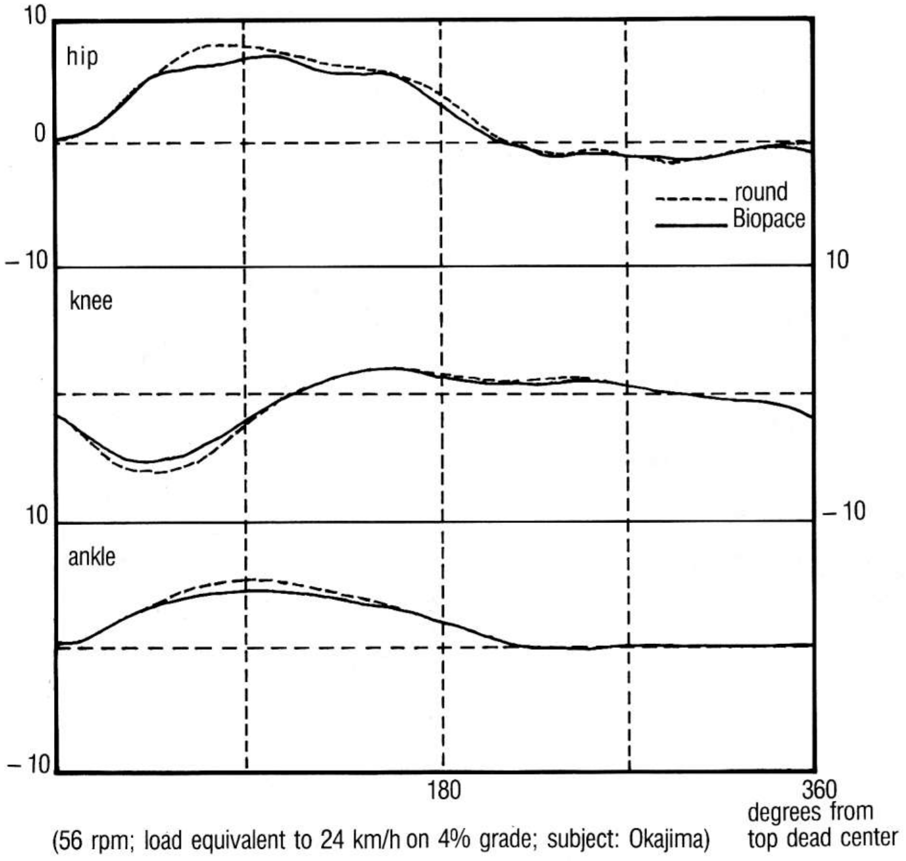


Figure 9: Joint torques for round and Biopace chainwheels at equal cadences and loads.

The Bottom Line

We tested the validity of our final version in two ways, both of them derived from data taken with human subjects riding a bicycle equipped alternately with a round chainwheel and with the new chainwheel: we took measurements from which we calculated joint torque, and we took direct measurements of the electrical magnitude of muscle function.

The joint torque calculations were performed by measuring the forces applied to the bicycle and the positions of the rider's legs at many instants during riding (using the equipment described in the Appendix), and performing computer modeling, using the equations in Figure 4b, to deduce the joint torques that would agree with the observed motions of the leg masses. The results are plotted in Figure 9.

The muscle measurements, known as electromyography, are shown in Figure 10, and reflect the intensity of contraction of the actual individual muscles monitored.

Both these methods of analysis show that by allowing faster contraction speed, the uneven crank velocity allows the leg muscles to reduce the force exerted while still maintaining a given power output. Upper body vibration is also reduced for a given power output, because less lifting torque is applied at the hip joint.

Conclusion

The Biopace chainwheel is designed to allow the rider, as a power unit, to increase contraction velocity and obtain higher power where it is available (i.e., when the leg joint torque is large) but avoid increases in energy loss (by decreasing the velocity where joint torque is small, and decreasing the force where muscle switching occurs).

The idea called impedance matching introduced here is commonly used in the design of many power sources. The idea will also be useful for designing any tool driven by muscle contraction.

We open this paper for discussion and hope it draws the interest of other researchers.

Acknowledgements — Works accomplished by Dr. P. R. Cavanagh and Dr. C. Kyle inspired us to start our study. Many other papers, by fellows of the American College of Sports Medicine and by members of the International Society of Biomechanics, have saved us a lot of time and sustained our confidence as well.

Appendix

• Why 48-38-28? A Biopace chainwheel requires extra capacity (cage height) in the front derailleur; about two teeth for the 48-toothed chainwheel and three teeth for the 28. For a 25-tooth derailleur capacity, then (the largest generally available) the overall range in our chainwheels can be 20 teeth. As we wanted 48 for cruising, 28 was an automatic decision.

48-13 is high enough for touring (or the rider can select rear cogs of 11 or 12). The Biopace 28 gives a performance about like a 26-tooth round chainwheel.

We would have liked to make the center chainwheel a size that would give a crank speed halfway between those of the large and small chainwheels. That size would be the reciprocal mean, or 35.37. But to get acceptable downshifting performance, we had to raise it to 38.

Other selections are under consideration.

• Improving shifting performance: At the maximum-radius parts of the chainwheel, where the chain is under high tension which would normally make shifting difficult, each of the three chainwheels has its teeth aligned to hook the chain exactly in line with the links. Computer-aided design calculated the alignment by taking into account the length of the chain segment that spans the three-dimensional diagonal jump between chainwheels.

• Specific designs for different terrains: 48 is designed for flat and downhill or high-speed riding; 38 for flat and uphill or steady riding; 28 for uphill with heavy load or on rough terrain. The riding form and the nature of load varies. Power consumed by air resistance is a cubic function of speed, while power consumed by gravity and friction varies in simple proportion to speed. To match these different types of resistance, a rider's crank torque (and thus joint torque) patterns vary even with round chainwheels. In accord with the impedance matching principle, therefore, the shape of each chainwheel is different.

• Laboratory hardware: A. Force data were acquired using dynamometer sensors built into the pedals, handlebar, and saddle. These transducers were designed so as not to increase weight and size, for use in open-air riding. On the seatpost and handlebar stem, we used glued-on foil strain-gauge matrices, arranged in bridge circuits (to eliminate interference between force components in different dimensions).

B. Angle data was obtained using two potentiometers for each angle, to eliminate a dead interval.

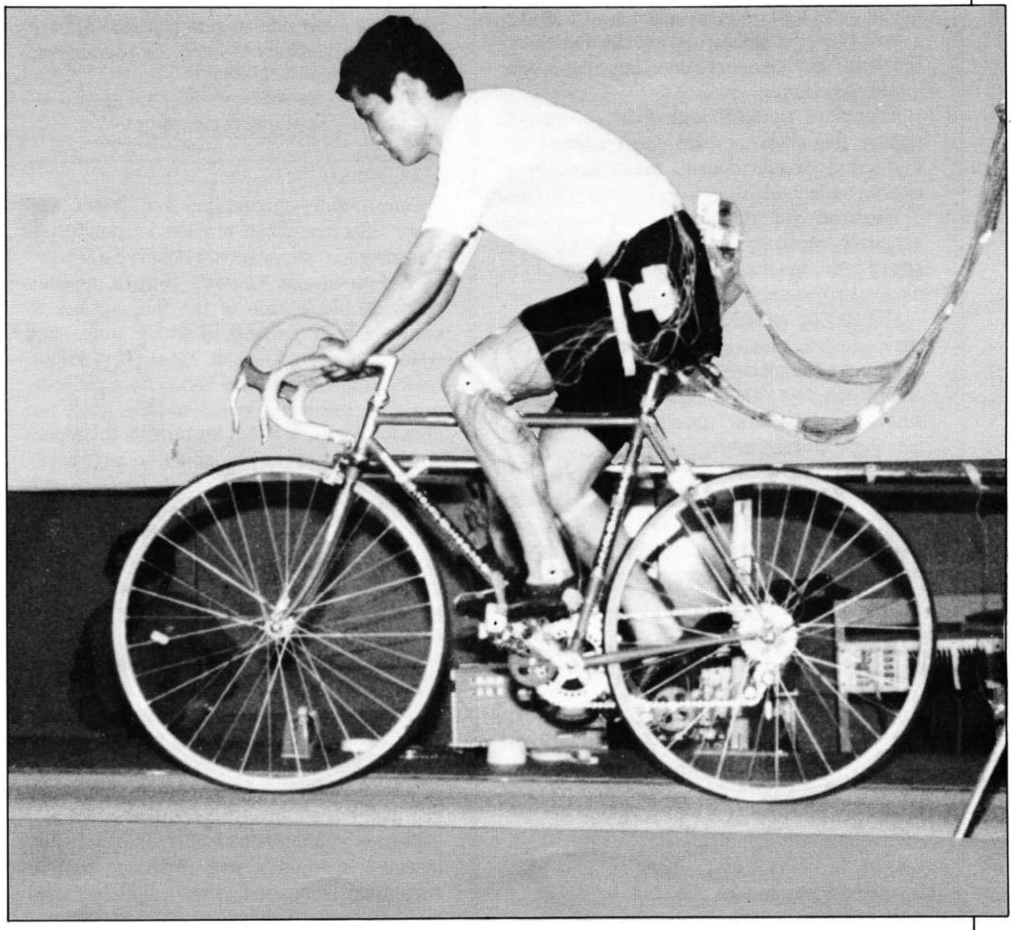
C. Joint displacement data was acquired using high-speed filming and a computerized position-sensing system. As the digitizing of film is a time-consuming job, we later used a setup which provided data in real time.

D. A treadmill with a hard surface was used in lab testing. Any apparatus mechanically holding a bike, or using friction for loading, is not suitable to our study because it influences the crank torque pattern. The body tends to move if the bike is locked to the ground. Friction does not simulate the load of actual cycling well, and tends to increase with heat buildup.

E. All data was digitized by an analog-to-digital converter and stored in a computer memory or disk. A monitoring program allowed us to check that everything was going well on a monitor TV. The thorough analysis for each purpose was performed afterward.

• Laboratory software: Raw data was studied first. Then 20 to 50 cycles of pedaling data were averaged according to the crank angle, and smoothed if necessary with a spline-function or moving-average method. Crank torque, power, crank angular velocity, joint torque, contraction force, and speed of actuators were calculated to compare the data from different trials.

These data were correlated with the riders' opinions and performance, then fed back to the design of the chainwheel.



The author rides the instrumented bike in a treadmill test, wearing electromyograph sensors over his leg muscles and light-emitting diodes at the locations of his joints.

SHOP TALK

Wheelbuilding — A Tension Method

Eric Hjertberg

Tension is the bicycle wheel's most important asset. Even the best rims and spokes cannot make up for incorrect tension. And straightness without uniform tension is short-lived. Accordingly, building methods which monitor and balance spoke tension should produce better wheels than those based primarily on visual straightness.

At Wheelsmith, our long interest in spoke tension has produced building techniques that help balance tension as well as simply increase it. Rather than prolonging construction, these tricks create more stable wheels which are actually quicker to finish. Once complete, these wheels are more evenly braced against road shocks and more resistant to spoke breakage because loads are better shared.

Aluminum rims, though light, are rigid enough that they can hide uneven tensions. Just being straight doesn't mean a wheel is evenly tensioned.

Whatever your building system, tension-sensitive methods can help improve and speed your results. Try adding these techniques to your building routine:

1) Start by creating "Ground Zero" — a perfectly true, *low tension* state which will serve as a foundation for further tightening.

2) Upon this stable base add tension in small, equal "layers" (one-half turn per nipple, each time around).

3) Following each layer of increase, first correct roundness and then fix side-to-side errors with tension balancing.

4) Avoid "overstressing" the finished wheel with sideways bouncing.

Ground Zero

After loosely assembling the wheel, tighten each nipple until only two threads on the spoke shaft are still visible. If the spoke elbows do not perfectly fit the hub flange, bend them gently into place. Then tighten each nipple one-half turn and look for enough tension to begin truing. After several half-turn advances, spokes will start to feel snug rather than relaxed.

In this lowest tension condition, truing is easier than at any other time. Now is the time to address problems with the rim seam (joint); with low tension in the spokes, it can be squeezed or levered with little chance of wheel collapse. Adjust side-to-side errors, roundness, and dishing until the rim is *as true as you want the finished wheel.*

This foundation of perfect straightness at lowest tension, which we call Ground Zero, is the basis of an excellent wheel. When tensions are this low they are within mere pounds of equality, because otherwise some spokes would be completely slack. From such a foundation we can confidently add tension with a minimum of fuss. I cannot overemphasize the importance of Ground Zero in eventually obtaining even tensions. Extra time spent perfecting the wheel now is well worth the effort.

Layers of Tension

Add tension to the zeroed wheel. Depending on your experience, the increase you add each time around may be as small as one-half turn per nipple or as much as two full turns. In general, smaller increases are easier to handle.

Inspect roundness. A lightweight rim's roundness is a good indicator of spoke tightness. When the rim is round, tensions are more even and side-to-side adjustments are quicker and unlikely to spoil the roundness.

Tension Balancing

Side-to-side corrections are faster and more effective when tension is monitored. For instance, suppose you observe a wobble in a four-spoke region. Before making changes, pluck each of the four spokes to see which is tightest (highest note) and which is loosest (lowest note). If possible, relieve these extremes more than the others. Let the rim's visual wobble mark the spot for correction, but let tension (by plucking) help decide which spokes to adjust.

Since tension is closely related to roundness, tension balancing allows simultaneous improvements to side-to-side *and* roundness errors. Remember, though, that rear wheels have tighter spokes on the freewheel side, so tension comparisons are only valid among spokes from the same side.

Truing a wheel by tone is a strange idea to many builders who depend primarily on visual displacement and let tension distribution "just happen." Learning to balance is like opening a new set of eyes. In my own case wheels became more responsive, and easier and faster to true.

Before adding each new layer of tension, measure symmetry with a dishing tool. If correction is needed, simply add the next one-half turn to only one side to improve symmetry. Now check and correct roundness and fix any side-to-side wobbles. Add another half-turn of tension and repeat the checks. Having started from a stable Ground Zero, you will find roundness exceptionally stable during these tightening sequences.

Continue small layers of tension increase and patient corrections until the wheel feels as tight as a known good wheel of similar design. Do not wait for the wheel to become unstable or badly warped as evidence of too much tension. By then the rim may be permanently deformed.

At full tension little straightening is needed because the wheel has never been allowed far from finished trueness. Do not be caught making many corrections at high tension. A fully tightened wheel resists change — that is why it's so durable on the road.

Avoid Overstressing

A tension-balanced wheel is remarkably stable and does not need the prestressing or overstressing often applied to lesser wheels. Stressing usually tries to prevent two types of problems. One is the stretching of parts during building and use, which loosens the wheel. The second is spoke windup created during tightening, which, when released, can cause wobbles.

You can extract troublesome stretch quickly but safely by grabbing pairs of parallel spokes in each hand (one pair on each side of the wheel at the same rim region) and squeezing firmly. Squeeze once shortly after Ground Zero, and a final time near completion.

Windup can be minimized by lubricating the threads, and by compensating with the spoke wrench after every turn. Hold the spoke shaft between your fingers as you tighten it to judge the amount of correction required, usually about one-quarter turn.

If spokes are squeezed and unwound, sideways "bouncing" of the wheel becomes unnecessary. Laying a wheel on its side, grabbing the rim at three and nine o'clock, and pushing down vigorously is a popular but traumatic and dangerous technique. Six-speed rear spacing and low spoke numbers are especially vulnerable to this crash simulation.

Some builders have perfected this move for their own use, but learning it is costly, and most admit to occasional failures. Avoid such extreme measures. Surviving a massive side load might be a sign of strength, but it might also ruin or permanently weaken a wheel.

These tension balancing methods should be part of every builder's routine. They have improved and speeded our building at Wheelsmith, and we think you will discover the same.

Additional information on wheel design, lacing, and truing can be found in The Bicycle Wheel *by Jobst Brandt, published by Avocet, Inc., of Menlo Park, California.*

TEST RESULTS

Calibrated Destructive Testing of Bicycle Frames

Jacquie Phelan with Charles Cunningham

What effect does oversized aluminum tubing really have on the strength of a bicycle frame? Its theoretical advantages for strength, and especially for rigidity, have been discussed extensively (see, for example, Gary Klein, "A Hundred Years of Monopoly: Is Steel the Ultimate Frame Material?", *Bicycling*, September/October 1981; and Crispin Miller, "Tubing Rigidity," *Bike Tech*, August 1982), but there hasn't been much actual data gathered on real frames.

Charles Cunningham of Fairfax, California, has a strong interest in this question because he builds aluminum bicycle frames — mostly not for road bikes, though, as do most aluminum builders, but for a somewhat more rugged existence — the off-road "ballooner" bicycles that have evolved in his neighborhood (better known as Marin County).

Cunningham continually strives to advance the design of his brand of off-road bicycle. Through several years of seeing different kinds of bikes get bent and broken in various ways (sometimes even making deliberately underbuilt frames and riding them himself to see where they would fail), he has refined his designs a great deal, but all this experience still left him looking for hard numbers. In particular, how would one of his frames stack up (so to speak) against a steel frame in an identical impact? Cunningham decided to find out.

Trashing a Frame Scientifically

He built a setup to simulate the frame-buckling forces applied by a head-on crash. It consists of a scissor jack modified to pull together rather than push apart, an accurate force gauge, and some welded steel to serve as a dummy fork and steerer tube (Figure 1).

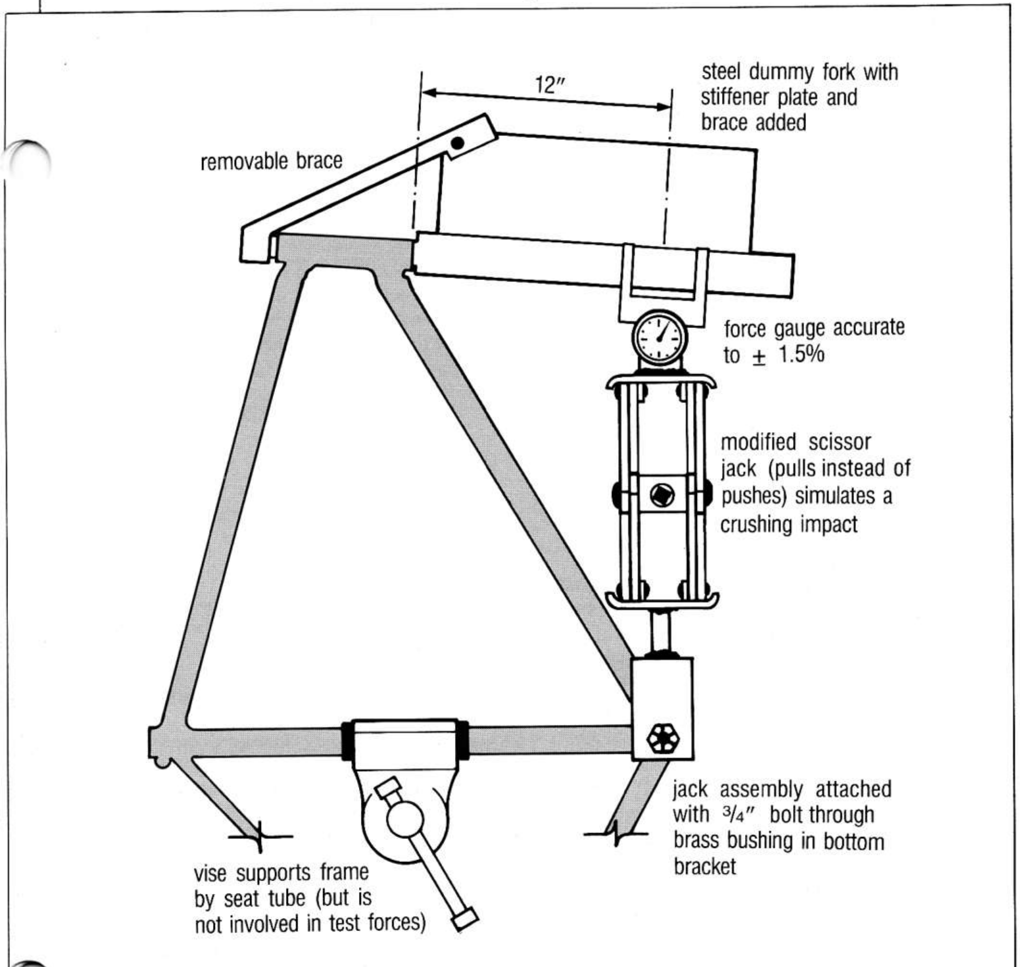


Figure 1: Test Setup

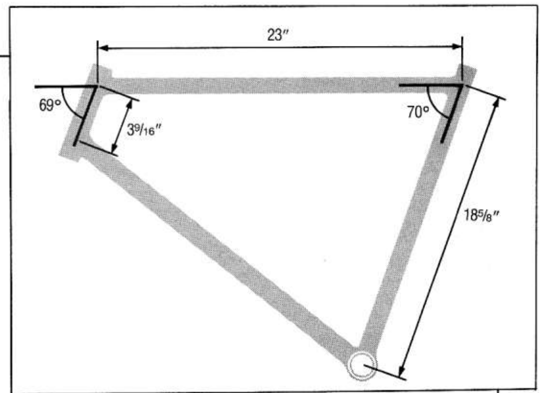


Figure 2: Geometry of Frames Tested

He performed his test on front triangles of two frame designs, identical in size and geometry (Figure 2), but of different materials. The first was a Cunningham heat-treated one of 6061-T6 aluminum[1] with a 1½ × 0.065-inch top tube and a 1⅝ × 0.058-inch down tube; the triangle weighed 2.0 pounds, corresponding to a whole-frame weight of 3.5 pounds.

The second was of 4130 chrome-moly tubing, built by Steve Potts, an accomplished off-road builder also of Marin County. The chrome-moly triangle was made with a 1⅛ × 0.035-inch top tube and a 1¼ × 0.049-inch down tube, and had brazed lugless joints (which is considered the standard in custom off-road frames). It weighed 3.1 pounds, corresponding to a whole-frame weight of 6.5 pounds.

After setting up the equipment and bolting in the aluminum frame, Cunningham gathered a crowd of local framebuilders and cyclists to witness and assist the desecration of these rare and valuable frames for the sake of science. The level of suspense was high; professional pride was at stake. Nevertheless, a survey of our faces showed only detached curiosity; the real object here was not to devalue existing designs, but to gather knowledge that could be used to improve them.

We quickly cranked the load through the low range, and then, as we began to look for failure, we slowed down and increased the tension loads in 25-pound increments, noting the angle of deflection of the head tube at every 50-pound increase, and inspecting the frame for signs of permanent deformation

[1]*A system of four-digit numerical designations is used for identifying wrought aluminum and its alloys. The major alloying element is indicated by the first digit. In the 6000 series, magnesium and silicon are the major alloying elements. The second number indicates modifications in impurity limits. A zero means that there is no special control on individual impurities. In the 6000 group, the last two of the four digits have no special significance, but serve only to identify the individual alloys in the group.*

The T6 is a temper designation. The T refers to thermal treatment, and the digit indicates specific sequences of basic treatments. T6 means: heat-treated and then artificially aged.

(From the 1972 Ducommun Catalog, *Ducommun Metals and Supply, Los Angeles; section J, pp. 53-54).*

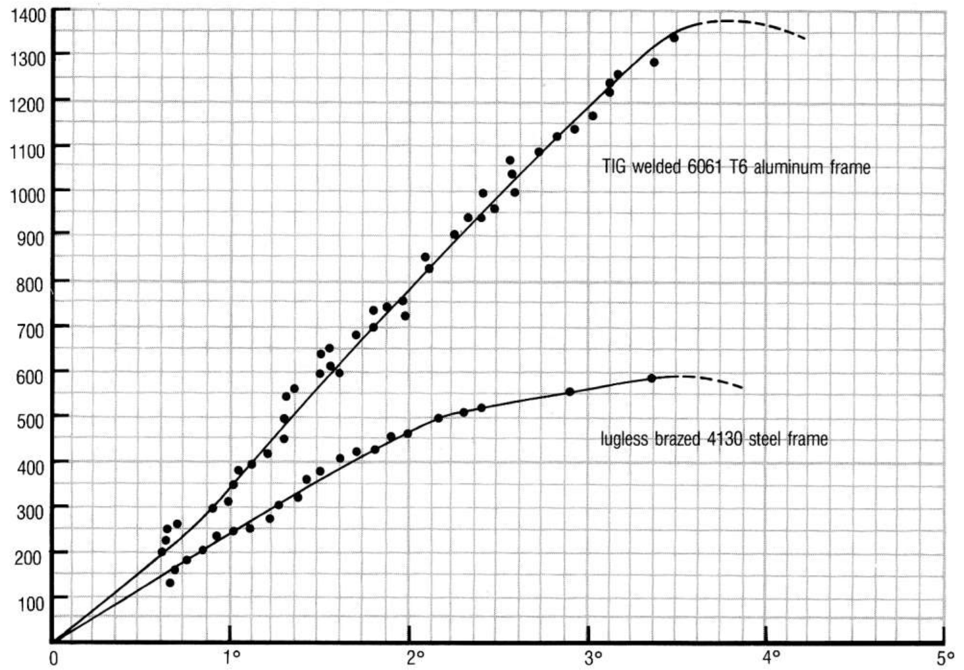


Figure 3: Frame Strength Tests

(also called "plastic deformation"). At last, at 850 pounds, the force gauge stopped showing any tension increase, so we stopped and Cunningham removed the frame for inspection.

(The yield point, or elastic limit, is easily defined by this event, at which the stress — which variès with the gauge reading — is no longer proportional to the strain — which varies with the position of the jack. As the jack is cranked, the reading on the force gauge stops climbing in proportion to the turns of the crank, and levels out. Instead of continuing to store the work of deformation elastically as a spring, something is bending permanently and will not spring back to its original shape.)

Surprise

Cunningham removed the frame from the test jig, and much to everyone's astonishment, it was the 1¹/₈-inch diameter solid steel dummy fork that had bent, and the aluminum frame remained unaffected! The addition of a welded brace to the "fork" enabled the test to proceed.

This time the gauge reached a reading of 1,325 pounds before the frame began to yield. We added additional force to determine where and how the frame would fail. The tension reached a final level of 1,380; after that the gauge reading rose no further (Figure 3). Interestingly, the down tube buckled at the point just behind the tapered down-tube-reinforcing lug, some 8 inches behind the head tube. The nature of the bend made it apparent that the stress distribution in this area could be further improved by slightly altering the shape of the lug. For example, future frames could feature a thinner tang on the lug's tip, which would better diffuse the stress.

Testing the elite 4130 chrome-moly tube frame took less than half as long. At 540 pounds the frame was bent irreversibly, behind the head tube, both at the top and down tubes. The wrinkles produced by the test were identical to those observed in crashed frames.

The aluminum frame showed a failure load 2.4 times as great as the steel frame (see graph). The implications of this result are far-reaching: imagine the sense of comfort derived from knowing that you could crash into 2.4 times as big a tree, or perhaps a tree of 2.4 times greater density, or, better yet, careen 2.4 times as fast into a tree, on an aluminum bicycle! In seriousness, though, there are a couple of points of interest beyond the strength itself:

In the steel frame it appeared that deformation of the tubes was quite localized in an area that abutted the brass fillet. While of course the bending moment is greatest here, this area also seems to be much softer than the rest of the tube, probably because the large brass fillet acts as a substantial thermal mass during brazing, causing slow cooling of the nearby tubing. This has a tempering (partial annealing) effect. Cunningham suspects that a T I G welded 4130 frame of the same tube dimensions would be slightly stronger, as the heat-affected zone would be smaller, and the cooling rate higher. Cunningham hopes to test such a frame, built by Scot Nicol of Walnut Creek. He is also working with Steve Potts to build an improved (lighter and stronger) brazed 4130 frame.

A second interesting point is that the ratio of failure strengths is much greater than would be predicted by theoretical figures.

Bending strength is usually evaluated as the product of section modulus (a property of the member's shape) and yield strength (a property of the material) (see Crispin Miller, "Tubing Rigidity," *Bike Tech*, August 1982).

The section moduli of the aluminum tubes in this test exceed those of the steel tubes, by a ratio of 3.18 for the top tubes, and of 2.02 for the down tubes (which are more important since, being larger, they carry more of the load, and bear higher stresses). However, the yield strength of aluminum is lower than that of steel, and this offsets the advantage of section modulus somewhat.

Exactly how much is hard to say because the strength of both metals depends strongly on their history of heat treatment. The yield strength of 6061-T6 is generally accepted as 40,000 psi;[2] the yield strength of 4130 steel in thin-walled tubing is less less well-defined, but for the tubing in this test it probably falls between 75,000 psi and 107,000 psi.[3]

Plugging in these values, the bending strength ratio (aluminum to steel) should be in the range of 1.19 to 1.70 for the top tubes, and in the range of 0.76 to 1.08 for the down tubes.

Even with the uncertainty of these values, therefore, it becomes clear that something else is going on — neither of these ratios approaches the factor of 2.4 found in the test.

Some of the discrepancy undoubtedly results from the tempering of the steel tubes: the after-brazing yield strength of 4130 can theoretically drop as low as 52,250 psi (if somehow the cooling is so slow that the steel is completely annealed).[4] This change would bring the theoretical ratio of strengths somewhat closer to the 2.4 value observed, but still would leave a significant disagreement, at least for the larger and more important down tubes.

[2] *See, for example, p. 2250,* Machinery's Handbook *(twenty-first edition), Industrial Press, Inc. New York, 1980.*

[3] *The steel frame was made from aircraft-grade 4130 tubing in the "normalized" condition, i. e., heated above transformation point and then air-cooled. For tubing in this size range (diameters above 1/2-inch and wall thickness below 0.188-inch), the yield strength is guaranteed to be at least 75,000 psi (information supplied by Tubesales, Inc., of Cherry Hill, New Jersey), but may be higher since it varies with cooling rate, which depends on wall thickness. 107,000 psi is probably a reasonable upper limit, being the yield strength advertised for Columbus tubing, whose composition meets 4130 specifications (see Mario Emiliani, "Straight Talk on Steel,"* Bicycling, *July 1982).*

[4] *Annealed yield strength from "ASM Data Book,"* published in *Metal Progress, v. 112 no. 1, mid-June 1977, American Society for Metals, Metals Park, Ohio.*

Another place to look for explanations is at the difference in relative wall thickness of the tubes. Thin-walled tubes often fail at loads below their theoretical strength, by "local buckling" — crumpling of the walls. This form of failure is very common in bent head joints of steel frames, and, as mentioned before, was the form of failure in the steel frame in this test. It may be that section modulus — which describes resistance to large-scale, *smooth* bending — is not particularly relevant to this test.

The steel frame had thickness-to-diameter ratios of 32.1 : 1 for the top tube and 25.5 : 1 for the down tube. The aluminum frame had a somewhat heavier ratio — 23.1 : 1 — for the top tube, but for the down tube the ratio was 28.0 : 1 — slightly lighter than the corresponding steel one.

However, there was an important modification to this tube: Cunningham reinforces the underside of his down tubes at the head joint, adding a second layer of aluminum which starts at the joint and extends down the tube seven inches.

This modification (pioneered by Cunningham) has two effects. To begin with, it increases the overall rigidity of the down tube, and, in the process, decreases the maximum stress to the underside of the tube. In the type of loading found in this test (which is by far the most common type of severe load on a bicycle frame), the stress in this location is the most important one: since both of the head joints must deflect through approximately the same angle when the fork is forced back, the strain (and associated stress) in the top and down tubes will be roughly proportional to their diameters, and therefore slightly worse in the down tube. And since local buckling is a compressive failure (when a tube is bent, only the compressed side can fail by wrinkling; the tensioned side simply stays straight and begins to stretch), the first modifications to a structure threatened by local buckling should be aimed at reducing the worst compressive stresses. So this strange-looking partial lug is exactly where it's needed: the compression side of the large tube.

The other effect, of course, is that the reinforcement greatly increases the *local* rigidity of the tube surface where it sits — by doubling the wall thickness, it multiplies the rigidity against local flexure (i.e., wrinkles) several times (up to eight fold, if the two layers are joined completely), so that local buckling is much less of a problem in the first place.

From the way the aluminum frame finally failed, this effect appears to be the dominant one. Even when the thickening piece has tapered to a narrow strip, whose stress-reducing effect is relatively small, it still prevents buckling — the buckling in the test occurred below the reinforcement, although the bending moment that far down the tube is only about half the moment at the head joint. (The relatively thick-walled top tube, incidentally, didn't fail at all; it broke its weld instead.)

Next?

Cunningham hopes that other frame-builders will submit their work (with identical geometries and tube lengths) for similar tests — such tests could improve everyone's understanding of frame design. (The prohibitive cost of *buying* frames for such tests rules out this method of acquisition — also, all that's really needed is the front triangle.) Those interested should write for details to Charles Cunningham at 121 Wood Lane, Fairfax, CA 94930, or call him at 415/457-1779.

Another test Cunningham intends to conduct is one on the relative strengths of forks. Ideally, forks should be a little less strong than the frames they're paired with, so that it is the fork and not the frame which gets munched in a crash. Cunningham has designed one such fork for off-road use, with replaceable blades.

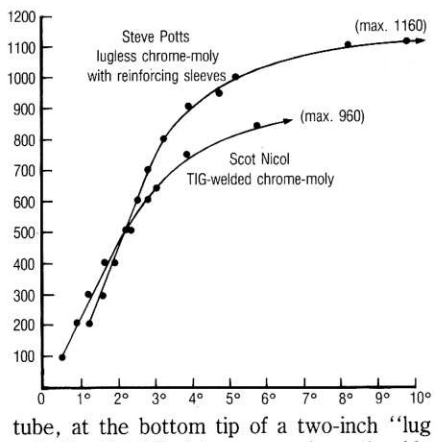
Jacquie Phelan is a writer and bicycle racer who works in Charles Cunningham's shop. She recently rode a Cunningham frame to win the women's division of the 1983 Rock-Hopper off-road race at Santa Rosa, California.

Two More Bite the Dust

As we prepared to go to press, Charles Cunningham telephoned to report tests on two more frames: a TIG-welded one from Scot Nicol, and a new Steve Potts prototype with reinforced joints.

The failure mode for these two new frames showed an interesting difference from the earlier two; while the new ones did show buckling at their final and complete stage of failure, they both failed much more gradually than the earlier ones, and during the long period of partial failure (when some plastic deformation had occurred, but the frame's resisting force would still increase as the jack was tightened), the tubes yielded in smooth curves instead of buckled kinks.

The TIG-welded frame (with the same tube diameters and gauges as in the brazed frame, and a front-triangle weight of 3.0 pounds) began to show a very slight yielding (0.3 degrees) at 500 pounds, but not a visible amount until 750 pounds. Under continued increases of displacement, its resisting force finally stopped increasing at 960 pounds (see diagram).

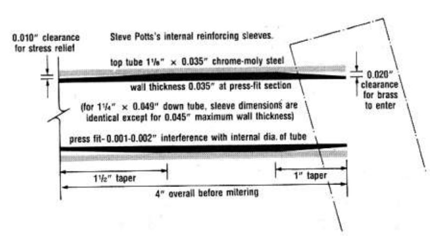


At failure, this frame showed broad, arching curves in both tubes, a crack in the upper side of the upper head joint, and a small amount of buckling underneath the down tube, at the bottom tip of a two-inch "lug point" which Nicol brazes on the underside of his down tubes (analogous to the reinforcement used by Cunningham). Cunningham concludes from the absence of buckling at either joint that the heat-treatment history resulting from TIG welding must leave the tubing harder near the joint than does that of brazing.

The new brazed frame was like the first one, except that its tubes were reinforced near the head joints with internal sleeves 4 inches long, tapered at the back ends to give a smooth transition to the single-thickness tubes beyond. To avoid tempering the tube *beyond* these inserts, the inserts were pressed in without brazing (except for the inch nearest the joint, where they were tapered to make room for the brass), so except right at the joint they served simply as unattached supports to steady the inside of the tube wall. The front-triangle weight with the inserts came to 3.25 pounds; i.e., the inserts added about two ounces.

The change in strength with the reinforcing inserts was quite dramatic. The first permanent deformation (0.9 degrees) occurred when the load was raised to 840 pounds; wrinkles began to show in the down tube at around 1,000 pounds; and the force finally stopped increasing at 1,160 pounds, when the upper side of the top tube parted audibly just behind the fillet. In spite of these local failures, most of the deformation occurred as smooth yielding of the overall shape of the tubes, producing curves more than 10 inches long, fairly even though slightly sharper around the back ends of the reinforcing inserts.

Crispin Mount Miller



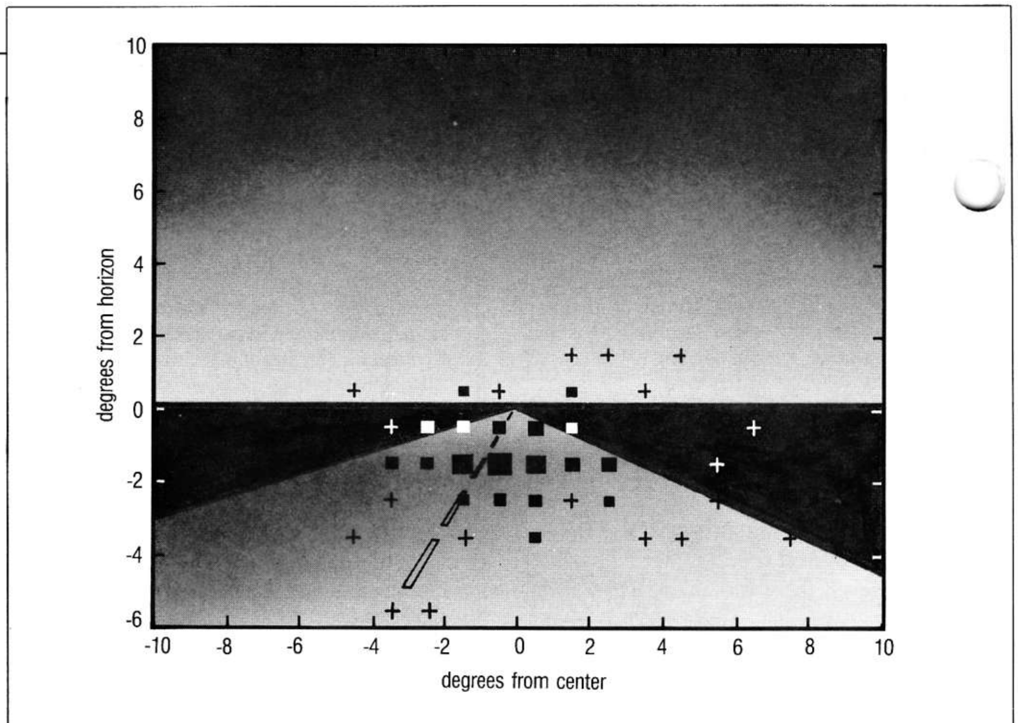
Developing Lighting and Reflectorization Standards

Fred DeLong with John S. Allen

Vision often works in deceptive ways. Because most of us associate vision so closely with reality, we often assume that when something comes in front of us, we automatically see it as soon as it's visible.

For cyclists in traffic this is a dangerous assumption. A cyclist may not be noticed until an approaching driver happens to look exactly in the cyclist's direction; the driver's eyes do not constantly focus on the cyclist's location on the road. As Dr. Helmut Zwahlen of Ohio University has shown, the eyes scan the road by an endless series of brief glances or "fixations," constantly shifting from side to side and up and down. Figure 1 shows the results of a typical test run, with the number of the driver's eye fixations totalled in areas of the visual field one degree on a side. The driver's eyes may fix on a reflector on a fencepost, a distant headlamp or taillamp, the center or edge of the road, or a sign. On a left curve, they fix more on the center of the road; on a right curve, on the road edge. And, of importance in this discussion, they may skip entirely past the bicyclist's position on the roadway.

Dr. Zwahlen has determined that the intensity of a lamp must be 1,000 times the threshold of perceptibility if the lamp is to gain the driver's attention 98 percent of the time. Figure 2 gives the source intensity in candelas necessary to achieve this level at various distances under fairly good conditions (not quite crystal clear but less than a light haze). As shown, 2 candelas are necessary to achieve this level at 800 feet; 0.75 candela is necessary at 500 feet, and 0.3 candela at 300 feet.



George Retseck

Figure 1: Driver fixation plot for night driving (taken on a straight level two-lane road at an average speed of 44 mph with low headlight beams). Average distance to fixation point, 117 feet; average fixation time duration, 0.46 sec. with a standard deviation of 0.34 sec. Rectangles' areas are proportional to percentage of time spent looking in each one-degree region; crosses indicate less than one percent.

Dr. Zwahlen's Recommendations

Based on light intensities and detection distances from research of which Figures 1 and 2 are an example, Dr. Zwahlen has put forward recommendations (shown in Table 1) which must be fulfilled to attain 1,000 times threshold illumination under fairly normal clear weather conditions.

United States Federal Highway Administration Report FHWA-RD-78-78 gives sight distances for highway design and traffic control. For the speeds listed here distances for the full process of avoiding an obstacle (detection, recognition, decision and response initiation, and maneuvering) are as follows. Dr. Zwahlen's recommended light intensities from Figure 2 are also shown.

speed	distance	light intensity
30 mph	450-625 feet	0.7-1 candela
50 mph	750-1025 feet	2-3 candelas
70 mph	1100-1450 feet	3-6 candelas

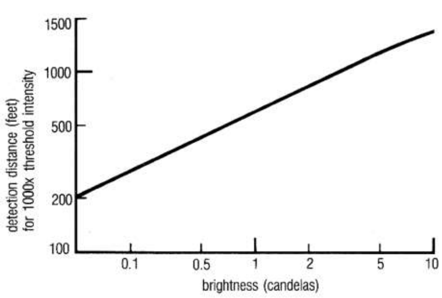


Figure 2: Detection distances for point light sources (for 98 percent chance of detection) as a function of source intensity.

(Figures 1 and 2 from Transportation Research Circular No. 229, from the Transportation Research Board of the National Academy of Science, Washington, DC.)

It should be recognized, however, that the factor of 1,000 times threshold visibility is not an absolute. As already stated, it represents a purely statistical choice — that a taillamp will draw the attention of 98 percent of drivers at a given distance. Additional factors can increase or decrease the detection distance. Those tending to decrease it are the driver's use of alcohol; problems with eyesight; background illumination — particularly, oncoming headlamps and reflections off wet pavement; and conditions which degrade the sight path, such as fog and dirty windshields.

Street lamps may help illuminate the bicyclist — yet street lamps beyond the bicyclist may help cancel out the illumination contrast that makes the bicyclist visible.

Standardized Patterns for Recognition

Other factors have a more certain effect in improving the detection distance. A bicyclist's reflectors and light-colored clothing can enhance the effect of a lamp — and often will be detected before the lamp under favorable conditions. Furthermore, the ability to take appropriate action is increased if the driver can *recognize* the bicyclist, estimate size and speed, and so be able to anticipate the bicyclist's maneuvers.

Unlike an automobile's two headlamps and taillamps, a bicyclist's single lamp or reflector does not fulfill this requirement; but light-colored or reflective clothing, pedal reflectors, and reflective tires give the bicyclist a recognizable "signature." Some foreign countries are exploring the use of spaced lights and/or reflectors in a standardized pattern to help increase this effect; many bicycles in Japan have two headlamps side by side, while Holland is considering two headlamps mounted one above the other as a way to identify a bicycle and avoid confusion with a more distant car.

Unreliability of Reflectors

Low beam automotive headlamps do not carry into the range of distances necessary to make standard bicycle reflectors (or light-colored clothing) meet Dr. Zwahlen's requirements at higher road speeds.

If the road is curved or dips and rises — or if the driver is about to cross the bicyclist's path at an intersection — the car's headlamps are not aimed at the bicyclist. And the light output of reflectors falls off drastically with an increasing *entrance angle* — that is, if they are not aimed squarely at the light source. "Wide-angle" front and rear reflectors as specified by the ISO reduce this problem for angles to the right and left — though not up and down. The orientation of "wide-angle" side reflectors mounted in the wheels changes as the wheels turn; they may not be effective at large entrance angles in the horizontal plane when the bicycle is stopped.

Also, the light from reflectors — or light-colored clothing — has had to traverse twice the path length of that from the bicyclist's lamp. The reduction in light reaching the observer due to distance, and to dust or mist in the air, is squared; at twice the distance under clear conditions, reflectors deliver one-sixteenth their original amount of light while a lamp delivers one-fourth of its.

In addition, at shorter distances, the angle between a vehicle's headlamps and the driver's eyes increases at the reflector. Since the principle of reflectors is to throw light back in the direction from which it came, this increasing *observation angle* can seriously degrade their brightness for drivers of large trucks and buses, who are often seated several feet above their headlamps.

ISO reflector specifications are at only two observation angles, 0.2 and 1.5 degrees. As shown in the table in the reflector specifications, the intensity at 1.5 degrees is permitted to be as low as 1/100 of that at 0.2 degrees; in fact, intensity may fall off precipitously just beyond 0.2 degrees. At a distance of 500 feet, 0.2 degrees is only 1.75 feet, far less than the distance between the headlights and line of sight of many truck or bus drivers — or of a car driver with a burned-out left headlight. Dr. Zwahlen is especially concerned about this failing.

And if the driver who must see the reflector is another bicyclist, there's an additional problem: a bicycle headlamp, weaker than automobile headlamps, may not be sufficient to illuminate the reflectors of another bicyclist ahead.

To be sure, most drivers are not inclined to spend much time traveling at speeds at which their headlamp beams are grossly inadequate to show up obstacles in the road ahead; many possible obstacles such as fallen tree branches, potholes, and pedestrians have no reflectors or lamps whatever. This does bring in a compensating safety factor for bicyclists when reflectors don't work — for example on hillcrests, on curves, and when the air is unclear. However, the "freak" failings of reflectors have been shown to be a direct cause of many serious accidents for bicyclists who mistakenly relied on them alone.

Lamps, of course, may fail too, electrically or mechanically, but the rider can tell when this happens. A taillamp — or a headlamp in its conspicuity role — is not subject to the unpredictable influences which can decrease the brightness of a reflector to unusable levels even when it is at a distance at which it would normally be effective. The ISO standard can therefore specify a required *brightness* for lamps, but due to the unpredictable influences only a required *reflectivity* for reflectors, which falls off with increasing entrance and observation angles.

A Strategy for the Bicyclist

For bicyclists, the best advice is to use both lights and reflectors which exceed the standards. Particularly when riding on roadways with high-speed traffic, hills, and curves, use bright lights and plenty of additional reflective material to increase the assurance that drivers will detect and recognize you in time to take appropriate action.

The ISO standards set a base level for bicycle conspicuity equipment; recent advances in available commercial products — for example the Varta taillamp, halogen bulbs, high-powered battery lamp systems, and well-designed reflective clothing and bicycle luggage — serve the needs of the bicyclist who is willing to pay for additional assurance when riding under demanding conditions.

Fred DeLong is a member of the American National Standards Association Technical Advisory Group to ISO TC 149, Bicycles.

Summary of ISO DIS 6742/2 Reflector Specifications

Compared with U.S. SAE and CPSC Standards

Comments by John S. Allen

Reflectors: Mechanical and Environmental Tests

Impact: Reflector must withstand a blow to the surface caused by dropping a 13 mm steel ball from a height of 760 mm.

Moisture Penetration: Shall withstand immersion 20 mm below the surface of water at 50° C for 10 minutes, followed by immersion in water at 25° C.

Mount Alignment: Shall withstand a 90 N (20 lb) pull in any of three directions which appear most likely to misalign it.

Saline Mist Resistance: No corrosion shall be evident that would affect the integrity of mounting or housings after two 24-hour 20 percent NaCl immersions separated by two-hour drying.

Resistance to Fuel: Soak for five minutes in 70 percent N-heptane and 30 percent toluol,

Table 1: Bicycle Taillamp Beam Intensity Measuring Points as Recommended by Dr. Helmut Zwahlen, 1982

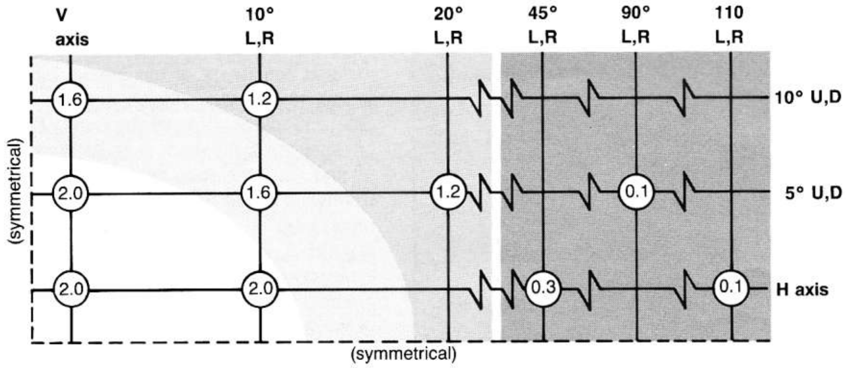


Table 1: Bicycle taillamp beam intensity as recommended by Dr. Helmut Zwahlen, 1982

Table: Coefficient of Reflectance

Coefficient of luminous intensity, millicandela/lux: Clear Reflectors

A. ISO DIS 6742/2 Reflectors

at entrance angles, in degrees	V axis	10° U,D	20° L,R	30° L,R	40° L,R	50° L,R
Observation angle 0.2 degree	2500	1650	850	750	650	550
1.5 degree	26	18	11	11	11	11

(Yellow reflectors 0.625 of the above, red 0.25)

B. United States Consumer Products Safety Commission Bicycle Reflectors

at entrance angles, in degrees	V axis	10° U,D	20° L,R	30° L,R	40° L,R	50° L,R
Observation angle 0.2 degrees	2500	1650	850	750	650	550
1.5 degrees	26	18	11	11	11	11

(Yellow reflectors 0.625 of the above, red 0.25)

C. SAE Automotive Reflectors, for comparison: SAE J594F

at entrance angles, in degrees	0	20° L,R	10° U,D
Observation angle 0.2 degree	1680	560	1120
1.5 degree	24	12	20

(Yellow reflectors 0.625 of the above, red 0.25)

Note: CPSC (United States Consumer Products Safety Commission) bicycle reflectors — except pedal reflectors — and the ISO bicycle reflectors, which are based on the CPSC standards, are required to be wide-angle reflectors; their surface must be divided into smaller panels which reflect light in different directions.

The CPSC omits lamps from its requirement (tempting uninformed riders to conclude mistakenly that wide-angle reflectors are a satisfactory substitute). The appropriateness of the ISO's adopting the CPSC's reflector standards is an open question, since the ISO does specify lamps.

There has been much controversy as to whether it is desirable to reduce the reflective area visible to a driver directly behind the bicyclist in order to provide reflectivity to a driver who is diagonally behind: the latter driver is rarely on a collision course, and almost certainly not if the bicyclist is maneuvering legally. However, the ISO or CPSC reflector is in fact required to be *brighter* directly rearward than the SAE reflector.

The actual performance depends on the quality of manufacture; a carefully manufactured SAE reflector can greatly exceed the SAE standard (sometimes by 500 percent), but it is harder for an ISO or CPSC reflector greatly to exceed its standard. The effects of aging are also likely to be greater on a reflector with a smaller area, which relies on precision rather than mere size for its performance; and the visible size of the reflector may also affect its conspicuity.

The SAE standard does not specify a size for a reflector, however (and neither do the CPSC or the ISO). Any size up to a rather large maximum (12 in^2 for the SAE reflector) is permissible as long as the net brightness of the returned light meets the standard. SAE reflectors are commonly available in two-and three-inch diameters. CPSC and ISO reflectors are required to be better sealed against water than SAE reflectors.

The ISO seems to have followed the CPSC in two mistakes: neglecting the ability of a rider's legs or of baggage to hide wheel reflectors, and ignoring that "wide-angle" wheel reflectors may not function as such when the bicycle is stopped, if the wheel is in the wrong position. (See the article "ISO's Bicycle Safety Standard: Just How Safe Is It" in *Bike Tech* Vol. 1, No. 3, October 1982.

Reflectors in the spokes also cause noticeable wheel imbalance, and pose a safety risk if they turn sideways in the wheel; an issue which the ISO does not seem to have addressed. Also, there is nothing in these standards specifying that items of equipment likely to hide reflectors or lamps on a bare bike must be provided with a fitting to reattach them where visible.

Reflective Tires (reflective band dia 670 mm)

Observation angle	at entrance angles, degrees -4	20	40	50
0.2 degree	811	710	469	141
1.5 degree	81	71	47	14

Pedal Reflectors (yellow)

Observation angle	at entrance angles, degrees H,V 0°	V±10° H 0°	V 0° H±20°
0.2 degree	450	350	175
1.5 degree	16.5	11.5	7.5

Note: While the ISO uses the unit of millicandelas per (incident) lux, most U.S. reflector performance specifications use candelas per (incident) footcandle. The conversion ratio is

$$92.9\ \frac{\text{millicandelas}}{\text{lux}} = 1\ \frac{\text{candela}}{\text{footcandle}}$$

then wash in water with detergent, and rinse; reflector must still meet structural and optical requirements.

Resistance to Lubricating Oil: Wipe with detergent motor oil and let sit for five minutes. Wash in water with detergent, and rinse; reflector must still meet structural and optical requirements.

Reflective Tires

Temperature Resistance: Shall withstand:
- 12 hr. at 65° C, ±5°, 5-15 percent relative humidity;
- 1 hr. at 23° C, ±5°, 40-60 percent relative humidity;
- 15 hr. at −20°C, ±5°.

Performance When Wet: After a one-minute soak, reflectance (coefficient of luminous intensity) must not be less than 50 percent of dry value.

Tape Adherence: After 30 minutes at 50° C, 30 minutes at 23° C, cut reflective strip with a sharp knife. Try to pull it loose with a force of one N (0.22 lb.) per mm width.

Abrasion Resistance: Scrub with wet wire brush. Brush must remove tire material as well as tape.

Glossary of Technical Terms

John S. Allen

Candela: The intensity of light in any direction from a light source of one candlepower, assuming that the light from this source is radiated uniformly in all directions. The candela may also be used as a unit of measurement for the intensity of light of a nonuniform source (such as a bicycle lamp) at specified angles. If a nonuniform source has an intensity of one candela at a given angle, this intensity is the same as that of a uniform one-candlepower source.

Candlepower: The total amount of light radiated (in all directions) by a standard light source more or less equivalent to an ordinary candle. This is an actual measurement of power, but with a correcting factor for the unequal sensitivity of the human eye at different wavelengths of light, corresponding to different colors. Beyond the ends of the visible spectrum, a source may be very powerful and yet have zero output in candlepower (though it might give you sunburn if it is an ultraviolet source, or radiate noticeable warmth if an infrared source). Note that the candlepower (and the candela) define the amount of light in terms of *angles*. At a greater distance, a given solid angle covers a larger area, and a one-candlepower source will appear dimmer.

Detection: A driver's noticing an object in the visual field (such as a bicyclist's lamp) without necessarily being able to identify it.

Entrance Angle: The angle from which light strikes a reflector (0° is defined as perpendicular to the plane of the reflector, with light striking it squarely).

Footcandle (not used here, but included for the benefit of those familiar with it): Lumens per square foot — the English-unit counterpart of the lux. For a one-candlepower source, the intensity of illumination on a surface one foot away is one footcandle. One footcandle equals 10.76 lux.

Halogen Bulb: An incandescent filament lamp bulb inside which the gaseous mixture includes halogens (chlorine, bromine, flourine) which serve to redeposit evaporated metal back onto the filament. A halogen bulb burns hotter while maintaining an acceptable lifetime; also, metal does not deposit on the sides of the bulb and darken them. For both reasons, a halogen bulb gives more light. However, a halogen bulb is operating closer to the melting temperature of the tungsten filament, and so is more vulnerable to overvoltage conditions.

Lumen: $^1/_{4\pi}$ candlepower: the total light radiated into an area of one square meter on the inside of a hollow sphere of one meter's radius, at the center of which there is a uniform one-candlepower light source. Don't let the "one square meter" confuse you; this is still a measurement in terms of angle. At a distance of two meters, for example, the one lumen will be spread out over four square meters.

Lux: Lumens per square meter. This is the unit of light intensity in terms of *area*. In the example just given for lumens, the intensity of illumination at 1 meter from the source is 1 lux, and at 2 meters, $^1/_4$ lux, in keeping with the inverse square law. To specify the performance of a retroreflector, for example, the incident light must be measured in lux or footcandles in order to get an equivalent result regardless of the distance of the light source.

Millicandela: One one-thousandth of a candela.

Millicandela per Lux: A measurement of the amount of light reflected at a given angle from an object, when light with an intensity of 1 lux is impinging on the surface. This figure will increase directly with the size of the object and with its reflective efficiency. A diffusely reflecting, perfectly white square surface 7 cm on a side — about the size of a common retroreflector — has an intensity of approximately 1.6 millicandela per lux when viewed broadside. Retroflectors improve on this figure by concentrating light in the direction from which it came.

Observation Angle: The angle between a driver's headlights and the driver's eyes, from the position of the bicyclist's reflector. The greater the observation angle, the more poorly the reflector performs.

Recognition: A driver's identifying an object, for example a bicyclist. Recognition can occur at the same time as detection or later.

LETTERS

Tire Rolling — A Postscript

To my considerable embarrassment, I must admit that a number of poorly controlled variables significantly biased the data in my recent article "Rolling Resistance of Bicycle Tires" (*Bike Tech*, April 1983). Also there are a number of minor points which did not come across properly in the text or the illustrations. Allow me to submit the following summary of the major corrections.

1) The illustration Figure 5, which shows the test setup, shows the setup as designed, but differs from the actual setup used in the tests for table 2: the actual height of the highest point of the ramp was not 60 centimeters as shown, but 44 centimeters (approximately 17 inches) (dictated by the height of the concrete ridge on which I leaned the board) with a slope length of 2.00 meters (6 feet - 7 inches approximately). This brought the contact point between the rear tire and the ramp to a height of 37 cm above the level surface.

2) There may have been a slight (less than 1 percent) up-slope in the case of the rough-surface tests, which makes the differences between the rough and the smooth surface tests appear more pronounced than they should actually have been. In addition, the description "surface reminiscent of an asphalt road due for resurfacing" probably suggests a less coarse surface than appropriate. A more accurate description might have been "coarse and heavily pitted concrete."

3) The inflation pressure of 6 bar corresponds to an *absolute* pressure of 88 *psia*, which is 63 *psig gauge* pressure, not 88, as might be assumed from the reference in the text. In addition, the pressures of the tires with butyl tubes had to be measured with a different gauge (because it had a different valve type, which did not match the precision gauge for the Presta valves used on the tubulars and lighter tubes).

4) The shortest coasting distance (worst case for roughest surface, as measured for a Metzler tire with Semperit thornproof butyl tube) was measured to be 12.5 meters, which is the value quoted (roughly) in the text as though it applied for the mean value for three runs under these conditions. In fact, the mean value for three runs was somewhat higher, namely 13.5 meters. I had based my table on multiples of the absolute shortest run, whereas the editor (with my permission) converted this table to express coasting distances in multiples of the worst mean value, forgetting to correct the reference to the shortest distance in the text.

5) The illustration Figure 9, which shows the response of different tires to depression, is somewhat inaccurate, in that it appears to indicate that only the sidewall deflects. As a matter of fact, tread flexibility seems to be equally significant, as shown in Figure 10.

In summary, this was a suitable test, poorly controlled, leading to results which, though directionally correct, seemed more dramatic than actually appropriate. Being poorly equipped for quantitative testing, I did not set out to produce a quantitative test: I merely wanted to establish reasonable criteria for tire selection and to "explain" rolling resistance as a function of both road surface quality, tire pressure, and tire construction. I believe the conclusions to be valid in the direction of the differences they show, though I must warn the reader not to attach great value to my figures. I should also mention here that the comparative tests for tire construction variables took place under better controlled circumstances, namely on a (very) roughly surfaced sidewalk area near my home, each time using the same tube and the same inflation pressure, during no-wind conditions, with the same rider and — very time consuming — the same wheel for each run. Under these conditions, the coasting length (always coasting in the same direction, to compensate for any slope in the surface) varied from 29 meters (97 feet) to 36 meters (120 feet) from a 52 centimeter high ramp. On a smooth surface the differences between these different tires would have been less pronounced. But keep in mind: these figures are useful only for comparison — not for quantitative use!

Rob Van der Plas
Frankfurt, Germany

Spoke Tension — A Constant Climb?

I found Eric Hjertberg's article on spoke tension interesting ("How Tight Is Right?", *Bike Tech*, April 1983), but I found the diagram surprising.

The diagram (spoke tension versus turns of the nipples) appears to resemble a third-power function. Remembering Hooke's law from school (i. e., stress is proportional to strain), I would have expected a more-or-less straight line.

To examine this question I conducted some spoke-tension tests with my associates here. We used the following components:

20-mm wide Endrick steel rims (made of 0.7-mm steel sheet folded double and welded in the center) with 4.4-mm spoke holes, with no socket or ferrules.

A freewheel hub with the holes in its flange simply punched (not countersunk).

A coaster-brake hub with flange holes countersunk on both sides.

Spokes with a bend angle of 105 degrees and a head angle of 90 degrees, and a diameter of 1.8 mm for the freewheel hub and 2.0 mm for the coaster brake.

Our procedure was to build the wheel and tension all the spokes to a nipple torque of 5-7 kg-cm (4-6 inch-pounds), and then to loosen them one at a time, oil the thread and nipple seat, and measure tension every quarter-turn as they were retightened. Since we don't have a Hozan spoke tensiometer, we used a Snap-On torque wrench and calculated tension from nipple torque. For both wheel types, our results indicate a very nearly constant ratio of tension to number of turns; the diagram shows our data for the freewheel-hubbed wheel.

In addition to the numerical results, we noted the following effects:

None of the tested spoke nipples deformed the rim measurably around the nipple holes, and there was no deformation visible in the chrome-plated surface. The slight burr from punching was flattened during the initial tensioning and no further deformation was noticeable.

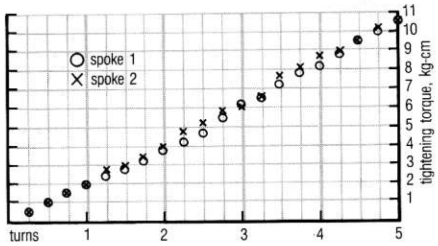
On the coaster-brake hubs no marks were visible around or in the spoke holes after the test. On the freewheel hub there was no deformation noticeable on the side occupied by the spoke head, but on the side where the bend rested the spoke deformed the sharp edge of the hole to a rounded edge, with a radius of curvature of about 1 millimeter.

We stopped testing at about five turns because the slot in the nipple head would strip as the torque increased, but we already had the information we wanted. By my calculations we would have reached the yield stress of the spokes (assuming it to be 85-90 kp/mm^2 — 120,000-130,000 psi) very soon afterward.

While this is not an extensive investigation, it seems to me to give clear results. If anyone wants to repeat it, it would take only about two hours' work for two persons.

I work at Roma Ind. Mec. S/A, where we strive to make high-quality bicycles.

Sven Wildheim
Sao Paulo, Brazil



Tension (expressed as tightening torque) versus number of turns, for 1.8-millimeter spoke in freewheel hub.

Eric Hjertberg: These results are indeed an elegant demonstration of a linear relationship between spoke tension and elongation. The components in this test were quite different from the ones we generally use, but we assume that the same principles apply, and that the plotted results are different because the testing procedures were different.

The test described here was done on spokes that had already been fully tightened, so that any plastic deformations (seating of parts against one another, and perhaps minor bending of spoke ends and rim surface) had already taken place.

In our test, on the other hand, the spokes were being tightened for the first time. Some of the initial turns, then, were taking up distance yielded by these initial deformations, and the tension in the spoke didn't begin to rise at a steady rapid rate until these deformations had "bottomed out." The upper portion of our graph is nearly a straight line, which we consider to be consistent with this new test.

We presented the graph as we did because we were concerned with the behavior of spoke tension from the wheelbuilder's point of view, rather than with the physical theory behind it. In particular, the transition from the initial slow increase to the later rapid increase takes many new builders by surprise, and we wanted to call attention to it.

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Bike Tech and Bicycling to Increase ISO Representation

On April 19, 1983, John S. Allen, Editor-at-Large of *Bike Tech* and Contributing Editor to *Bicycling*, attended a meeting of the American National Standards Institute Technical Advisory Group to the International Standards Association Technical Committee 149 (ANSI TAG to ISO TC 149). Pending a vote of the TAG, Mr. Allen will become a member of the TAG. *Bike Tech* and *Bicycling* have also agreed to make a monetary contribution to the TAG, which is funded by its members. Fred DeLong, Contributing Editor to *Bike Tech* and Bicycling, was unable to attend the April 19 meeting but also continues as a member of the TAG.

Bike Tech will have continuing coverage of the ISO's work by Mr. Allen and Mr. DeLong: for example, the article on lighting and reflectorization in this issue.

Let Us Hear

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

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