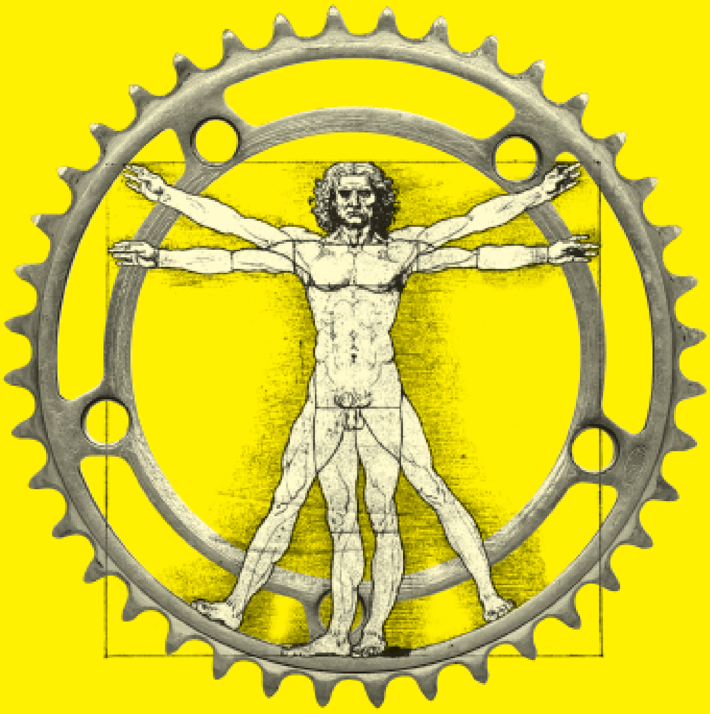

# HUMAN POWER

## TECHNICAL JOURNAL OF THE IHPVA

***NUMBER 51 FALL 2001***



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**Editor**
David Gordon Wilson
21 Winthrop Street
Winchester, MA 01890-2851 USA
dgwilson@mediaone.net

**Associate editors**
Toshio Kataoka, Japan
1-7-2-818 Hiranomiya-Machi
Hirano-ku, Osaka-shi, Japan 547-0046
HQI04553@niftyserve.ne.jp

Theodor Schmidt, Europe
Ortbühlweg 44
CH-3612 Steffisburg, Switzerland
tschmidt@mus.ch

Philip Thiel, Watercraft
4720 - 7th Avenue, NE
Seattle, WA 98105 USA

**Production**
JS Design & JW Stephens

**IHPVA**
Paul MacCready, Honorary president
Chris Broome, USA, Chair
Ben Wichers Schreuer, The Netherlands, Vice-chair,
Open, Secretary/treasurer

**Publisher**
IHPVA
PO Box 1307
San Luis Obispo, CA 93406-1307 USA
Phone: +805-545-9003; hp@ihpva.org

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### *ERRATUM*

In *HP* 50, p. 11, figure 3, the labels for the two lower lines were inadvertently reversed. The lowest line is that for the Ritchey OCR rim, and the middle line is for the Bontrager. Apologies to Vernon Forbes and to these manufacturers.

### *IN THIS ISSUE*

**Optimum crank-arm length for recumbents**

Danny Too and Chris Williams tested nineteen subjects using the recumbent-seating position found in earlier studies to permit maximum power output to be developed. Each person pedaled at maximum effort using, in turn, five different crank lengths. (One subject produced over 1.1 kW). The recommendations for the best lengths of cranks for different races are bound to be followed closely.

**Bicycle pitchovers**

Fred Matteson is concerned for the safety of bicyclists, particularly when braking on steep descents. His analysis has produced a graph on which each rider can enter her/his body and bike characteristics, and thereby learn on which hills her/his level of braking can be critical.

**CdA and Crr measurements**

John Snyder has developed two methods of measuring one's coefficients of aerodynamic drag and of rolling drag. The first uses two hills of different slope but similar surface, and the terminal coasting velocities and other easily measured data give the coefficients. The second method involves one hill, and two coasting runs down the hill, in one case with a drogue chute. John gives all instructions.

**Technical notes**

*Chain-drive efficiency.* Claire Walton and John Walton have analyzed the Spicer data from the last issue of HP, and have shown graphically how increased chain tension increases transmission efficiency.

*Improvements in chain-loss measurements?* In another technical note on the Spicer data, John Allen suggests a feedback system for the driver torquemeter from the driven torque, so avoiding the inexactness of measuring the difference between two similar quantities.

*Bicycle stability after front-tire deflation.* Your editor reports on studies by Andy Oury and others on tires that produce instabilities when they go flat. The prime recommendation is that standards of tire-to-rim fits be promulgated.

**Book reviews**

*The Athenian Trireme*, a book on a human-powered warship of several-hundred years BC, is reviewed enthusiastically by Theo Schmidt.

*The Dancing Chain*, by Frank Berto, Ron Shepherd and Raymond Henry, is a wonderful compendium of derailleur gears from the earliest to the latest times, favorably reviewed by your editor.

*Bicycle Design* by Mike Burrows is another placed in the "must read" category by your editor.

**Editorials**

Joachim Fuchs contributes a guest editorial on velomobiles.

Your editor writes a sad "farewell and thank you" to Open Road, publisher of *Encycleopedia* and *Bike Culture Quarterly*, among other notable productions. I also write somewhat angrily, again, about tires.

—*Dave Wilson*

### CONTRIBUTIONS TO *HUMAN POWER*

The editor and associate editors (you may choose with whom to correspond) welcome contributions to *Human Power*. They should be of long-term technical interest (notices and reports of meetings, results of races and record attempts and articles in the style of "Building my HPV" should be sent to *HPV News*). Contributions should be understandable by any English-speaker in any part of the world: units should be in S.I. (with local units optional), and the use of local expressions such as "two-by-fours" should be either avoided or explained. Ask the editor for the contributor's guide (available in paper, e-mail and pdf formats). Many contributions are sent out for review by specialists. Alas! We cannot pay for contributions. They are, however, extremely valuable for the growth of the human-power movement. Contributions include papers, articles, reviews and letters. We welcome all types of contributions, from IHPVA members and nonmembers.



# Determination of the crank-arm length to maximize power production in recumbent-cycle ergometry

*Danny Too and Chris Williams*

**ABSTRACT**

The purpose of this study was to determine the crank-arm length that would maximize peak, mean and minimum power outputs in a recumbent cycling position. Nineteen male volunteers were each tested with five pedal-crank-arm lengths (110, 145, 180, 230 and 265 mm) according to a randomized sequence on a free-weight Monark cycle ergometer. The 30-second Wingate Anaerobic Cycling test was performed in a recumbent position (75° seat-tube angle, backrest perpendicular to the ground) against a resistance of 85 g/kg of the subject's body mass (5.0 J/crank rev/kg BM). Curve estimation with regression analysis revealed that the crank-arm lengths to maximize peak power, mean power and minimum power are 124 mm, 175 mm and 215 mm, respectively.

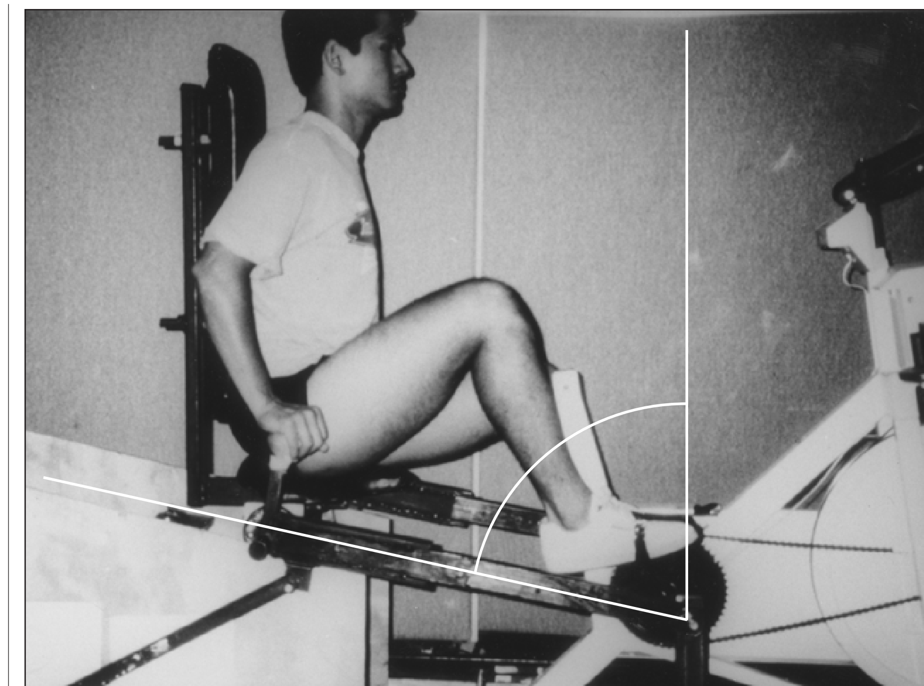

Figure 1. Recumbent position with a 75 degree seat-tube angle

**INTRODUCTION**

It is well documented that recumbent human-powered vehicles with aerodynamic fairings, having a smaller drag coefficient and cross-sectional area, are faster than the standard racing bicycle (Kyle, 1982). However, with the current speed record of 117.06 km/hr 72.74 mph), established in 2000 by a single rider (Sam Witting -ham) on a Varna recumbent bicycle "Mephisto", designed and built by Georgi Georgiev, it becomes questionable whether a more aerodynamically effective human-powered vehicle can be designed. If future speed records are to be attained, it is necessary to focus not only on the aerodynamics, but also to examine the variables that affect power production in recumbent cycling and the interactions that would maximize it. Investigations in this area of recumbent cycling and power production have included an examination of changes in seat-tube angle (Too, 1991) and trunk/backrest angle (Too, 1994).

Too (1991), examining a systematic change in seat-tube angle (0°, 25°, 50°, 75° and 100°), reported the largest peak power and mean power to be found with the 75° seat-tube angle and a parabolic curve (quadratic trend) best describing the change in peak power and mean power with changing seat-tube angles. Seat-tube angle was defined by the angle formed between the seat tube and a vertical line (perpendicular to the ground) passing through the crank spindle. Using a 75° seat-tube angle, Too (1994) investigated the effect of three trunk/seat-backrest angles (60°, 90° and 120°) on power production. A parabolic trend in peak power and mean power was found with changes in trunk/seat-backrest angle, with the largest peak power and mean power reported using the 90° trunk angle.

Based on muscle force-length and force-velocity power relationships, changes in crank-arm length will affect joint angles, muscle length, force, torque and power production in cycling. Since the literature involving traditional upright cycling positions have reported an effect on power output with changes in crank-arm length (Hull & Gonzalez, 1988; Inbar, Dotan, Trousil & Dvir, 1983; Too & Landwer, 2000), it can be assumed that power production will also be affected in a recumbent cycling position with different crank-arm lengths. Therefore the purpose of this study was to determine the trend in power production with changes in crank-arm length, and the crank-arm length that would maximize peak power, mean power and minimum power in a recumbent cycling position.

**METHOD**

Nineteen healthy volunteer male participants (mean age = 24.8 ± 4.4 yr., weight = 81.76 ± 11.84 kg, height = 1.80 ± 0.08 m) subjects were tested with a free-weight Monark cycle ergometer (Model 814E) at five pedal-crank-arm lengths (110, 145, 180, 230 and 265 mm), as defined by the distance between the center of the crank spindle and pedal spindle. (The normal crank-arm length for a Monark cycle ergometer is 170 mm). To accomplish this, two adjustable crank arms allowing for manipulations from 0 to 300 mm were used (Too & Landwer, 2000). All subjects were

| | | Crank-arm length (mm) | | | | |
|---|---|---|---|---|---|---|
| | | 110 | 145 | 180 | 230 | 265 |
| PP(W) | M | 1139 | 1144 | 1097 | 1025 | 916 |
| | SD | 206 | 214 | 223 | 193 | 167 |
| MP(W) | M | 802 | 845 | 845 | 819 | 762 |
| | SD | 177 | 192 | 168 | 166 | 134 |
| MINP(W) | M | 555 | 598 | 619 | 627 | 612 |
| | SD | 141 | 156 | 145 | 148 | 117 |
| MAXPED(rpm) | M | 174.1 | 171.7 | 167.5 | 153.1 | 135.2 |
| | SD | 11.7 | 10.7 | 12.4 | 11.1 | 11.3 |
| MINPED(rpm) | M | 82.0 | 88.4 | 91.9 | 92.7 | 91.4 |
| | SD | 12.7 | 12.2 | 12.0 | 12.3 | 9.1 |

PP = peak power; MP = mean power; MINP = minimum power
MAXPED = maximum pedaling rate;
MINPED = minimum pedaling rate

**Table 1. Peak power, mean power, minimum power, maximum and minimum pedaling rate with changes in crank arm length**

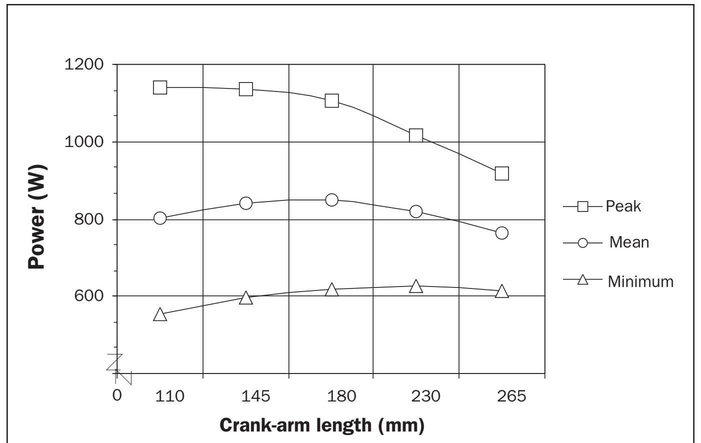


**Figure 2. Predicted peak power, mean power and minimum power with increasing crank-arm length.**

tested in each of the five pedal-crank-arm-length conditions, with the order of testing randomly assigned. There was a minimum of 24 hours of recovery between test sessions. For each condition, pedal toe clips were worn, and the subject was strapped to the seating apparatus at the hip and trunk.

The recumbent cycling position used for all test sessions, was defined by a 75°angle formed between the bicycle seat tube and a vertical line passing through the crank spindle (see figure 1; Too, 1991). To obtain this seating position, a variable seating apparatus, allowing for manipulations in seat-tube angle, backrest angle and seat-to-pedal distance was used and interfaced to a Monark cycle ergometer (Model 814E). The seat backrest was kept perpendicular to the ground and the seat-to-pedal distance adjusted to 100% of the total leg length of each subject, as measured from the right femur to the ground (Too, 1991). The test protocol involved a computerized 30-second Wingate Anaerobic Cycling Test. To initiate the test, the subject pedaled the cycle ergometer with no load. Once the ergometer's inertial resistance had been overcome, the appropriate load (85 g/kg of the subject's body mass) was instantaneously applied using calibration weights, and the subject pedaled as hard and as fast as possible for 30 seconds. A Sports Medicine Industry (SMI) opto-sensor (Model 2000) with a sampling rate of 50 Hz, interfaced with a Zenith 386 micro-computer, in conjunction with 16 reflective markers on the ergometer flywheel, was used to monitor and record flywheel revolutions during the test. Peak power was calculated from the highest average flywheel speed during any consecutive five seconds, mean power was determined from the mean flywheel speed for the entire 30-second test, and minimum power was calculated from the lowest mean flywheel speed during any consecutive five seconds (which was always the last five seconds). The different power variables were calculated using the following equation:

Peak power (watts) = [load (N)] × [distance covered by flywheel with one revolution (1.615 meters per revolution) × average number of recorded flywheel revolutions for five seconds (rpm)]/[1 min/60 sec].

Additionally, maximum and minimum pedaling rates were calculated from flywheel speed recorded for peak power and minimum power, respectively. The equation used in this calculation was:

Pedaling rate (rpm) = average flywheel rpm for five seconds / 3.7 flywheel revolution per pedal-crank revolution (Gledhill and Jamnik, 1995).

This would be equivalent to a 52/14 gear ratio. Curve estimation with regression analysis was used to determine: (1) the trend in peak power, mean power and minimum power with changes in crank-arm length; and (2) the crank-arm length that would maximize peak power, mean power and minimum power during a 30-second test.

## Results

With changes in crank-arm lengths, the mean ± SD values of peak power, mean power, minimum power, maximum and minimum pedaling rates are presented in table 1.

Based on regression analysis the change in peak power, mean power and minimum power with increasing crank-arm length, appears to be best described by a parabolic curve, represented by the equation: $y = -x^2 + x + C$ (where y represents power and x represents crank-arm length) as shown in figure 2. The specific regression equations for the various measures of power were as follows:

Peak power (quadratic trend, $p = 0.006$): $y = -0.011x^2 + 2.8x + 972$ (SE = 11)

Mean power (quadratic trend, $p = 0.011$): $y = -0.011x^2 + 3.8x + 513$ (SE = 5)

Minimum power (quadratic trend, $p = 0.002$): $y = -0.007x^2 + 2.8x - 325$ (SE = 2).

From table 1, several observations can be made: (1) regardless of crank-arm length, peak power is greater than mean power, and mean power is greater than minimum power; (2) peak power is greatest with the 145-mm crank-arm length and least with the 265-mm crank-arm length; (3) mean power is greatest with the 145- and 180-mm crank-arm lengths and least with the 265-mm crank-arm length; (4) minimum power is greatest with the 230-mm and least with the 110-mm crank-arm length; and (5) maximal



and minimal pedaling rates occur with the 110-mm crank-arm length. From regression equations, the predicted crank-arm lengths to maximize peak power, mean power and minimum power are 124 mm, 175 mm and 215 mm, respectively.

## Discussion

Since no literature could be found examining the effect of changes in crank-arm length on cycling performance in a recumbent position, comparisons will be made with the literature available for an upright position. The parabolic curve observed in peak power and mean power with increasing crank-arm length is consistent with the trend for an upright position reported by: (1) Inbar *et al.* (1983) for five crank-arm lengths (125, 150, 175, 200 and 225 mm); and (2) Too and Landwer (2000) for five crank-arm lengths (110, 145, 180, 230 and 265 mm). From best-fitting parabolic curves, Inbar *et al.* (1983) described the peak power and mean power to occur at a crank-arm length of 166 mm and 164 mm, respectively; whereas Too and Landwer (2000) predicted peak power and mean power to be maximized with crank-arm lengths of 164 and 200 mm, respectively. This is quite in contrast with the predicted crank-arm lengths (124 and 175 mm) to maximize peak power and mean power, respectively, for a recumbent position.

The largest peak power (762.7 W) and mean power (615.9 W) values reported by Inbar *et al.* (1983), and those reported by Too and Landwer (2000; largest peak power and mean power values to be 968 W and 718 W, respectively) are less than the largest peak power (1144 W) and mean power (845 W) values recorded for the recumbent position in this investigation. In fact, except for the 265 mm crank-arm length condition, peak power values (and all mean power values) in the recumbent position were greater than the largest peak and mean power values reported by Inbar et al (1983) and by Too and Landwer (2000) for an upright position. The smaller peak power and mean power values reported by Inbar *et al.* (1983) may be attributed to a smaller load used (75 g/kg body mass) and/or to the different stature of the subjects tested (approximately 10.5 kg smaller, 73 mm shorter than the subjects of this investigation). However, the smaller peak and mean power values reported by Too and Landwer (2000) are probably attributed to differences in lower-limb joint angles (between an upright and recumbent position) and/or to a smaller force production potential in an upright position (since there is no seat-backrest to push against).

Based on the predicted crank-arm lengths to maximize the different power variables, and the trend of peak power, mean power and minimum power with changes in crank-arm length, it would appear that an interaction exists between crank-arm length and power production, with the optimal crank-arm length to maximize power dependent on load and pedaling rate. Since power is a function of both force and velocity, the optimal crank-arm length to maximize peak power would be one where the maximum pedaling rate is produced and maintained with the largest load that can be applied. Although manipulation of load was not examined in this investigation, changes in crank-arm length would alter the torque on the crank arm (when the same force is applied) and would be analogous to a change in load. Based on the force-velocity relationship, a longer crank-arm length resulting in a lower "load" experienced by the lower limbs will result in a greater linear velocity at the pedal (when compared to the same pedaling rate with a shorter crank-arm length). This was confirmed when the maximal pedaling rates determined for the different crank-arm lengths of this investigation were converted to maximal linear pedal velocity. The maximal linear pedal velocity was found to increase (although the maximal pedaling rate decreased) with increasing crank-arm lengths from 110 to 265 mm. Similarly, an increase in crank-arm length from 110 to 265 mm also resulted in an increase in minimum linear pedal velocity (as determined from the minimum pedaling rates) and is also consistent with that expected from force-velocity relationships. Since parabolic curves in power were observed with increasing crank-arm lengths, and the largest values for peak, mean and minimum power were found with three different crank-arm lengths, this would indicate that the optimal crank-arm length to maximize power is dependent on the type of power examined.

In this investigation, the optimal crank-arm lengths predicted to maximize peak, mean and minimum power with a load of 85 g/kg BM, were 124 mm, 175 mm and 215 mm, respectively. The interaction between crank-arm length, pedaling rates and load (as evidenced by parabolic curves for power), would suggest that the optimal crank-arm length for peak, mean and minimum power would change with different loads. Based on the force-velocity-power relationship, increased loads to maximize power, resulting in a decreased pedal rate would favor longer crank-arm lengths.

Changes in crank-arm length will affect not only the force-velocity-power relationship, but also the muscle force-length relationship. From the force-length curve, a muscle can produce its largest force at resting length, with a decrement in force at increasing or decreasing lengths. Systematic increments in crank-arm length (from 110 to 265 mm) for an upright cycling position have been reported to result in significant decrements in minimum hip and knee angle, and significant increments in hip and knee range of motion (Too and Landwer, 2000). Whether it is more advantageous to use a long crank arm or a short crank arm is unknown because there is a complex interaction among changes in joint angles, muscle length and muscle-moment-arm length to produce force and torque with changes in crank-arm length. This complexity is further increased when multi-joint muscles that cross the hip and knee, or knee and ankle are involved and interact with force-velocity-power relationships. Additional research into the interaction of crank-arm length, pedaling rate and load on power production is needed before the limits of performance in human-powered vehicles can be reached.

**Summary**

The predicted crank-arm lengths to maximize peak power, mean power and minimum power in a recumbent cycling position, using a resistance load of 85 g/kg body mass, were 124 mm, 175 mm and 215 mm, respectively. This would suggest that for human-powered vehicle competitions of short duration, where maximal peak power is necessary, a shorter crank-arm length is recommended. For competitions of longer duration where fatigue is a factor and the largest mean power and minimum power become important, it is suggested that longer crank-arm lengths be used.

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**About the author**

Danny Too is an associate professor in the Department of Physical Education and Sport at the State University of New York at Brockport and has been involved in human-powered-vehicle research since 1985.

*—Danny Too*
*State University of NY at Brockport*
*350 New Campus Drive*
*Brockport, NY 14420 USA*
*E-mail: <dtoo@brockport.edu>*

# Bicycle pitchover characteristics

***by Frederick H. Matteson***

**Summary**

Pitchover is explained and a graph developed showing boundaries versus slopes. Situations and road characteristics are discussed.

**Introduction**

Pitchovers, wherein the bicycle and rider rotate forward about the front wheel, have been a problem since the early days of cycling when the high-wheel, direct-drive bicycle, commonly referred to as the "Ordinary" and later, derisively, as the "Penny Farthing", was used. The position of the rider, high and forward with respect to the front axle, made these cycles likely to pitch forward particularly in descents with braking. It was this danger that led to the development of the chain-driven "safety bicycle" still in use today. Today's bicycles, with the rider well back between the wheels, are far safer, but pitchovers can and do still occur.

This article concerns the matter of hills. That steep hills are an expectation of touring cyclists is evident by the installation of triple chainwheels and wide-range gearing on touring machines. Before the time of the automobile steep roads were common in this country. Horses could climb steep hills, but cars had limited climbing ability. The author recalls seeing Ford Model T's stop at the bottom of a hill, turn around and back up because they could climb a steeper hill in reverse gear. Such a practice was not reasonable and the trend has been towards less-steep public roads in the United States. The process of building safe, high-speed roads has consisted of straightening and leveling, often at great expense and difficulty. Abroad, and in particular in lesser-developed lands or where there are fewer automobiles, even main roads may be unsealed, crooked and containing steep slopes. Safety features common in the USA may not be present. Signs, striping, guard rails and signals may be absent. Dangers may appear without warning and such hazards as very steep local inclines, improper banking, holes or damage may suddenly appear. Very narrow roads are common in mountainous areas, sometimes wide enough for only one vehicle. These dangers may well result in the cyclist having to use sudden and strong braking. Braking on steep descents can easily lead to a pitchover accident and this will be discussed as the main subject of this article.

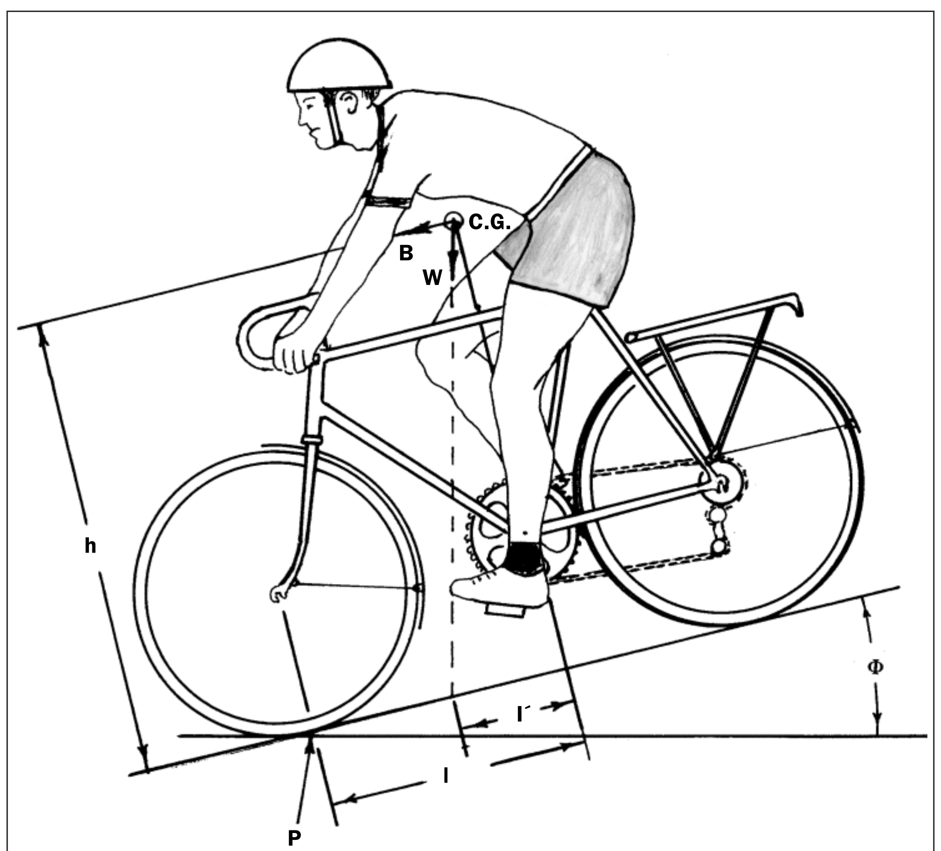


Figure 1. Sketch of forces and moment arms on bicycle and rider





**Analysis of pitchover**

Because the rider on a conventional bicycle sits high and the wheelbase is fairly short, braking tends to cause the bicycle and rider to pitch over. On level ground rather severe braking is required to result in a pitchover. However, as the bicycle inclines forward, such as in going down a hill, the weight vector inclines forward such that more weight is carried by the front wheel than on level ground. Application of brakes produces a pitching moment as on level ground. Much less braking than on level ground can result in a pitchover. Analysis can tell us how much. The subject of pitchover has been covered by DeLong (1978, 208–209) [and Sharp (1977 [1896], 216–220) *Ed.*]. Herein an equation will be derived showing the braking required as a fraction of the total weight of bicycle and rider for a range of slopes of the road. The sketch (figure 1) shows a bicycle and rider on a road with a slope of Φ. The height of the combined bicycle-and-rider center-of-gravity on level ground is h. The distance of the center-of-gravity from the front-wheel contact point, P, on level ground is l. The total weight of bicycle and rider is W. The braking force is B. The inclination of the bicycle causes the line of action of the weight vector to go forward an amount l′ on the ground. The pitchover will be initiated when the inertial moment from the braking about P is equal to the moment of the weight. Noting that,

$$l' = h \tan \Phi \qquad \textit{Eq. 1}$$

$$\Sigma M_P = 0 = Bh - W(l - l') \cos \Phi \qquad \textit{Eq. 2}$$

Substituting for l, and simplifying,

$$B/W = l/h \cos \Phi - \sin \Phi \qquad \textit{Eq. 3}$$

This expression yields the brake force as a fraction of the total weight where pitchover will take place for various ratios of l/h and angles, Φ. It may be seen that on level ground (Φ = 0, cos Φ = 1.0, and sin Φ = 0) that B/W = l/h. Equation 3 is plotted for a range of values of l/h versus the slope, Φ, on figure 2.

If one is to descend a hill at a constant speed and not accelerate, the resistance to motion must equal the accelerating component of gravity. Assuming that that resistance is the force supplied by the brakes, B,

$$B = W \tan \Phi \qquad \textit{Eq. 4}$$

The braking force for this steady state increases as the slope increases. On a steep hill this required braking force could exceed that which would result in a pitchover. Under these conditions the rider is in serious trouble. The boundary is shown on figure 2 as a dashed line. To show where actual bicycles and riders would lie on the figure, the author's two touring bicycles, with him on them, were chosen. One is a lightweight touring bicycle of conventional design and the other is a Bike Friday New World Tourist model. These two bicycles differ greatly in appearance, but fit and perform similarly. Each was fitted with fenders (mudguards), rear pannier rack, frame pump and empty water bottle. The values of l and h were determined. Because both bicycles are custom built, the values are not considered applicable to other people's bicycles. The curves of B/W versus Φ are shown. The differences between the two curves is surprisingly small. The center-of-gravity of the Bike Friday was slightly lower than the conventional bicycle, making it a bit more stable.

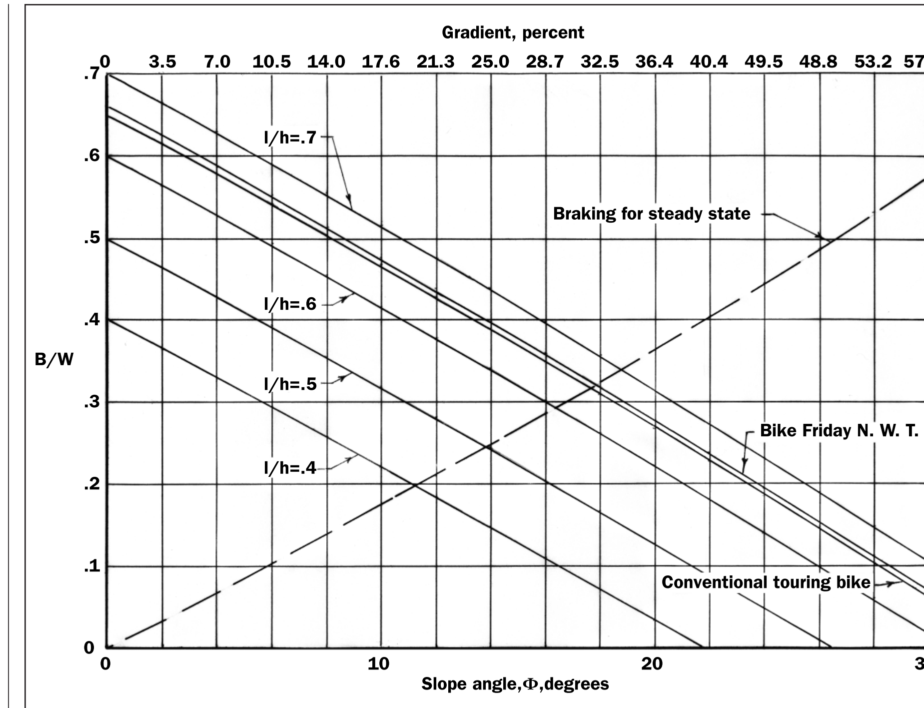


Figure 2. Pitchover points versus descent angle

**Definitions**

- **F slope**
- **h height of the combined bicycle-and-rider center-of-gravity on level ground**
- **P front-wheel contact point**
- **W total weight of bicycle and rider**
- **B braking force**
- **l horizontal distance from bicycle-and-rider center of gravity on level ground to contact point, P**
- **l′ = h tan Φ**

**Discussion**

Figure 2 shows, as an example, that if one is on a bicycle and the value of l/h is 0.5 (curves descending to the right) and the slope is ten degrees (vertical lines) that their intersection occurs at a value of B/W (horizontal lines) of approximately 0.32. That means that a braking force of about a third of the weight of the bicycle plus rider would be enough to cause a pitchover. This level of braking can be easily attained. But because the point lies well above the curve for braking for steady state, less braking applied than that for pitchover would allow deceleration. If the slope were to increase to about 14 degrees, the pitchover-braking and steady-state-braking curves coincide and there is no margin for slowing. If one follows the l/h curves down to the x-axis, those angles where tan $\Phi = h/l$, result in a pitchover with no braking. The importance of figure 2 to the cyclist is not in such calculations, but rather in permitting an understanding of the nature of the problem. In practice one probably does not know the steepness of the hill precisely. The values of l/h vary with the position of the rider; the rider can increase l/h by lowering his body or sliding back on the saddle. Further, the rider can only roughly judge the amount of braking force. Lastly, the likely situation for a pitchover will often not be in a steady descent, but in some sort of emergency condition.

It is obvious from the analysis that the slope of the roadway is a most critical variable. In the USA criteria and standards exist governing slopes of roads. The American Association of State Highway and Transportation Officials (Merritt, 1983, Table 16-6) has set forth limiting standards for slopes. On Interstate highways and primary roads the limits are five to seven percent (3–4 degrees). On secondary roads and in mountainous terrain limits of 10 percent are suggested (6 degrees). These standards are suggested standards only and each state, in fact, establishes its own standards. In reality roads may vary widely and often exceed the standards in the USA. In other countries the slopes may be greater.

In the vicinity of Hollister, California, the author measured slopes. On local roads and highways the slopes measured did not exceed eight degrees (14 percent). However in residential subdivisions, where children live, roads used by cyclists often approached or reached ten degrees (17.6 percent). The county [San Benito] allows up to 15-percent slopes for long stretches or 16 percent for up to 122 meters (400 feet).

In mountainous areas of the world the cyclist is likely to encounter sharp turns often referred to as hairpin turns or switchbacks. These turns pose special dangers. Good practice dictates that the turn not be too sharp, i.e., that an ample radius of the inner road edge should exist. In the case of a descending turn, to keep the cross section of the roadway level requires that the inner portion of the road be cut away along radial lines resulting in a helical shape. If the road extends to the center of the curve then that portion of the road at the center descends vertically. Such turns exist. These turns often restrict visibility severely. As one rounds a turn one may find oneself facing a bus or truck inching up the slope, blocking the road and requiring sudden braking and possibly making the cyclist hug the inner side of the road. An accident may not be avoidable. It is the local slope of the surface that is critical and the braking used, even momentarily, which will result in the pitchover. Speed is important too because, although it does not in itself determine pitchover, the moving bicycle pitching over can act as a catapult on the rider and make the accident more serious.

What can be done to avoid pitching over? If the rider has time to react he might lower himself or move aft on the bicycle. It would seem helpful to lower or move aft the baggage. Panniers can be carried low. They tend to get dirty from splashing and sometimes hit curbs, but otherwise seem to be satisfactory. Moving the load aft is difficult and may adversely affect handling. However, if, as often happens in touring, the baggage is removed, say, to climb a mountain for sightseeing, the load is not available for the dangerous descent.

Although they may have other shortcomings in hills, tandems do have superior longitudinal stability. Recumbent bicycles come in a large variety of forms, but the lower center of gravity should generally alleviate the pitchover problem. The long-wheelbase style should be very stable. Sometimes planning can be beneficial. If one is making a round trip on a road with a steep hill in one direction, the direction of travel should be safer climbing rather than descending that hill. It is one purpose of this article that understanding the danger and the mechanism of pitchover should make for a safer cyclist. Another hope is that road designers and builders will understand that some designs and practices may not be particularly hazardous to motor vehicles and yet be very hazardous to bicycles. In many cases hazards can be rectified, reduced or avoided.

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### The author

Frederick H. Matteson, a retired engineer, has been riding bicycles for 70 years and touring in the USA and abroad over a span of almost 60 years.


*—Frederick H. Matteson*
*1874 Cushman Street*
*Hollister, CA 95023-5525 USA*
*Phone: 831-637-6598*




# CdA and Crr measurement


***by John C. Snyder, Jr.***



### Abstract

This paper provides the conceptual basis, and examples, of ways to estimate an HPV's rolling and aerodynamic resistance by utilizing the slope-intercept form of a linear equation as applied to coastdowns on down-grades.


### Introduction

This testing protocol was developed to evaluate objectively two of the qualities affecting the performance of a bicycle. The procedures detailed do not seek to model fully all of the physical factors involved. Instead, they serve as a means to make quantitative observations during actual riding conditions. Of significance, the methods rely upon measurable steady states.

Requirements for constant-slope hills and for wind-free conditions impose a limit to when and where the protocol may be employed. The reader is encouraged to study works by Chester Kyle and others listed in the suggested reading list, to explore indoor and similar methods used for evaluating aerodynamic drag and rolling resistance.

### Premise

There are six losses that a cyclist must overcome while pedaling: gravity (if she or he is climbing), air drag (if adverse), rolling resistance, linear and rotational inertia changes (if accelerating), and drive-train inefficiencies [1]. At a constant velocity, inertia changes are null. While coasting, drive-train losses are non-existent [2]. Terminal velocity of a downhill-coasting bicycle refers to the phenomenon whereby the down-road component of the force of gravity achieves equilibrium with the two remaining primary loads: aerodynamic drag and rolling resistance.

$$\mathrm{Cd} = \frac{\text{mass} \times \text{gravity} \times [\text{sine(angle)} - \mathrm{Crr}]}{\text{air density} \times \text{area} \times \text{terminal velocity}^2 \times 0.5}$$

The above formula, describing the coefficient of drag (Cd) of a land vehicle when coasting at terminal speed, squares velocity and contains a sine function. Therefore it does not appear at first inspection to follow the pattern of a linear equation. The equation may be treated as though it were linear when only Cd and the coefficient of rolling resistance (Crr) serve as its variables and its other terms are regarded as constants.

Employing the following redistribution and substitutions facilitates setting the equation into linear slope-intercept form:

$$\mathrm{CdA} = \frac{\text{mass} \times \text{gravity}}{\text{air density} \times \text{terminal velocity}^2 \times 0.5} \times [\text{sine(angle)} - \mathrm{Crr}],$$

and putting

$Y \equiv \mathrm{CdA}$,

$X \equiv \mathrm{Crr}$,

$$m \equiv \frac{\text{mass} \times \text{gravity}}{\text{air density} \times \text{terminal velocity}^2 \times 0.5}, \text{ and}$$

$b \equiv m \times \text{sine(angle)}$, then the equation can be given as

$$Y = (-1)mX + b.$$

By establishing values for mass, gravity, air density, terminal velocity, and hill angle occurring in two dissimilar settings (prime and double prime), i.e., each producing unique values for linear slope (m) and the y-axis intercept (b), it becomes plausible to solve for Crr (X) using:

$$\mathrm{Crr} = \frac{b' - b''}{-m'' + m'}\ .$$

Once Crr is known, the corresponding CdA (Y) value may be determined with either:

$\mathrm{CdA} = (-1)m'X + b'$, or

$\mathrm{CdA} = (-1)m''X + b''$ .

The opportunity now exists to identify experimental situations whereby two separate coasting events might be conducted, measured, and compared.

### Two-hill comparison

Four constants appear within the formula (gravity, mass, air density, and angle) that, when modified, result in a corresponding change in coasting velocity. For the moment, the most consequential of those four to alter by a known amount is angle. It is hypothesized that a comparison of terminal velocities as achieved on two unlike hills, with similar road textures, permits quantifying a bicycle's unique CdA and Crr solution set pair.

In practice, comparing the terminal velocities achieved on two different hills, as well as the single-hill test that appears later, should be recognized as a method to approximate the values describing performance.

**Input values and dimensions** [3]

Reflect for a moment on the constants. Each needs to be known with some assurance as it applies to a specific coasting event. Knowing if calm wind conditions exist is also essential, as the formulation tacitly implies air and ground velocity equal one another. Fortunately, convivial means exist to quantify each.

***Mass***

The system's total mass can be found quickly by standing on a scale while holding the bicycle and all equipment that will be carried while riding. Ideally this measurement should be taken just prior to coasting.

***Gravity***

Acceleration due to gravity varies only slightly at different points on the earth's surface. It can generally be considered to be a value of 9.81 $m/s^2$.

***Wind***

A length of about 300 mm of single-ply tissue paper when held hanging straight down reveals whether or not calm wind conditions prevail. This deceptively uncomplicated tool demonstrates sensitivity to subtle movements of air. For the present need, a wind-speed-measuring device need only indicate zero wind velocity (see table 1).

***Air density***

The density of air, often referred to by the Greek letter $\rho$, varies considerably with altitude, temperature, and humidity.

| Force | Strength | km/h | mph | Observation |
|---|---|---|---|---|
| 0 | Calm | 0–1 | — | Smoke rises vertically; tissue hangs vertically. |
| 1 | Light air | 1–5 | 1–3 | Smoke drifts; tissue moves slighty. |
| 2 | Slight breeze | 6–11 | 4–7 | Leaves rustle; tissue becomes horizontal |

**Table 1. A portion of the Beaufort wind scale**

However, a reasonable value may be obtained by applying air temperature and pressure to the ideal-gas law:

$\rho = P/(RT)$

where:

$\rho \equiv$ air density (kg/m$^3$)
$P \equiv$ pressure (Pa)
$R \equiv$ constant (J/kg K)
$T \equiv$ temperature (K)

After accounting for modern weather services' custom of using hPa and Celsius, the equation becomes

$$\rho\ (\text{kg/m}^3) = \frac{\text{hPa}}{2.87 \times (\text{degrees C} + 273.15)}$$

***Hill angle***

Obtaining the angle of the roadway can be forthright as well. One credible method is to acquire a large protractor with a hole at its vertex through which to affix a length of thread or thin string. The other end of the thread will be tied to a small weight. A straight hollow cylinder, such as a drinking straw, taped or glued to the base of the protractor serves to form a sighting guide (fig. 1).

While looking through the sight one moves the device until the inverted protractor's base becomes parallel with the slope of the road. Then by pinching the thread tight to the protractor, the hill's angle in degrees may be read directly.

Alternatively, one may use a commercially available inclinometer such as found on some magnetic pocket compasses, or a tripod-mounted transit designed specifically for the purpose.

Occasionally, reliable survey data will be available. If a hill's slope is expressed as a grade percent, convert to degrees of angle with the following:

degrees = arc tangent(grade/100)

Some computer spreadsheet programs perform trigonometric functions exclusively with radians, which can be converted with the following relationships:

1 degree = $\pi$/180 radians,
1 radian = 180/$\pi$ degrees.

***Terminal velocity***

Transient conditions influencing coasting hold the potential of exerting profound, even if subtle, accumulating effects. Any coasting event should be recognized as the sum result of many unidentifiable and a few identifiable controlling factors. As examples: edging past a tiny unseen pebble, traveling through a small dip in the road bed, or even making slight unavoidable movements in steering, will induce a momentary velocity change, thus modifying the total time and distance traveled.

When observing an equilibrium condition, such as represented by the essentially unchanging value of terminal velocity, minor variations in a road bed and other transient phenomena exhibit little or no influence.

The rate of velocity increase becomes minuscule as a downhill-coasting bicycle approaches terminal speed. The digital output of a cycle computer typically rounds units of kilometers or miles to one or two decimal places. For those reasons, a practical reading of the steady state will appear conveniently prior to the actual occurrence of terminal velocity. When coasting down a constant slope, one needs only to monitor a cycle computer to determine when velocity no longer increases or decreases.

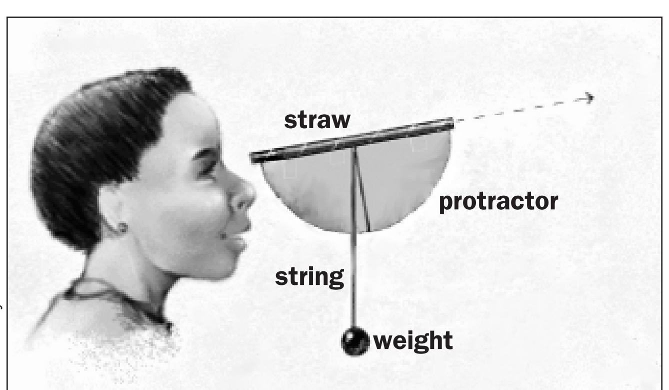


**Figure 1. Inclinometer**

The maximum-velocity function available on some digital cycle computers will display the terminal velocity value, but only if terminal velocity has not been exceeded due either to pedaling or variation of the roadbed. As with any data-collection procedure, obtaining as many samples as practical is desirable.

These unit conversions may prove helpful:

mph × 0.4469 = m/s
km/h × 0.2778 = m/s

**Example A**

The following depicts a bicycle that has coasted to its equilibrium condition on two separate hills [4].

***Conditions, hill #1 (prime)***

combined rider & vehicle mass: 100kg
grade: 5%
temperature: 26C
air pressure: 1014 hPa
terminal velocity: 12.5 m/s
surface: smooth asphalt
wind: still air

angle = arc tangent(grade/100)
= arc tangent(0.05)
= 2.86 degrees

$$\text{air density} = P/RT = \frac{1014\ \text{hPa}}{2.87 \times (26\text{C} + 273.15)} = 1.18\ \text{kg/m}^3$$

$$m' = \frac{\text{mass} \times \text{gravity}}{\text{air density} \times \text{terminal velocity}^2 \times 0.5} = \frac{100\text{kg} \times 9.81\ \text{m/s}^2}{1.18\ \text{kg/m}^3 \times 12.5^2\ \text{m/s} \times 0.5} = 10.64\ \text{m}^2$$

$$b' = m \times \text{sine(angle)} = 10.64\ \text{m}^2 \times \text{sine } 2.86 \text{ degrees} = 10.64\ \text{m}^2 \times 0.05 = 0.53\ \text{m}^2$$





***Conditions, hill #2 (double prime)***

combined rider & vehicle mass: 100kg
grade: 8%
temperature: 23C
air pressure: 1016 hPa
terminal velocity: 16 m/s
surface: smooth asphalt
wind: still air

angle = arc tangent(grade/100)
= arc tangent(0.08)
= 4.57 degrees

$$\text{air density} = P/RT = \frac{1016\ \text{hPa}}{2.87 \times (23\text{C} + 273.15)} = 1.20\ \text{kg/m}^3$$

$$m'' = \frac{\text{mass} \times \text{gravity}}{\text{air density} \times \text{terminal velocity}^2 \times 0.5} = \frac{100\text{kg} \times 9.81\ \text{m/s}^2}{1.20\ \text{kg/m}^3 \times 16^2\ \text{m/s} \times 0.5} = 6.41\ \text{m}^2$$

$$b'' = m \times \text{sine(angle)} = 6.41\ \text{m}^2 \times \text{sine } 4.57 \text{ degrees} = 10.64\ \text{m}^2 \times 0.08 = 0.51\ \text{m}^2$$

***Determination of CdA and Crr***

Determination of the coefficients of rolling resistance and aerodynamic drag times area now occurs in the following manner:

$$\text{Crr} = \frac{b' - b''}{-m'' + m'} = \frac{0.53\text{m}^2 - 0.51\text{m}^2}{-6.41\text{m}^2 + 10.64\text{m}^2} = 0.0047, \text{ and}$$

$$\text{CdA} = -m'(\text{Crr}) + b' = -10.63\text{m}^2 \times 0.0047 + 0.53\text{m}^2 = 0.48\text{m}^2$$

**Single-hill comparisons**

The previous treatment creates a foundation for additional testing methods. Physically modifying a bicycle's CdA by a known amount during one of two coastdowns will cause the terminal velocity to change in a predictable fashion (Kyle, 1984, 22–40 [5]). This observation suggests comparative coastings to determine CdA and Crr might be conducted on a single hill in conjunction with a drogue device (see page 13).

**Auxiliary drag**

If
Y vehicle ≡ CdA,
Y drogue ≡ a known modifying value of CdA, and if
(Y vehicle + Y drogue) = CdA total, then
(Y vehicle + Y drogue) = (–1)mX + b, and
Y vehicle = (–1)mX + b – Y drogue.

These relationships permit the comparison of a coasting vehicle's unmodified (prime) and modified (double prime) configurations to give Crr:

$$(-1)m'X + b' = (-1)m''X + b'' - \text{Y drogue}$$

$$X = \frac{b'' - b' - \text{Y drogue}}{-m' + m''} = \text{Crr}.$$

Though there exist various ways to alter a bicycle's aerodynamic properties, the deployment of a small parachute [6], or the attachment of a rigid plate off to the side of the vehicle [7], represent ideas that have been successfully adopted in the past for increasing total aerodynamic drag in a controlled manner. It is assumed by this mathematical model that any auxiliary source of drag will be configured such that it will not significantly interact with the normal performance characteristic of the tested vehicle. A drogue device also implies a more elegant handling.

When
M ≡ mass
g ≡ gravitational constant
G ≡ grade [8]
ρ ≡ air density
v ≡ velocity,

the following equation depicts the equilibrium between aerodynamic drag and rolling resistance, and grade's effect when at the steady state,

$(0.5)(\rho)(v^2)(\text{CdA}) + \text{Mg}(\text{Crr}) = (-1)\text{Mg}(\text{G})$

This equation may be rearranged into:

$(0.5)(\rho)(v^2)(\text{CdA}) = (-1)\text{Mg}(\text{G} + \text{Crr})$.

If a vehicle's CdA′ changes by a known amount (CdA″) then the following also applies:

$(0.5)(\rho'')(v''^2)(\text{CdA}' + \text{CdA}'') = (-1)\text{Mg}(\text{G} + \text{Crr})$

If two coastdowns to terminal velocity are conducted, only one of which has been modified, and both occur during identical weather conditions on the same hill, permitting cancellation of values that have not changed, then:

$(v'^2)(\text{CdA}') = (v''^2)(\text{CdA}' + \text{CdA}'')$,

which, after having been solved for CdA′, reveals the following description:

$$\text{CdA vehicle} = \frac{\text{CdA drogue}}{(v'/v'')^2 - 1}$$

By solving this equation, a bicycle's CdA may be assessed even if mass, grade, and air density are unknown values, but are consistent from one coasting event to the next.

**Example B**

The following depicts measuring the CdA of a bicycle coasting to terminal velocity twice on a single hill. The first coasting occurs without changes to the system. The second coasting is performed while a drogue device is deployed.

***Conditions***

CdA drogue = 0.44 m$^2$
terminal velocity, unmodified (v′) = 15 m/s
terminal velocity, modified (v″) = 9 m/s

***Determination of CdA***

$$\text{CdA} = \frac{\text{CdA drogue}}{(v'/v'')^2 - 1} = \frac{0.44\text{m}^2}{(15\text{ m/s} / 9\text{ m/s})^2 - 1} = 0.25\ \text{m}^2$$

**Limitations and strengths**

The accuracy of determining the hills' angles, air density, mass, and velocity values, ultimately control the quality of the solution. Though simple in design the suggested instruments are pragmatic and easily obtained. It is conceivable that by exercising due care any error brought about by input data could be made negligible.

There are several conceptual concerns. First is the model's presumption that a precise single CdA and Crr solution set pair exists at all. It is unlikely that Crr can be identical on distinctly different roadways. Cd is not a constant throughout a range of velocities [9,10]. The assumption that a drogue could be deployed such that it does not interact in any manner with the vehicle's normal performance characteristic is indefensible. However, by utilizing comparative velocities, and other conditions which are close in value to one another the significance of these inherent errors will be lessened.

There exist logistical concerns. Any

testing done while other traffic is present will be affected. The air must be still, a rare condition most frequently occurring shortly after sunrise. Constant slopes of sufficient length, with reasonable constancy and accessibility are essential. And, above all safety must be in the forefront of an investigator's thoughts at all times while operating a bicycle. These hurdles are often fully surmountable. With recognized limitations, testing as presented can provide an accessible way to estimate meaningful frontal CdA and composite Crr values for an individual vehicle. Most important, these estimates may be based on observation of steady-state conditions as occurring during actual road conditions.

### Notes and references


1. Witt, F. R. and Wilson, D. G. 1982. *Bicycling science*, 2nd ed., Cambridge, MA: MIT Press.
2. Crr has been modeled here as including all conditions which impede the rolling motion of a bicycle at a constant velocity. Thus, hub bearing losses are accounted for in the resulting Crr values as well as road texture, effects of tire design, tire air pressure, operating temperature, axle nut torque, et.al. It is recognized that bearing losses might be accounted for separately with additional handling and measurements.
3. Personal correspondence with Israel Urieli , Associate Professor, Dept. of Mechanical Engineering, Ohio University, Athens, Ohio, Fall 1999. (Reference applies to text section devoted to input values.)
4. Although m and b are shown as area, their unit serves only to facilitate dimensional analysis.
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7. Compton, Tom. "Coast-down tests", Hard-core-bicycle-science discussion thread (moderated e-mail list). 05 Aug 1997. http://www.cyclery.com/lists/hardcore-bicycle-science
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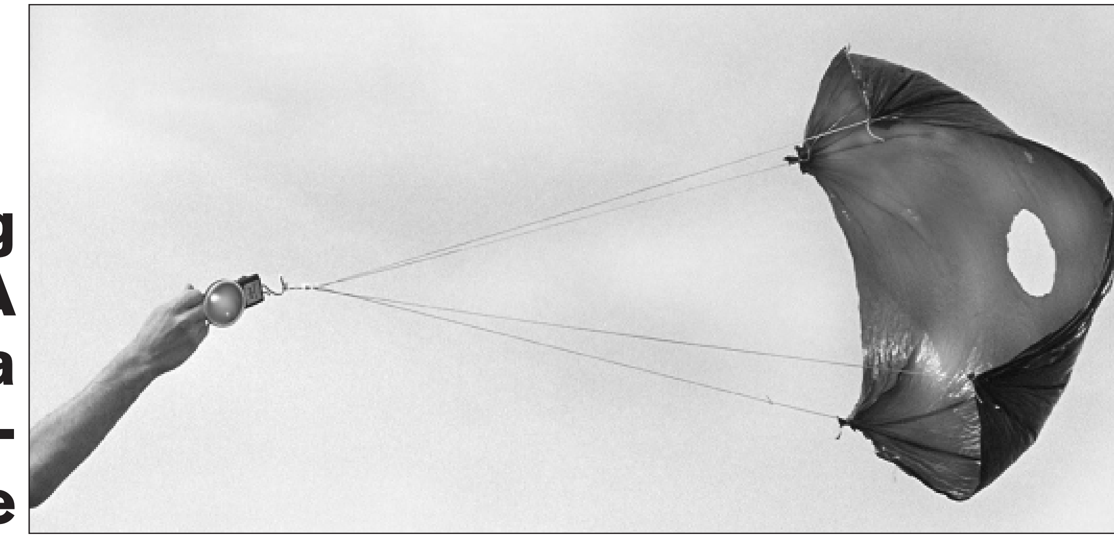
### The author

John Snyder is a cyclist from Great Falls, Montana.


*—John Clark Snyder, Jr.*
*12 Golden Valley Loop*
*Great Falls, MT 59404-6114 USA*
*Phone: 406-727-4132;*
*<JCSnyder.studio@worldnet.att.net>*




# Finding the CdA of a parachute



John C. Snyder

**Drogue parachute**

Construction of a drogue parachute suitable for HPV experimentation is an easy project. Using scissors, cut the two side creases and the bottom fold of a large, plastic lawn-and-leaf bag (2.4 micrometer thickness) to obtain two equally-sized panels of material. Set one aside; the other will serve as a rectangular parachute canopy. To each of the four corners, which have been knotted, tie lengths of thin nylon cord or twine. The other ends of the cords then can be attached, via a fishing-line swivel, to a handle. A circular hole cut in the center of the rectangle helps stabilizes the parachute when it's open and filled with moving air. The coefficient of aerodynamic drag of any drogue device is defined by:

$$\text{CdA drogue} \equiv \frac{\text{drag}}{\text{air density} \times \text{air velocity}^2 \times 0.5}$$

In order to determine the product of coefficient of drag (Cd) and area (A), it is not necessary to know the individual values of either Cd or A as they exist separately.
A dimensional analysis reveals

$$\text{CdA drogue} = \frac{\text{N}}{(\text{kg/m}^3) \times (\text{m/s})^2 \times (\text{no unit})}$$

$$= \frac{\text{kg m/s}^2}{\text{kg/m}^3 \times (\text{m/s})^2}$$

$$= (\text{m}^2).$$

Thus, a drogue device's CdA may be learned by finding representative values for: air velocity, air density and the force due to air resistance. If connected to a land vehicle traveling through still air, a drogue device's air velocity equals the vehicle's ground speed. Estimating air density occurs by applying the prevailing air temperature and air pressure to the ideal-gas law.

The force is found with a little more effort. A spring-scale may be adopted as a hand hold when attached to a small parachute's shroud lines. While riding aboard a tandem bicycle or an automobile, a passenger holds the open parachute by the scale's handle into a non-turbulent region of air flowing past the vehicle. The resulting drag causes an SI-unit instrument to display a reading in newtons (N). If the type of spring-scale available registers only kilograms, multiplying that unit number by 9.81 m/s$^2$(g) will yield the drag force (N).

**Example**

| | |
|---|---|
| wind conditions: | calm |
| relative air velocity: | 8.5 m/s |
| air drag: | 20.5 N |
| air temperature: | 26 C |
| air pressure: | 1014 hPa |

air density = P/RT

$$= \frac{1014 \text{ hPa}}{2.87 \times (26\text{C} + 273.15)}$$

$$= 1.18(\text{kg/m}^3)$$

$$\text{CdA drogue} = \frac{\text{drag}}{\text{air density} \times \text{air velocity}^2 \times 0.5}$$

$$= \frac{20.5 \text{ N}}{1.18 \text{ kg/m}^3 \times (8.50 \text{ m/s})^2 \times 0.5}$$

$$= 0.48(\text{m}^2)$$

When deploying a drogue parachute to estimate either the parachute's or a bicycle's CdA it is strongly advised first to deploy the canopy at low initial air/ground speeds, then slowly and cautiously increase the towing vehicle's velocity to a higher, though still modest, constant rate. The parachute described herein must not be used to slow an over-speeding vehicle.

**TECHNICAL NOTES**

# Efficiency of bicycle chain drives: results at constant velocity and supplied power

*by Claire L. Walton and John C. Walton*

The paper by James B. Spicer *et al.*, (2000) presents very useful and relevant information for the further understanding of HPV transmissions. Its conclusions concerning the effects of lubrication, rotation rate, and tension on efficiency are highly valuable. We believe this contribution will be viewed as even more significant when the data are presented from a slightly different perspective.

The testing apparatus was set up to maintain the rpm of the front chainring and the power applied to it at constant levels. A single 52-tooth chainring was tested. This leads to the observation that the largest rear wheel sprocket is most efficient. Though correct, this result is not necessarily widely applicable. When applied power, crank rpm, and chainring size are held constant, the velocity of the vehicle and the force applied to the rear wheel must vary. Since the same work and chainring rpm are producing different velocities, a different force must be reacting against the wheel. A physical analogy for the columns in table 1 and the results in figure 2 of the paper would be a situation where the 52:11 gearing represents downhill, 52:15 is level ground, and 52:21 is uphill. In this situation the 52:21 going uphill has greatest chain efficiency. This is a valid conclusion, but not the primary question in HPV design and operation. Similarly, in figure 3 the chain tension is kept constant. At constant tension larger sprockets are more efficient, but they would also be delivering more torque.

We are more interested in the case of constant power and constant rpm of the rear sprocket (i.e., constant velocity of a vehicle on the road at constant power supply to the wheel). What sprocket will be most efficient for a vehicle at constant velocity? The experimental results suggest a trade-off. At constant vehicle velocity (and other conditions) a smaller sprocket will have a greater chain tension than a larger sprocket. The higher tension in the smaller sprocket will tend to counteract the inherent lower efficiency of the smaller sprocket (when sprockets are compared at constant tension). Which effect is more important for the situation of a vehicle traveling at constant velocity?

In order to address this question we take the measured data (specifically the linear fits from figure 3 in Spicer *et al.*, (2000)) and present the results in a revised format. The experimental results are not changed: they are merely presented differently using simple algebraic manipulations.

The power supplied to the wheel, Pw and by the chain, Pc are given by:

$$P_w = 2\pi r_w \omega F_w \qquad \text{Eq. 1}$$

$$P_c = 2\pi r_s \omega T_c \qquad \text{Eq. 2}$$

The difference in power is from the loss of efficiency in the chain.

$$P_w = \zeta_s P_c \qquad \text{Eq. 3}$$

$$\zeta_s = \frac{P_w}{P_c} \qquad \text{Eq. 4}$$

Substituting the equations for power and simplifying.

$$2\pi r_w \omega F_w = 2\pi r_s \omega T_c \zeta_s \qquad \text{Eq. 5}$$

$$T_c = \frac{F_w r_w}{\zeta_c r_s} \qquad \text{Eq. 6}$$

In order to eliminate variables we take a ratio of the chain tension from the use of two different sprockets (11 and 21-tooth) while keeping power supplied to the rear wheel constant:

$$\frac{T_{21}}{T_{11}} = \frac{\zeta_{11} r_{11}}{\zeta_{21} r_{21}} = \frac{11\zeta_{11}}{21\zeta_{21}} \qquad \text{Eq. 7}$$

A similar equation is used for the 15-tooth sprocket. The linear fits of chain tension versus efficiency for the different sprockets in figure 3 of Spicer *et al.*, (2000) provide relationships between efficiency and tension for each sprocket. We assume that efficiency is independent of chain speed (as reported by Spicer *et al.*, 2000) and that most of the chain loss is associated with the rear sprocket. Figure 3 of the paper indicates that at equal tension, larger sprockets are more efficient. It follows logically that most of the loss in chain efficiency occurs in association with the rear sprockets which are all much smaller than the 52-tooth chainring.

Solving the system of equations provides an estimate of the chain efficiency for the case of constant vehicle velocity and power applied to the rear wheel. The alternative data presentation is shown here in figure 1 with efficiency for each sprocket size given as a function of the chain tension in the 11 tooth sprocket. For example, when the tension in the 11-tooth sprocket is 1/0.006 = 167 Newtons, the efficiencies are 92% for the 11-tooth sprocket, 90.5% for the 15-tooth sprocket, and 88.5% for the 21-tooth sprocket—assuming the same vehicle velocity and power to the rear wheel for all three sprockets. The corresponding tensions and efficiencies for the 15 and 21 tooth sprockets are calculated using equation 7. The lines are truncated approximately at the limits of the experimental data. Over most of the experimental range the smaller sprockets give greater chain efficiency. The lines appear to converge at high tensions, with all three sprockets giving high efficiency.

The surprising and counterintuitive result is that the smaller sprockets have greater estimated chain efficiency at constant vehicle velocity and applied power than the larger sprockets. Therefore, the increased efficiency from the higher chain tension is more important than the loss of efficiency from having the smaller sprocket. Clearly more experimental data using different chainring and sprocket combinations will be required to answer questions on chain efficiency definitively.

**NOMENCLATURE**

$r_w$ = wheel radius, m
$\omega$ = rotation rate, rev/s
$F_w$ = propulsive force, N
$r_s$ = effective sprocket radius, m
$T_c$ = chain tension, N
$\zeta_s$ = chain power loss factor or efficiency

**REFERENCE**

Spicer, J.B., J.K. Christopher, M.J. Richardson, J. Ehrlich, and J.R. Bernstein. 2000. On the efficiency of bicycle chain drives. *Human Power* 50:3–9.

**AUTHORS**

Claire Walton is a senior at Franklin High School in El Paso, Texas. She is interested in pursuing a career in math, physics, and music.

John Walton is an engineering professor at the University of Texas at El Paso who rides recumbent cycles.

*—John Walton <walton@utep.edu>*
*Associate Professor*
*Department of Civil Engineering*
*University of Texas at El Paso*
*El Paso, TX 79968 USA*

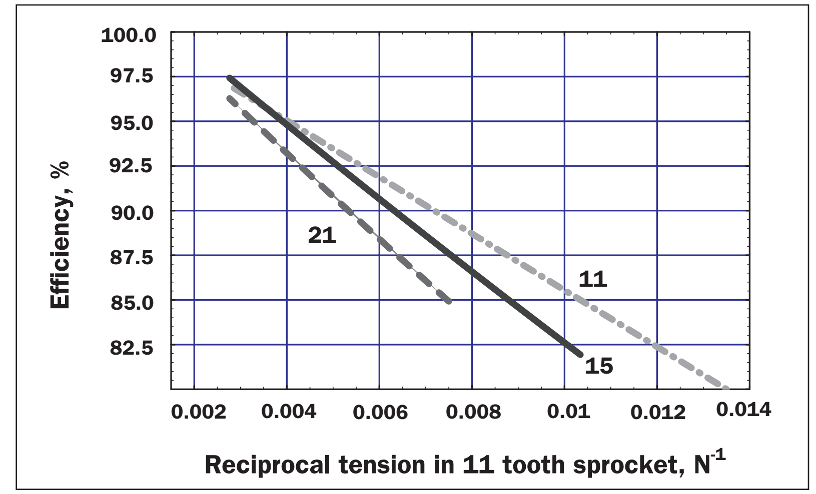



**Figure 1. Efficiency of the 11, 15, and 21-tooth sprockets at constant vehicle velocity and power to the rear wheel.**

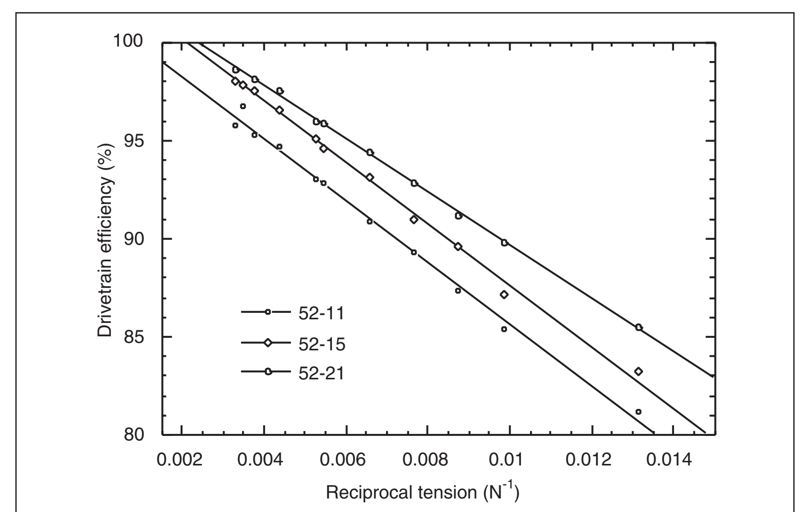



**Table 1. Drive efficiencies for different chain configurations**

| | 50 RPM 100 W | 60 RPM 100 W | 70 RPM 100W | 60 RPM 150 W | 60 RPM 175W |
|---|---|---|---|---|---|
| 52–11 | 92.5 | 91.1 | 88.7 | 94.6 | 95.5 |
| 52–15 | 94.7 | 92.3 | 90.4 | 96.2 | 97.5 |
| 52–21 | 95.2 | 93.8 | 92.0 | 97.4 | 98.2 |

**Figure 3 (left) and Table 1 (above) provided by James B. Spicer for his article in Human Power 50, "On the efficiency of bicycle chain drives" are reproduced here for the convenience of readers of the technical notes submitted by Claire C. Walton, John Walton, and John Allen.**

## FURTHER COMMENTS ON THE SPICER ARTICLE ON DRIVE-TRAIN EFFICIENCY

*John S. Allen*

The Spicer article on bicycle-drive-train efficiency is interesting, and the research appears to me to have been well-conducted. The conclusion that much of the power loss was not converted to heat (that is, went into vibration instead) is interesting, as are the conclusions that loss is not much greater with an unlubricated chain or with chainline offset (though I think the losses with offset would be greater with older types of chain with flat side plates which do not engage and disengage smoothly). The conclusion that larger sprockets increase efficiency is expected from theory.

Spicer used two torque-measuring devices, one attached to the crank and the other to the rear wheel. At the high efficiencies typical of chain drives, this approach to measurement is somewhat prone to error, because the measurement of interest—the difference between the actual efficiency and 100%—is a small difference between two large quantities.

One way around this problem is to use a single measuring device to measure a torque difference. Implementation of this approach is simple with a unity drive ratio: the torque from the motor at the input of the drive-train may be applied directly to the brake at the output, where it cancels except for the difference due to power loss. At 100% efficiency, the torque of the motor and brake cancel, and so do the measurement errors. Assuming that a reasonably accurate measurement of the input torque can be taken, this approach promises a high degree of accuracy for a high-efficiency system.

Implementing this approach is somewhat more difficult in the case of a bicycle chain drive, with its step-up ratio. The torque-combining system must have the same ratio. Suppose, for example, that the bicycle's chain drive has a 52/15 drive ratio. Then we could, for example, use another chain drive with the same 52/15 ratio to combine the torques at the motor and brake.

What objections might be made to this approach? A first objection might be that inefficiency of the torque-combining drive system would corrupt the measurement. But on second thought, it need not. The torque-combining drive system is not in motion, and so it absorbs only power which has already been lost through vibration of the primary drive system. And that vibration converts the sliding friction of the stationary torque-combining chain drive into viscous friction (as also happens, for example, with the pivots of phonograph tone-arms when subjected to the vibrations transmitted from the stylus in the record groove).

There is another, real and serious problem, however, and it also occurs because the torque-combining chain is not in motion. Chord factors average out in a chain drive that is in motion, but not in one which is stationary. The chord factor of the 15-tooth sprocket of our example is 1/cos 12°, or 1.022, and the chord factor of the 52-tooth chain-wheel is 1/cos 3.46° or 1.0018. The resulting range over which measurement may vary is the product of the two chord factors. An error range of over 2% is hardly desirable, given that the goal of the suggested measurement technique is inherent, high accuracy.

The chord-factor problem may be greatly reduced by doubling or tripling the number of teeth on the torque-combining chain-wheel and sprocket, or may be eliminated (at some cost in complication) by replacing the chain drive with a gear drive or with knife edges positioned by a jackscrew to achieve the desired torque ratios. I would be most interested to hear from someone who attempts any of these approaches.

Spicer's conclusions suggest some additional tests which he did not conduct. His infrared photographs show that much heat was generated in the derailleur pulleys. Most derailleur pulleys have primitive plastic sleeve bearings, yet no test was done of bearing lubrication, or of ball-bearing pulleys. Spicer never tested for how much friction is reduced by eliminating one or both derailleur pulleys. And would larger pulleys, with their smaller chord factor, increase efficiency by reducing vibration as well as bearing friction?

*—John S. Allen*
*<jsallen@bikexprt.com>*
*http://www.bikexprt.com/*

## Bicycle stability after front-tire deflation

***Dave Wilson (reporting partly for Andy Oury)***

*Drawings by author, 2000.11.13*

### The problem

On three occasions I have had front-tire blowouts, or at least rapid loss of pressure, that have resulted in my having been thrown off my bicycle with some violence. One was when riding a Moulton road bike as a bus was about to pass; one was on an Avatar LWB recumbent; and one on a CLWB recumbent, when I narrowly avoided being hit by a large truck. A friend told me about someone who was, in fact, killed after his front tire burst, causing him to be propelled into the path of a car. A photo intended to be humorous in *Bicycling* [Magazine] showed two men on a tandem competing in the Davis Double Century just after their front tire deflated (almost certainly after braking sharply at a corner during a mountain descent, thus overheating the rim). The caption stated that as they hit the ground their bones breaking sounded like a famous breakfast cereal ("snap! crackle! pop!").

The reporting from dead bicyclists is zero, and the reporting of and examination of bicycle accidents is so perfunctory that it is highly probable that a considerable number of deaths and serious injuries are the result of instability following front-tire deflation. Therefore this has to be regarded as a serious problem.

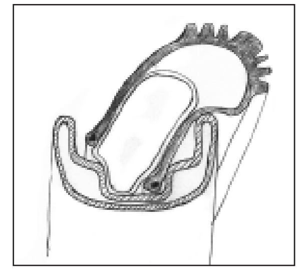


**Figure 1. Tire bead off seats: tread flops to side**

### Our study of the problem

In the summer of 1998 I wrote about flat-tire instability to a list [Internet discussion list] then called HBS, for "Hardcore bicycle science", moderated by Jim Papadopoulos (the name is now shortened to "Bicycle science" and it is moderated by Sheldon Brown). No one reported previous studies of this problem apart from one described by Doug Milliken, who wrote a letter "Flat-tire directional performance" to *Human Power* in spring 1991 (9:1, 17). He tested a motor-cycle fitted with proprietary run-flat tires on the rear wheel. The tires had a flap of rubber on the outside of the tire that fitted tightly over the rim and acted as a bead-retention system. One with a small flap did not in fact hold the bead when the tire was flat, and the bead fell into the "well" in the rim. The tire flopped around, causing the motor-bike to go unstable, even though the tire was on the rear wheel. The second tire with a wider flap held the beads in place.

With this tire, Milliken found that he could run the bike at high speed (80 km/h) and could perform various maneuvers without problem. He thought that good run-flat bicycle tires would probably be tubeless.

I wrote also to the HPV list, and several writers on this and on the HBS list contributed valuable experiences and suggestions. Some reported similar occurrences to mine, including Dave Larrington of the British Human Power Club, who had had "***instant*** crashes" from front-tire flats on regular bikes ("upwrong", in his words) and on recumbents, and Joshua Putnam, who considered the problem serious enough to institute the practice of letting the air completely out of the front tire when trying out a new bike. Bill Volk wrote, "I too find the situation to be unacceptable. I run heavy, inefficient thorn tubes because of my fear that a blowout at high speed would be a disaster. Why can't we have rims that retain the tires even at no inflation? And perhaps a rubber strip that is placed around the rim, under the tube, that supports the bike on loss of air pressure…. I had Performance semi-slick 26" tires that fit so snugly that you could safely ride no-inflation. That should be the standard."

Presumably because of a tight-fitting tire, Ed Deaton of Fools Crow Cycles, faced with difficult choices, rode five miles (8 km) on a flat front tire: he had IRC "Roadlites" with Sun M14 rims. Similarly Andy Milstein of Princeton had no trouble riding with a flat front tire. It was a Tioga Comp Pool, measured by Mark B. of Wheel Life Cycles to be 46-mm wide, on a Sun CR-18 20 x 1.75" rim of about 27-mm width. (That was significant because one of my early suspicions, and a concern of Larry Black, was that a wide tire on a narrow rim might have a greater tendency to "flop" alternately left and right.)

Bill Volk mentioned that Sutherland's Handbook for Bicycle Mechanics had a good section on fits between different brands of rims and tires, but my edition did not have this section, and I could not get an answer to my letter to Sutherland asking about standards of fit. John Allen, prominent bicycle expert and author, sent me a copy of his Japanese Industrial Standards D 9421, "Rims for Bicycles", giving a tolerance of +/- 3 mm for rim circumference, and of standard K 6302 "Rubber pneumatic tires for bicycles", which, he pointed out, gave neither tolerances nor dimensions of tire beads. (Later, Andy Oury, see below, found that the International Standards Organization ISO 5775/1 "Standards for bicycle tires and rims" also had tolerances for rim diameters but not, as far as he could determine, for tire beads. This was confirmed by Chris Juden, below.)

My instinct tells me that the old inch sizes had some specified or customary standards because my old 27x1-1/4" and other "inch" tires were all at-least "good" fits on the rims. Now, it seems from our experience and that of many people who wrote to me, it is entirely by chance that one gets a tire that is a tight fit on a rim and that will therefore provide a substantial degree of safety in the event of a front-tire blowout. However, Doug Milliken, a long-time consultant to Alex Moulton, wrote that Moulton controls both the rim diameter and the bead size of his 17" tires.

I wrote to Andrew M. Fischer, a Boston-area attorney who specializes in helping bicyclists with liability claims, but he had had no experience of this problem.

Chris Juden, technical officer of the Cyclists' Touring Club (UK), and a resource on every aspect of bicycle performance, wrote: "There are ISO standards for tyres and rims: ISO 5775 parts 1 and 2. The only trouble is: they were written by tyre manufacturers for their own convenience so -2 places rather tight tolerances upon rim-bead-seat diameter (plus or minus 0.48 mm) whilst -1 says nothing at all about the corresponding tyre-bead dimension!

"Having once been involved in rim manufacture, I can tell you we used to have some interesting arguments with Raleigh around the fact that a lightweight alloy rim inevitably shrinks some 0.46 mm in diameter when you put properly tensioned spokes in it! Since this standard doesn't say if it's talking about pre- or post-build dimensions, we had to restrict ourselves to only the top half of that measly tolerance or else Raleigh quality control would measure our bare rims or built wheels respectively, depending upon whether their latest shipment from Michelin were a tight or loose fit!

"On the BSI committee we found the tyre people always played their cards very close to the chest and would never be drawn when invited to submit appropriate limits and fits for their products or even the criteria for a simple blow off test. With many an 'Ah yes, but' and 'it's not so simple as that', the cycle-tyre industry has thus been remarkably successful in keeping the matter of product safety and testing entirely to themselves!"

Chris Juden is now on a committee of the European standards organization CEN, which enforces its standards on member countries. ISO standards are only, it seems, recommended.

John "LRaY" Stephens wrote "You should get some tire-and-rim-industry experts involved with this [question of standards for tire fits]. Unfortunately, I have never heard of any such persons. Tires just seem to float down out of heaven (or rise up from hell?)"…. After considerable efforts to reach tire manufacturers I was told that Vredestein, the Netherlands manufacturer of the tire on the German recumbent on which I had my most-recent episode of instability, was conducting a study on run-flat behavior. However, when eventually I received a courteous response from Mr. U. K. Banerjee, it turned out that he was investigating puncture prevention.

I bought a product called "Snake-charmer" from Bikewise International, Mammoth Lakes, CA. It was a length of dense solid trapezoidal-shaped rubber, intended to be fed into the rim—well under the tube to prevent "pinch" flats and presumably to give some run-flat capability. It was produced only for large ATB tires, at least at the time of my purchase, and I could not test it. It would add a considerable amount of mass to a wheel and, I would think, stress the tube, which would have to wrap around it.

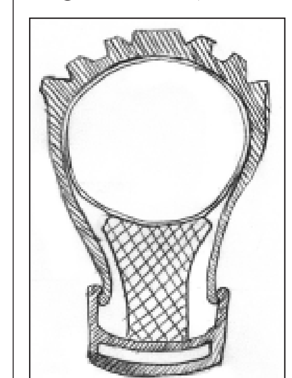


**Figure 2. "Snake-charmer" run-flat insert (note rim without bead seats)**

In September 1998 I added the problem statement on flat-tire stability to my list of undergraduate-thesis topics at MIT. Andy Oury, then a senior, responded enthusiastically, carried out several valuable experiments, and has allowed me to report some of his results here. We drew up a too-ambitious program in which we wanted to look not only at bead retention but also at the effect of the ratio of tire width to rim width (ATB tires in particular are usually bulbous, having a pear-shaped cross-section on what seems like a small rim) and of tire-sidewall stiffness. Andy Oury worked on what the correspondents just quoted thought was the most important factor, bead retention.

**The experiments**

We first thought that we could do a highly controlled experiment by having my troublesome bicycle wheel and tire, held in a frame, running on the surface of an inverted portable belt sander. However, the tire did not display the extraordinary alternating flops, left and right, that had thrown me off my bike, and that had prevented me even from pushing the bike subsequently. Oury found that, for the flopping behavior to occur, he had to rig up a bike to run along a simulated roadway with a similar number of degrees of freedom as has a bicycle when it is being ridden (or pushed).

The simplest way of producing bead retention on the shoulders of the wheel rim after deflation seemed indeed for them to be a tight fit. I have had tires that could be stretched over the wheel rims only with great difficulty. When these were inflated, the tire beads remained in the rim “well” until the tube inflation pressure reached around 80% of normal full pressure. They then “snapped” over the rim shoulders with a satisfying crack. My experience follows that of Doug Milliken and Bill Volk: I have never found tire instability with tires that were a tight fit on the rims, and which, therefore, did not flop loosely around in the rim when they became deflated. I confess that I cannot remember if I have had a front-tire blowout with a good-fitting tire. I would certainly remember something like the instability that made staying on the three bikes mentioned above absolutely impossible.

The tires that caused me the problems were exceedingly loose. This characteristic made puncture repair almost a pleasure, because one could get the tires on and off without tire levers. They were so loose, in fact, that centering them during subsequent inflation became difficult: it was easy to produce an eccentric rolling surface, even to the extent of having the tube pop out between tire and rim where the tire was high. Oury built up the rim shoulders using standard “masking” tape, and he put on fifteen layers before the tires were retained and the flat-tire flopping was inhibited. His experiments therefore did a great deal to confirm the premise: that a slack fit between tire bead and wheel rim is the prime cause of flat-tire instability and that a tight fit will therefore provide a substantial degree of safety in the event of a front-tire blowout.

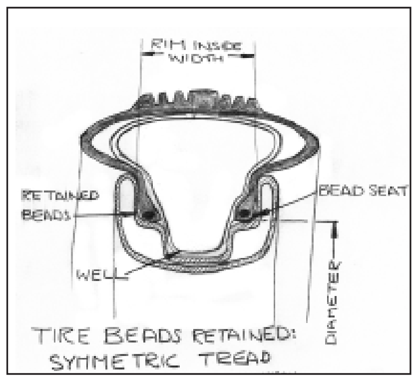



**Figure 3. Tire beads retained: symmetric tread**

**Recommendations**

The International Standards Organization should form a committee of tire and rim manufacturers to agree on standards of rim diameter and shape and of tire-bead diameters so that a tight fit could be relied upon in all circumstances. Then manufacturers should agree to observe these standards.

—*Dave Wilson*

*<dgwilson@mediaone.net>*

*Oury, Andrew P. (1999). “Run-flat performance of bicycle tires and rims” Thesis, B.S.M.E., M.I.T., Cambridge MA.

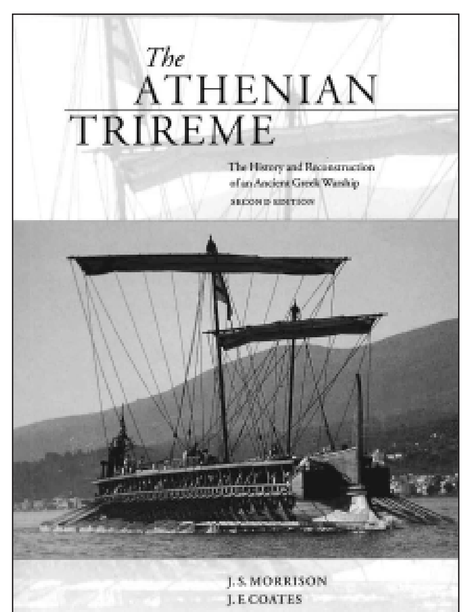
## *Reviews*

**THE ATHENIAN TRIREME, 2nd ed., by J. S. Morrison, J. E. Coates and N.B. Rankov, Cambridge University Press. ISBN 0-521-56456-5 (pbk.)**

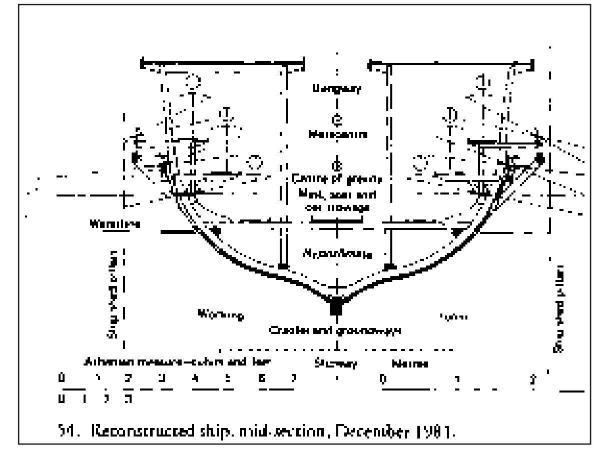
***Reviewed by Theo Schmidt***

*The Athenian Trireme* is a comprehensive report on the reconstruction and testing of a Greek trireme with 170 rowers, probably the largest and most successful such project in recent times, even if the object of interest dates back to several hundred years BC. These Greek warships were numerous and very successful in their day, but none survive, as they were unsinkable and were thus never preserved in bottom mud as were some other ships. It may also be characteristic of the time that a great deal was written about the battles in which these ships were used, but almost nothing about the vessels themselves. Well known is their technique of ramming and holing other ships with a protruding and strengthened bow.

Part I of the book, which was originally published in 1986, starts with painstaking detective work assembling and correlating the meagre data available such as not-to-scale pictures on vases, coins and reliefs. It is not even clear whether the name trireme (Greek: trieres) actually means three vertically displaced levels of single-oared rowers, whose feasibility the project aims to prove, or single levels of oars manned by several rowers each, which seems to be the more common arrangement depicted by artists. The pros and cons of the different arrangements are described as well as the reasoning which led to the reconstruction plan of the Olympias, which was then launched in 1987. However before this, models and even a full-scale floating section were built in order to study the rowing geometry. The ship was built with more or less traditional materials and methods, but not exclusively, as the major interest lay in operational research and not historical ship-building. It took the combined resources of two trusts (one British, one American) to build the Olympias, and also of the Hellenic navy, which owns and operates it.




**Right: Cover of the latest paperback version of this report on the reconstruction and testing of a Greek trireme. The first edition was published in 1986 (ISBN 0521564190) and as a textbook in 1990 (ISBN 0521311004). Hardcover edition of the 2nd ed., ISBN 0521564190**




**From the book jacket: “For this second edition, the text has been recast and a number of substantive changes have been made in the light of the sea trials and new research. In addition, there is an entirely new chapter which describes the trials of Olympias in detail, reports the performance figures obtained, and outlines the changes which the authors would wish to incorporate into any second reconstruction. There are nineteen new illustrations, including eleven photographs of Olympias at sea demonstrating features of the design which could be represented only by drawings in the first edition.”**

Part II is completely new in this second edition and describes the results of numerous sea trials in the years 1987–1992. The Olympias nearly lives up to its expectations, but its 5- to 5.5-knot cruising speed is slightly slower than hoped for. Peak speeds over 8 knots were recorded. The total efficiency of the oar system was estimated at about 1/3. Rowing the Olympias is hard, difficult, and sometimes unpleasant work, but apparently there is usually no shortage of volunteers, whether civilian or navy. Only the top level of rowers can see their oars, whereas the bottom-level rowers can see nothing in their poorly ventilated and smelly workspace (where they are also being dripped on by the sweat from the upper rowers). The logistics of operation are remarkable: for example, a day’s journey requires that almost two tons of water be on board for consumption by the rowers. Olympias also performs well under its two square sails. Combined sailing and rowing allows slightly higher speeds with a bit less effort.

The report concludes with a discussion of the results and suggestions for slight improvements. However, it appears that the Athenian Trireme project was just about optimal, and the same can be said for the book.

—*Theo Schmidt*

**THE DANCING CHAIN by Frank Berto, Ron Shepherd and Raymond Henry. ISBN 1-892495-21-X, available from Van der Plas Publications, San Francisco, CA or, signed, from Frank Berto <fberto@ix.netcom.com> for US$58.00 including shipping in the US, more for other destinations.**

***Reviewed by Dave Wilson***

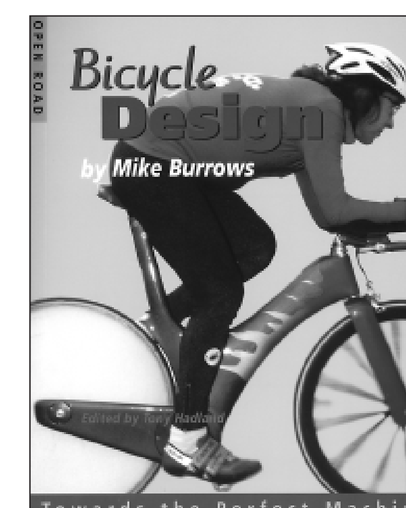
This is a large, beautiful, hard-cover, profusely illustrated and comprehensive book on derailleur gears. (It also has intriguing paragraphs on hub gears, bicycle companies, bicycle magazines, the people who invented, developed, sold and rode the gears and bikes, and much more.) I have been interested in bicycle gears for many decades, but I found myself continually saying "Wow!" to myself as I learned about aspects of gears that I had long wanted to know. I used to look forward to every article of Frank Berto's on his exhaustive tests of new derailleurs in Bicycling and other magazines, and I have great respect for him and his two co-authors, Raymond Henry from France and Ron Shepherd from Australia. Walter Ulreich of Germany and Tony Hadland of Britain also contributed to and checked parts of the book. Van der Plas Publications have put the same effort into producing this book as it does for the series of proceedings of the conferences on bicycle history.

Every reader of *Human Power* should, if you can afford it, buy this book. It gives you all the history, the fundamentals, the reasons for continual changes in design, the pitfalls to avoid, and so on that you will ever need. It may seem expensive, but it has been produced at what is likely to be a considerable loss (even if the authors receive no compensation for their years of dedicated work). It has been published by Frank Berto because no publisher would undertake so large a publication at so little possibility of sales sufficient to cover expenditures. Accordingly, there are only a few available. Rush to get your copy! And give thanks to Frank Berto and his collaborators for their major contributions to the human-power movement.

**BICYCLE DESIGN by Mike Burrows. Open Road Publishers, UK, ISBN 1-898457-07-7; US edition by AlpenBooks Press, ISBN 0-9669795-2-4**

***Reviewed by Dave Wilson***

Mike Burrows is the best-known and probably the foremost bicycle designer in the world today. He is also one of the top designers and builders of HPVs. He has therefore done a great deal to bring together two of the very different branches of bicycles and bicycling and to endow HPVs with respect from the “regular” biking community. He is also an everyday bicyclist and a racer in his Windcheetah Speedy tricycle or his



Ratcatcher short-wheelbase recumbent bike. Several years ago he was hired by Giant of Taiwan, one of the largest bicycle manufacturers in the world. His influence is therefore already major and is likely to increase.

Mike's book on bicycle design has been eagerly awaited. When Open Road Publishers failed recently we were concerned that we would not get to see it, but we are fortunate, at least on the American continent, that AlpenBooks has picked it up. It is a sturdily bound paperback of 160 pages, on bright-white stock, with some color "centerfolds" of "Mike's favourite bikes." All the photographs are clear and good, B/W and color, as are the diagrams. There are also several excellent cartoons by Jo Burt and Geoff Apps.

The book starts with a gracious foreword by Richard Ballantine, paying tribute to Burrows' many characteristics, including some that he has recently learned: diplomacy and gentle advocacy, which increase his effectiveness as something close to a revolutionary. Then we plunge into what I can only describe as pure Burrows: fifteen chapters of Mike's strong views on everything from ergonomics to monoblades and cantilever wheels. They are well written (or well edited by Tony Hadland) and expressed with nice modesty as well as pride in his many innovations, which he often credits to others. For instance, the monoblades and cantilever wheels he saw on an 1889 cross-frame "Invincible" in a museum. (He is also kind enough to state that he wants his book to fit in the gap between *Richard's Bicycle Book* and Whitt and Wilson's *Bicycling Science*. He succeeds superbly! He wanted no algebra or equations, and he managed that.) He apologizes that his book is written from a British perspective. It is, but he gives credit to non-Britons. The Giant company also comes out well.

Many of the topics that occupy much discussion space on HPV and bicycling mailing lists would be enlightened by Burrows' trenchant observations and opinions. In discussing frame design he draws a distinction between torsional stiffness and vertical compliance that makes a lot of sense. His guidance on deep-section "aero" wheels, which can have very stiff rims, and the need for more-forgiving rims on all-terrain bikes, and his frank statements on what he doesn't like, are all high-value and high-octane. Likewise his comments on suspension, braking and monoblades are pure common-sense that isn't as common as we would like.

As it happens, I'm in what I hope are the closing stages of writing the third edition of Bicycling Science (with Jim Papadopoulos), and I am working on the chapter on the future of bicycles. I've realized that just about everything that I hope to see in future bikes, like cantilever wheels that one can change rapidly when one has a flat or when one wants to put on a studded tire (as on the day of writing), and all-enclosed transmissions, and disk brakes, have been worked on by Mike Burrows. The man is a master and his book is a "must read".

*—Dave Wilson*

## GUEST EDITORIAL

### YOUR NEXT VEHICLE: A VELOMOBILE?

***Joachim Fuchs***

Velomobiles are fully faired recumbent vehicles for everyday use. Many people consider that they have the potential to play an increasing role within different types of human-powered vehicles. In addition, they could give a positive contribution on our future traffic. Or, more precisely: can fully faired everyday recumbents replace cars and normal bicycles? This article gives a view over the recent developments in Europe.

First of all, velomobiles are human-powered vehicles that differ from normal bicycles in function and appearance. There are many types, produced as prototypes and in small-scale manufacturing. Velomobiles are fully faired recumbent cycles that are constructed for everyday use and provide full rain protection. The fairing also gives better protection from accidents for the driver.

An important question is: Why should I use a velomobile, and what are the advantages compared to a bicycle on the one hand, and a car on the other hand?

An obvious example of the difference between the rider of a velomobile and that of a normal bicycle is that users of velomobiles wear almost the same clothes in summer and winter. This is one main argument for velomobiles: there is no need for a look outside in the morning. No shapeless rain suits hinder one's pedaling. In addition, both women and men can ride in business suits if they wish.

This implies that there must be good ventilation, an important factor in velomobile design. Because of the absence of direct wind, adjustable air flaps are integrated in the fairing. The air stream within the fairing is moderate compared with that on an unfaired bicycle.

Therefore, the rider learns to moderate his or her own power. My own experience shows that one sweats less in a well-ventilated velomobile even in summer. In contrast, on a normal bicycle one is getting "blown dry" by the wind and sweating starts intensively after riding. This is unpleasant when riding to work regularly. Properly mounted air flaps within a fairing can avoid this effectively.

When riding uphill, passive ventilation does reach its limit, on a regular bicycle and in a velomobile, because the speed of the vehicle is not enough to produce a sufficient air flow. The question is often asked: "Is it possible to ride uphill in a velomobile?" Velomobiles are around 15 per cent heavier than upright bicycles if the rider is included in both cases. The speed loss uphill can therefore never be larger than this 15 per cent. On small or moderate gradients uphill, the lower air resistance of the velomobile compensates for this disadvantage. Velomobiles normally have a smaller effective frontal area (which governs the air resistance). This is the reason for the higher speeds that can be reached with some velomobiles. Higher speeds are attractive especially for riders that are used to physical exercise and have fun riding with their own power. Those people who like riding at 1.5–3 m/s (5–10 mph) will not feel a difference. With a little more power input, riders who are not very sportive become astonished when they can ride at 13 m/s, 30 mph, for some time. This is indeed possible with "sportive" velomobiles.

Different velomobiles present at meeting in Oktober 2000 in Germany. From front to rear: Leitra, Aeolos and Cab-bike. In contrast to the other vehicles, Aeolos (a development of the author) is a two-wheeler. Further informations can be found at http://www.velomobile.de

A young velomobile enthusiast during a test... Children have a lot of fun sitting in velomobiles even they cannot see through the windshield!

There are, naturally, many different kinds of velomobiles. Most velomobiles are tricycles. They are stable, anyone can ride them immediately, and they have good luggage capacity. Two-wheelers are ridden by sportive people because they can normally go faster and can lean in turns. Examples of such velomobiles in Europe are *Aeolos* and *Desira*. In everyday use, the handling is very important. Getting in and out should not be hindered by the fairing. This is the precondition for switching from a bicycle to a velomobile: it should be practicable for short distances (buying bread rolls on Sunday morning…).

A velomobile that exhibits its advantages only in rainy weather would not find many users, because the additional place to park such a velomobile is a problem at least in urban cities in Europe. To some extent, one can say that velomobiles combine the advantages of cars and bicycles. As bicycles, velomobiles can be used on roads on which cars are banned. Often, everyday distances can be covered through a nice landscape whereas car drivers have to use boring main roads. Besides that, velomobiles are economic even though they are expensive when purchased. This is especially the case if velomobiles are often used and if they replace other vehicles. Compact velomobiles can be taken along in a train with a bicycle compartment, at least in some countries in Europe. Some designs can be taken apart to make them easier to stow, which is necessary with most tricycles. People who like to ride with other (non-velomobile) riders should take

The variety of velomobiles indicates that there is still potential for further developments.

into account that one can chat while pedaling. This can also be useful for communicating with other traffic users, mostly car drivers. A properly constructed velomobile can be pushed along sidewalks and shopping malls. (You will have a built-in shopping cart!) Most enthusiasts first think of rain protection when the talk is about velomobiles. On the one hand, there are sly unfaired riders who calculate that it rains only a small amount of time. On the other hand, in practice, people who get wet once are more likely to use the car next time.

To match the demands of practicability, constructors have to design their products with considerable skill. Protection from cold wind in winter time is as important as from rain. On a normal bicycle, it is often hard to choose the right clothes. After commuting some time, one begins to sweat under the warm clothes. Velomobiles avoid this problem, because one can adjust the air flow and do not have to change clothes. And what is the feeling when getting in a velomobile? First of all, velomobiles are quite narrow. This is a necessary property, because velomobiles should be built light in weight and compact to consume only little space when parking. The feeling in a velomobile is individually quite variable. Some elderly people feel ill at ease in the fairing even if they don't touch it. Others report that they feel safe and secure in the fairing because of the protection effect. The well-being is further influenced by other factors. The sight through the windscreen for example should not be affected by reflections. A velomobile should have a low noise level inside the fairing. All this contributes to the feeling that is specific for velomobiles. Most test riders get along quickly with the "new" vehicle. The main advantage of an "ideal" velomobile is that it is the proper means of transportation in most everyday situations. Thus, partially faired vehicles with the head outside the fairing have the disadvantage that the head might have to face a very strong wind. Nevertheless, some bicycle riders choose that kind of recumbent vehicle, believing that they have more advantages than upright bicycles. Velomobiles moreover offer a "built-in" rain protection, advanced aerodynamics and a protection from cold wind—and all that in one vehicle.

There are several velomobiles commercially available. The first velomobile to attain widespread use was the *Leitra*. The *Leitra* follows the classic concept with a lightweight steel frame and a glass-fiber fairing fixed to the frame. This offers the advantage that there is less noise than in a "monocoque" vehicle. Later velomobiles often use a construction that is easier to realize. One example of that kind is the Cab-bike. Such velomobiles don't have a frame in its own sense. Instead, the fairing forms a closed shell with all components mounted to it. Vehicles of that kind have fewer parts and are cheaper to produce. But in the case of damage, it is necessary to repair it skillfully to ensure the shell recovers its rigidity. There are further aspects to take into account, for example eye-level height, which is important in urban traffic. Besides the commercially available velomobiles there are some vehicles that were either produced for personal use or have at least the potential for a commercial product. The inventors of prototypes add to the diversity of velomobiles. To give some examples: *Veleric*, *Hajen*, *Jouta*, *Desira*, *Pedicar*, *Muscar*. Each vehicle was constructed for different purposes; the *Desira* even exists in several versions.
Although velomobiles offer lots of advantages, one should remember that velomobiles serve a niche market. The price of more than approx. US$5500.00 is far higher than that of most upright bicycles. The breakthrough would be if everyone could find a velomobile that fits the demands of daily commuting.

Can you see yourself in a velomobile soon?

### References


A lot of information can be found on the internet:
General information: www.ihpva.org
www.velomobile.de This is an internet platform for velomobiles that was just started (please have a look on it now and then to get current information). The velomobiles mentioned in this text are introduced or linked there.
Several publications in the Proceedings of the European Seminars On Velomobile Design I- IV. Further information at www.futurebike.ch
Printed papers:
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Dovydenas, Vytas. 1990. *Velomobile.* Berlin: Verlag Technik GmbH [ISBN 3-341-00790-3; originally published in Lithuania and translated into German; out of print]


*—Joachim Fuchs <j-fuchs@foni.net>*

## Editorials

### The end of a dream

My principal activity other than work on *Human Power* seems to be trying to finish (with Jim Papadopoulos) the third edition of *Bicycling Science.* I have been working on the chapter about what we can expect and what we might want in our future bicycles. The easy way out was to refer readers to *Encycleopedia* and *Bike Culture Quarterly* and other publications of Open Road. But the shocking news of the bankruptcy and the laying off of all employees of Open Road has just hit us. It has seemed to be a miracle that the company could do what it did: to produce (since 1993) a series of superbly produced texts and magazines and videos on alternatives in cycling. Every issue of everything it did was not only a resource for the cycling enthusiast: every photo was so beautifully done and reproduced that each item of output became a "coffee-table book". I'm using that as a term of admiration, not disparagement. Each publication could be left on a table at a doctor's office and would be guaranteed to be looked at with wonder by a wide range of people. Thus it spread acceptance and even respect for the more adventurous, and the quirkier, human-powered vehicles. We enthusiasts would seize each issue of each series and be inspired by the quality of the publication and by the ingenuity of the subjects. We marveled that this could be done without advertisements of macho trucks and SUVs on every other page. Open Road had been going from strength to strength for the better part of a decade, publishing in English and German, with agents in four countries, organizing "Bike Culture Weeks" in its home territory in and near York, UK, and, lately, publishing two superb books.

I had thought that Open Road must have an "angel funder", in the way that the early IHPVA had infusions for prizes from Du Pont especially, but it seems that there was none. The speed of the collapse of Open Road, and the large amount of debt at the end, are saddening and sobering. We in the new, reorganized IHPVA are far less ambitious, even timid by comparison, and yet we are hanging by a slim financial thread. The temptation to draw some parallels is irresistible: if we want superb publications like those of Open Road to continue, and the less-ambitious but irreplaceable magazines such as *Recumbent Cyclist News (RCN)*, *Recumbent UK*, and all the other publications of our national and local HPV associations (in which we hope we may include *Human Power*), we must support them with subscriptions and with the recruitment of others to join. People like Jim McGurn and his associates at Open Road performed the miracles they did in the spirit of missionaries with a vision, at low or zero pay. Many selfless people also invested in a dream, and have lost all their money. We give heartfelt thanks and appreciation to all involved for their all-too-short period of brilliance, one that shone on us all. We hear that some of the former staff have plans for new publications to try to carry on the tradition, and we wish them god speed.

*—Dave Wilson*

### Tiresome

Pneumatic tires were patented twice, in 1845 by Thomson and in 1888 by another Scot, Dunlop. (Patent procedures can still be as capricious.) When I was a child, motor-tire failures were to be expected in regular driving. Nowadays a flat on almost any motor vehicle is, or was, very rare. Racers go around the turns of Indianapolis and the twists of European Grand Prix circuits at over three-hundred km/h, at very high tire temperatures, with amazing reliability considering the conditions. At one time the favorite tires at "Indy" were Firestone. So how did it come about that Firestone tires were implicated in many failures in Ford Explorer vehicles at far-lower speeds and temperatures? How could Ford design a vehicle that would roll over after an event as expectable as a tire failure? And how could Ford make a vehicle (on which its profit margin is allegedly very high) that, when it rolled, had no inbuilt roll cage, so that it crushed passengers still in the vehicle?

Tires have also been blamed for the crash of the supersonic Concorde. The investigators have tentatively concluded that, during a take-off run, one tire or pair of tires picked up a piece of metal that was on the runway, and either spun it off like a projectile into a fuel tank, or spun off pieces of tire that perforated the fuel tank(s). This seemed to be a horrible piece of bad luck, until reports were aired that tires on Concordes had failed in similar fashion more than once previously. So had engineers or managers just wished the problem wasn't going to recur? An approximately similar number of people died as allegedly did from the Firestone-Ford tire problems.

In this issue of *Human Power* we report on a problem with bicycle tires: a flip-flop behavior that can throw riders suddenly off their machines when a front tire deflates. It appears to be caused simply by poor fits of tires on rims. If this is so, it could be solved quickly by industry agreement on the dimensions of rims and tires, spurred possibly by government specifications. We don't know how many lives have been lost from this unnecessary series of failures. Bicycle accidents are not taken seriously enough to be investigated in depth. There has been no outcry. Your editor's letters to the U.S. Consumer Product Safety Commission and to several industry organizations have remained unanswered.

Remedies for bicyclists have the same status as so-called "orphan drugs". These drugs are not developed for fatal but relatively rare diseases because drug companies see insufficient profit. Is the bicycle-tire-rim case a situation where industry is not being sued enough? The much-maligned product-liability lawyers can correct serious deficiencies in industry responses, or lack of responses, to shoddy practice.

*—Dave Wilson*

### Human Power publishing record, 1995–2000

*Human Power* 11:4 (Fall/Winter 1994–95)
*Human Power* 12:1 (Spring 1995)
*Human Power* 12:2 (Fall 1995)
*Human Power* 12:3 (Winter/Summer 1996)
*Human Power* 12:4 (Spring 1997)
*Human Power* 13:1 (Fall 1997)
*Human Power* 13:2 (Spring 1998)
Volume 13:3 (Summer/Fall 1998)

In 1998 we moved to a simpler numbering system for *Human Power*, since we are not able to publish on a regular, pre-defined schedule.

Adding up all the issues we could find back to issue 1:1, we numbered the next issue #46. After a long-time member noticed that we had left out a number, we re-numbered that winter 1998–99 issue #47.

Thus, for 1999 and 2000, we published the following:
*Human Power* 48 (Summer 1999)
*Human Power* 49 (Winter 1999–2000)
*Human Power* 50 (Spring 2000)
*Human Power* 51 (this issue)

We expect to publish at least two issues of *Human Power* in 2001 and have already begun work on *Human Power* 52.

# International Human Powered Vehicle Association

IHPVA
PO Box 1307
San Luis Obispo, CA 93406 USA
http://www.ihpva.org