Materials • Mechanics • Physiology • Engineering • Aerodynamics

BIKE TECH

Bicycling Magazine's Newsletter for the Technical Enthusiast

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SHOP TALK

Why do Cranks Break?

Back to the Drawing Board

John Booth Davies

Editor's note: Why do cranks sometimes break? A series of readers' letters on this question appeared recently in the British Cycling, *ranging from the barely credible (one individual with ten crank failures) to others who have pounded the same crank for years with no problem. Dr. John Booth Davies, a Senior Lecturer in Psychology at University of Strathclyde, asked all readers who experienced such failures to write him a letter with the details. He said, "the only advice given is to tap the cranks on with a wooden block or mallet, thereby apparently wrecking the bearings, or also to search regularly for hairline cracks with a magnifier. Given the potentially disastrous consequences of crank breakages, I think the time has come to investigate this matter more thoroughly." Here are his results. If any* Bike Tech *readers are interested in conducting calibrated destructive tests of cranksets, please let us know.*

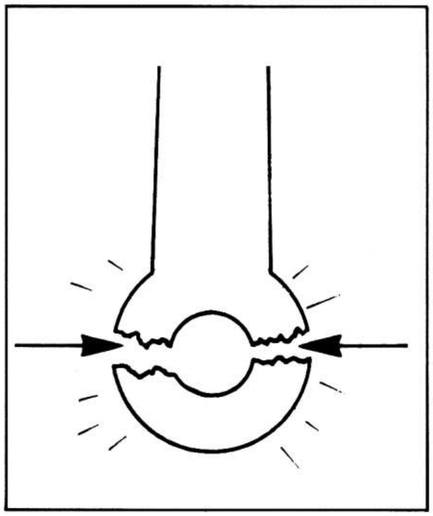
In response to my letter I received reports of 44 crank breakages from readers. Some readers sent photographs, pieces of crank, and even a complete chainset. I am extremely grateful to all those who took the time to reply. Below are some preliminary comments, though I hope to take things further with the help of a metallurgist at some time in the near future.

The first thing to emerge is that much of the previous correspondence may have been aimed at a relatively minor issue, namely failure of cranks at the bottom bracket spindle. Only three of the 44 reports concerned this type of failure, whereas 33 readers reported breakages at or near the pedal spindle. Of these, 27 reported failure actually through the pedal eye, and readers' comments plus the broken bits I received confirm a highly characteristic failure pattern at this point.

The failure actually involves two tears, one on each side of the pedal eye, the second presumably precipitated by the first (see figure 1). The samples I received show areas of dark and bright metal in the broken surfaces, suggesting that the crack develops over a period of time and should be detectable before total failure occurs. I cannot tell whether there is a tendency for the leading or trailing crack to develop first, but the siting of the dark areas is identical on all the samples I received, and indicates that the crack develops in the outside face (of either crack) and works its way inward.

Readers cited a fairly broad range of manufacturers' products, and I list these here for

Figure 1: Typical failure points of crankset shown by arrows.



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interest. It is vital that one does not make careless inferences from this list because (a) readers of *Cycling* are not a fully representative cross-section of the cycling community, they tend probably to be more keen on their hobby and perhaps subject their machines to more stress, and (b) the sales figures for the various brands are not known. Thus, the fact that three of the best-known and trusted brands head the table for crank-failures may merely indicate that more keen cyclists use their products. Breakages by brand were—

Stronglight	**13 failures**
(11 of these 49D)	
Campagnolo	**9**
TA	**9**
Ofmega	**3**
Silstar	**2**
Sugino	**2**
Galli, SR, Zeus	**1 each**
Unnamed	**3**

While these figures cannot be used to make relative reliability judgments, they do show that failures can and do occur in some numbers, even among the most prestigious brand-names. The regularity with which the 49D is cited seems worthy of more examination, as is the absence of anything from Shimano, who are, I think, one of the market leaders. Has anyone broken a Shimano set, I wonder?

I have one or two further comments. First, while failures occurred most often at the pedal, they were also reported in mid-crank and at the bottom bracket. This suggests that overall there is something marginal about current designs, and the materials used to implement those designs. For example, a pair of cut-out Zeus cranks I examined at Billy Bilsand's shop had bent as well as torn apart at one side. Maybe a little more "meat" is worth the extra weight.

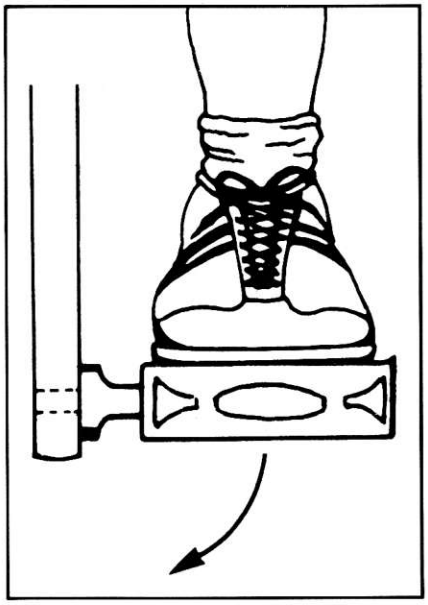

Figure 2: Sideways bending moment due to pedaling force.

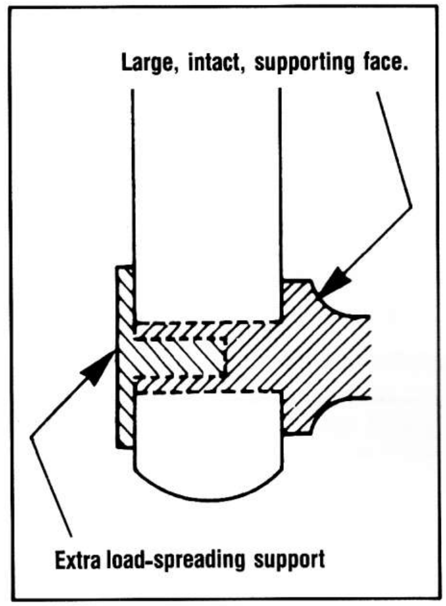

Figure 3

The failures at the pedal eye are interesting. The nature of the breakages suggests a crack starting in the outer crank face, and this is consistent with many readers' reports that the breakage occurred when they were out of the saddle either climbing or moving away from lights. The position of the pedal on the crank means that considerable sideways movement is imparted towards the non-pedal side. This is accentuated when "honking," since the rider uses the upper body and arms to lever the bike against his/her feet, both increasing the sideways movement and, as the bike tilts, making its angle even more acute (see figure 2).

It seems to me that an improvement in pedal design might possibly help with the problem. At the present time, some pedals have a complete round "washer" face which butts against the crank; whereas others have only a partial butting face, with the metal completely removed at the flats for the pedal spanner. If these flats, representing potential support which would spread the sideways movement over a larger area if it was there, take up a particular position when fully home, there may be an absence of support in the critical area.

This problem will not be effectively overcome by using loose washers, which prevent surface damage to the crank, but are not a rigid, load-spreading part of the pedal itself. My feeling that pedals as well as cranks might be improved is supported by reports from two "multiple crank breakers," both of whom reported shifting the same pedals between their numerous sets of cranks. More support would be provided by a larger, rigid, complete washer-type face on the pedal side. Furthermore, if the pedal axle were itself drilled and tapped, a further bolt with a large washer-face could be screwed in from the other side, offering further support (see figure 3).

All of this would, of course, be helped by beefing up the sides of the pedal eye. Finally, although the dangers of overtightening have been stressed many times before, it will be apparent that any load-sensing function provided by the flat face on the pedal spindle will be nullified if this is not snugged securely against the crank face. A pedal which almost (but not quite) goes all the way home (at which point the spanner-operator may well desist for fear of overtightening) will place considerable stress on the pedal eye through not properly supporting itself, and thus concentrating its sideways movement in a very small area.

Ultimately, however, I feel the fact that cranks can and do fail at the pedal, in the middle, and at the bottom bracket, suggests that a new chunkier design is called for. All the tinkering about with torque wrenches, pedal washers and what-have-you will provide at best only stop-gap solutions if the problem lies in marginal crank design. I hope, presently, to pass on the views of an engineer and a metallurgist, but my feeling is that much light could be shed by a series of properly conducted destructive tests. Anyone interested in having his cranks smashed up in the cause of science?

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A "Standard" Crankset Flexibility Test

Robert G. Flower

The original goal of this research was to shake out the controversies in crankset design. We have all heard claims and counterclaims: "Aerodynamic cranksets are too flexible," "Cranks with an offset crankarm are more rigid," and so forth. The curious fact is: there is nothing even close to an established procedure in the bicycle industry for measuring or comparing the flexibility of cranksets.

The Rodale Press Product Testing Department, which provides technical services to *Bicycling* magazine and *Bike Tech* on occasion, was called into action. My job, as engineering consultant to the Testing Department, was to develop a standard crankset flexibility test. This test is described here in detail. The second task was to test a group of cranksets that are typical of what is available on the market today. These results are reported here, and also appeared in shortened form in *Bicycling* magazine, Sept/Oct 1984. In a word, we found that the most flexible crankset tested was about 30 percent more flexible than the stiffest.

The third task was supposed to be figuring out what design features made the "stiffest" crankset. We started by calculating the amount of elastic strain energy absorbed by each of the cranks (a direct measure of the "lost motion" which the rider experiences) under hillclimbing loads. The results were a surprise: even the most flexible crank tested was not a major contributor to lost motion. Less than 1.7 percent of the rider's pedaling energy would be dissipated by flexing of this most flexible crank. For the stiffest set, the figure would drop to about 1.3 percent. The difference between the two, 0.4 percent, might make a difference in track competitions (if they included hillclimbing), but would probably be undetectable in outdoor road riding.

We've concluded from all this that it's not all that important to find the stiffest crankset design. Instead, factors such as weight and chainring interchangeability should be considered.

Test Equipment

Our two main concerns in developing the flexibility test were: 1) the loading and constraint conditions should be as realistic as possible, and 2) the test setup should use simple and readily-available components so that others could replicate the results.

The basic test stand (see photo) is a steel post (2 3/4 inch square tubular bar, 1/4 inch wall) welded to a 1/2 inch thick steel base plate and threaded with standard bottom bracket threads (1.34 inch x 24 TPI). To provide an "unmoving" sprocket against which the chain can pull, we welded two standard freewheel bodies to side plates (3/8 inch steel); these plates also serve to strengthen the vertical post which holds the bottom bracket. The freewheels are mounted "backwards"; that is, they are free to rotate in the direction *opposite* to the chain pull. This provides a large degree of adjustability when aligning the crankarm to a prespecified angle for the tests.

A standard pedal spindle (without pedal) was threaded into the eye of each crankarm; in fact, the *same* pedal spindle was used on all the cranksets tested so that any flex in the pedal spindle itself would be the same in all the tests. Later measurements found no detectable flexing of the pedal spindle. The force representing the rider's leg-force was supplied by a pneumatic actuator ("air cylinder") fitted between the pedal spindle and base plate. A set-screw collar held the piston rod of the air cylinder in position on the pedal spindle. This position was chosen to represent the center of a typical pedal.

We used the following pneumatic components, which we found to provide an accurate and flexible system for a variety of mechanical tests in the shop:

- —actuator: **Bellofram Diaphragm Air Cylinder**, Size 6; nominal 2.8 inch bore diameter, 2.4 inch stroke length, approximate price $80 (Bellofram Corp., Burlington, MA).
- —pressure regulator: **Bellofram Precision Air Regulator**, Type 41; inlet pressure 250 psi max., outlet pressure adjustable 0 - 120 psi, approximate price $20.
- —pressure gauge: **Helicoid Test Gauge**, Type G1D; 0 - 60 psi, 1/4 percent accuracy, approximate price $60 (Helicoid Div., Bristol-Babcock Inc., Waterbury, CT).

It is necessary to calibrate the "bore size" of the air cylinder, so that the pressure measured on the test gauge can be converted to the force exerted by the actuator. We performed this calibration by using a digital laboratory scale (0.1% accuracy) to measure the actual force exerted at a number of psi readings; back-calculation then found that the cylinder's actual bore area was 6.1575 in^2.

The test arrangement shown in the photo corresponds to a vertical downward force applied to the pedal on the right side of the bicycle, with the crankarm horizontal. Other arrangements are possible: the crankarm can be set at any angle to the horizontal simply by adjusting it to the desired angle when installing the chain. Our tests used 0, 30, 60, and 90 degree angles, as measured on a v-

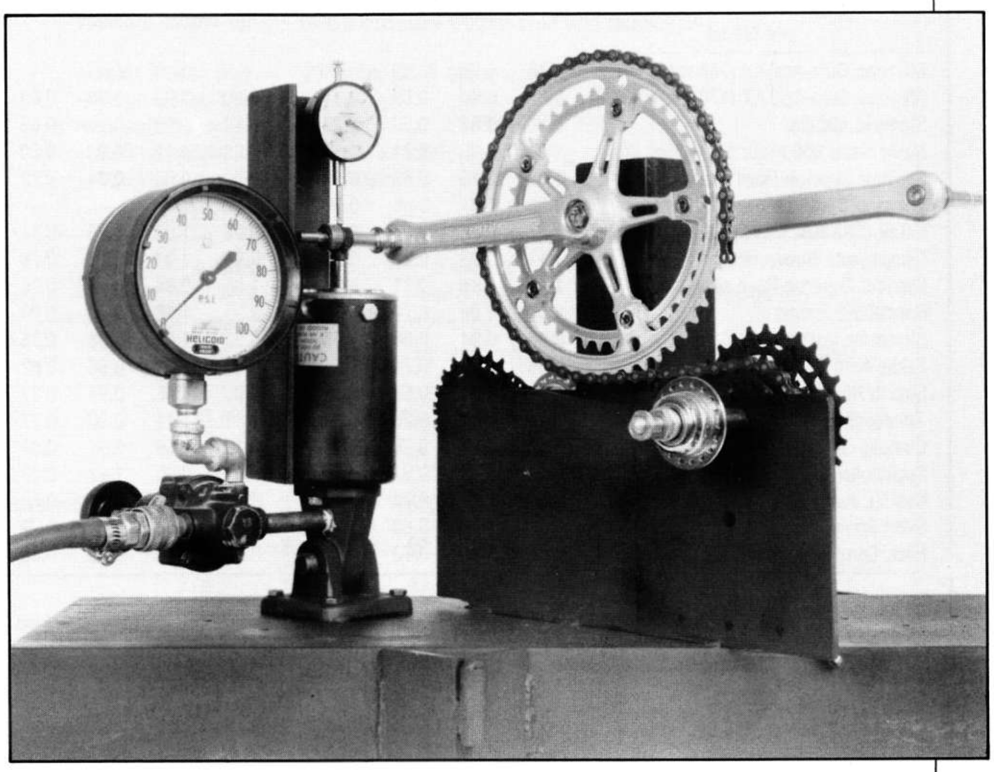


Table 1: Measured Flexibility of Cranksets (thousandths of an inch per lb force applied)

Force applied to ------------------------------> **LEFT PEDAL**

Direction of applied force ------------------>	**Vertical Force**				**15 deg Inward Force** *(Note 1)*			
Angle of crank *(Note 2)* ------------------->	0	30	60	90	0	30	60	90
Manufacturer/Distributor and Model								
Shimano Dura-Ace AX (165 mm)	1.90	1.50	0.82	0.45	—	—	—	—
Shimano Dura-Ace AX (170 mm)	1.91	1.47	0.75	0.47	1.80	1.58	0.98	0.72
Shimano 600 EX	2.05	1.60	0.81	0.43	1.86	1.53	0.95	0.65
Mavic Serie 600 *(Note 3)*	2.11	1.79	0.99	0.56	2.02	1.88	1.19	0.88
SunTour Superbe Road *(Note 4)*	2.12	1.70	0.92	0.54	1.97	1.69	1.04	0.79
Shimano Dura-Ace AX (175 mm)	2.12	1.60	0.92	0.50	—	—	—	—
Sakae CRC 301	2.19	1.71	0.87	0.45	2.11	1.83	1.08	0.74
Campagnolo Nuovo Record Road	2.21	1.72	0.91	0.55	2.13	1.69	1.08	0.74
SunTour Superbe Road *(Note 5)*	2.01	1.81	1.01	0.63	2.01	1.79	1.31	0.95
Specialized Racing	2.20	1.78	0.99	0.59	2.07	1.84	1.22	0.85
Gipiemme Special	2.23	1.83	0.97	0.55	2.00	1.67	1.09	0.77
Sakae Aero X	2.24	1.89	0.92	0.52	2.14	1.72	1.09	0.73
Sugino Aero Mighty	2.25	1.81	0.94	0.51	2.18	1.86	1.12	0.74
Stronglight 106	2.26	1.88	1.01	0.62	2.02	1.78	1.06	0.77
Campagnolo Gran Sport	2.26	1.80	0.98	0.62	2.08	1.80	1.13	0.81
Sugino Aero Tour	2.34	1.79	0.90	0.48	2.21	1.80	1.04	0.72
Galli XL Aero	2.47	1.98	1.03	0.56	2.34	1.97	1.20	0.81
Excel Rhino	2.52	1.89	0.92	0.57	2.23	1.87	1.05	0.70
Edco Competition Aero	2.56	1.91	1.07	0.64	2.38	2.03	1.24	0.84

Force applied to ------------------------------> **RIGHT PEDAL**

Direction of applied force ------------->	**Vertical Force**				**15 deg Inward Force** *(Note 1)*			
Angle of crank *(Note 2)* ------------------->	0	30	60	90	0	30	60	90
Manufacturer/Distributor and Model								
Shimano Dura-Ace AX (165 mm)	0.86	0.74	0.60	0.48	—	—	—	—
Shimano Dura-Ace AX (170 mm)	0.82	0.90	0.59	0.51	0.80	0.93	0.83	0.69
Shimano 600 EX	0.90	0.83	0.52	0.37	0.79	0.93	0.75	0.66
Mavic Serie 600 *(Note 3)*	1.03	1.03	0.71	0.56	0.95	1.12	0.94	0.80
SunTour Superbe Road *(Note 4)*	1.04	0.16	0.65	0.51	0.90	0.96	0.84	0.72
Shimano Dura-Ace AX (175 mm)	0.98	0.84	0.66	0.56	—	—	—	—
Sakae CRC 301	1.01	0.86	0.64	0.51	0.99	1.03	0.86	0.71
Campagnolo Nuovo Record Road	1.11	0.99	0.69	0.55	0.97	1.04	0.89	0.78
SunTour Superbe Road *(Note 5)*	1.10	0.95	0.71	0.53	1.08	0.96	0.89	0.76
Specialized Racing	1.05	1.04	0.74	0.63	0.99	1.16	0.93	0.94
Gipiemme Special	0.98	0.94	0.68	0.55	0.90	1.08	0.86	0.75
Sakae Aero X	0.98	0.92	0.71	0.55	0.88	1.01	0.96	0.82
Sugino Aero Mighty	1.09	0.89	0.67	0.52	1.03	1.05	0.91	0.77
Stronglight 106	1.24	1.07	0.76	0.61	1.11	1.11	0.92	0.77
Campagnolo Gran Sport	1.19	1.04	0.76	0.58	1.12	1.09	0.91	0.87
Sugino Aero Tour	0.98	0.86	0.67	0.57	0.92	0.90	1.22	0.77
Galli XL Aero	1.10	1.05	0.74	0.59	0.99	1.08	0.91	0.84
Excel Rhino	1.08	1.01	0.69	0.56	1.03	1.08	0.92	0.79
Edco Competition Aero	1.06	0.99	0.73	0.56	0.98	1.05	0.89	0.83

Notes:
1. "Inward" applied force represents the situation where the rider "tosses" the bike from side to side, as in hillclimbing.
2. Angle measured with respect to horizontal with crank forward; thus 90 degree crank angle means bottom-dead-center.
3. Mavic Serie 600 tested with Campagnolo Nuovo Record axle and bearings.
4. Tested with sealed bearing axle.
5. Tested with regular bearings and axle.

base "angle-finder" strapped to the crankarm. Another variation is that the air cylinder can be attached at various locations on the base plate in order to apply "sideways" forces. These forces represent the situation (such as climbing steep hills) when the rider swings the bike from side to side and, in effect, pushes inward on the pedals.

Deflections In-Line

We measured deflection of the crankset to the nearest half-thousandth of an inch, using a standard dial indicator that was mounted *in-line* with the axis of the air cylinder. The dial indicator was mounted on a steel 90-degree angle channel attached (by band clamps) to the air cylinder itself. This in-line arrangement is an important detail: it provides the easiest way to calculate the elastic strain energy absorbed by the crankset. Recalling that strain energy (a scalar) equals the dot-product of the force vector times the displacement vector, we see that a simple multiplication (deflection measured in-line with force times force) is all that's needed. We will use this relationship later to estimate loss of pedaling efficiency caused by crank flex.

Test Procedure

An experienced mechanic installed each crankset on the test stand, setting the bearing cups for minimum free play, and using a torque wrench to apply 18 to 20 ft-lbs to the crank fixing bolts. Before starting each test, a 60 lb pre-load was applied to the pedal spindle, and everything was tapped with a plastic mallet to settle the chain and remove looseness. Then forces corresponding to 10, 20, 30, and 40 psi on the test gauge (approximately 60, 120, 180, and 240 lbs) were applied, and the resulting displacements were recorded. The air pressure was then reduced to about 5 psi (30 lbs force) and the cycle repeated. If the displacements on the second cycle differed from those on the first by more than +/- 0.002 inch, we rejected the data and re-tightened everything on the crankset and test stand before redoing the entire run.

A plot of applied force versus measured displacement showed that the data points came very close to falling on a straight line. In fact, the slope of this "best fit" line is the crankset's *flexibility* for the loading condition of the test. (Flexibility is defined here as the amount of deflection in line with the force per unit of applied force.) We calculated the slope of these lines, for each loading condition and each crankset, using the standard linear regression formula. In all cases, the resulting regression coefficient (r squared) was greater than 0.999, which indicates a very good linear relationship between force and deflection. The flexibility values obtained in this way are all listed in Table 1.

Aerodynamics vs. Weight: Quantifying the Trade-Off

Daniel Kirshner

Editor's note: We are beginning to see aerodynamic windshield-like fairings appear on all-terrain bikes. While the fairings certainly do enhance the rakish appearance of these machines, we have to ask whether wind resistance is that much of a hindrance at slow mountain-climbing speeds. Author Daniel Kirshner asks a similar question here about practical HPV design: in the stop-and-go, uphill-downhill riding of the commuter or day-tripper, is low air drag or light weight more important? His surprising answer is that they're about equal, and he defines a trade-off parameter to quantify exactly how important each of these two factors is in a variety of conditions. For example, if you've designed an improved HPV fairing that reduces the vehicle's drag by 10 percent, it better not increase the vehicle's weight by more than 10 percent; it if did, the extra weight would hurt more than the lowered drag helps. Aero enthusiasts and HPV designers will surely find much grist here for the mill.

Dan Kirshner rides his custom recumbent to work in the "flatlands" of Berkeley, California, where he helps organize a local chapter of the International Human Powered Vehicles Association. His undergraduate thesis in physics concerned the handling of bicycles (see "Some Non-Explanations of Bicycle Stability," American Journal of Physics, *January 1980). Dan would like to acknowledge the assistance and useful comments of Anthony Wexler in the work that led to this article.*

Cyclists know that extra weight slows you down. And through the efforts of the International Human Powered Vehicle Association (IHPVA) and others, cyclists are learning that aerodynamic efficiency can speed you up. But increased aerodynamic efficiency often entails extra weight, for example, for fairings, recumbent frame designs, and so forth. This article describes a method I have developed to calculate the trade-offs between aerodynamic efficiency and weight under a wide variety of riding conditions, including the power level of the rider, the slope of the ground, and the frequency of occurrence of stop signs. The results of this calculation are shown in the graphs and tables in this article; I expect that designers of future "practical" human powered vehicles (HPVs) could use this information to evaluate potential designs and design changes.

One surprising result: for a "typical" HPV under "typical" riding conditions, a small percentage reduction in the vehicle's weight is just as beneficial—in terms of overall efficiency—as the same percentage reduction in aerodynamic drag. This fact could be quite useful to designers because, at this stage of HPV evolution, shaving off a few pounds of weight might be easier than improving the aerodynamic efficiency of a fully-faired vehicle.

Measuring the Trade-Off

My basic assumption is that there is no such thing as perfectly level ground and that typical HPV riding conditions always include starts and stops. For the calculations presented here, I've assumed that the specified road course is a closed loop that includes equal distances of equal uphill and downhill slope (even nominally "level" ground has shallow uphills and downhills), and also includes stop signs at regular intervals.

My criterion of overall efficiency is average velocity (on the specified closed road course) for a given level of power output (i.e., muscular exertion) from the rider. To quantify the trade-off between aerodynamic efficiency and weight, I first calculate the overall average velocity which can be obtained with a "typical" HPV under the specified riding conditions. Then I change the aerodynamic efficiency and find the new value for the vehicle's mass which will result in the same average velocity. For example, if the aerodynamic drag is increased (leading to a lower average velocity), the mass must be decreased to attain the same average velocity. All of these calculations are carried out by a microcomputer program written in BASIC. The actual trade-offs which result will depend, of course, on the specific values of aerodynamic efficiency, mass, mechanical efficiency, etc. which characterize the baseline "typical" HPV. See the sidebar to this article for further details about the computer calculations and for exact definitions of terms like "average velocity" "aerodynamic drag," etc.

Table 1: Results of a typical computer run. For each specified value of Effective Frontal Area (AC_D), the computer adjusts the mass (of rider plus vehicle) so that average velocity remains the same as that (13.6 mi/hr) of the "baseline" design. (Each line in Table 2 is generated in this manner, using various values of rider power, slope, and stops per mile.) Results for the "conventional bike" (above) were calculated by assuming realistic values for the bike's effective frontal area and mass.

- **rider power (P_{in}) = 100 watts (0.13 hp)**
- **slope (s) = 1%**
- **stops/mile = 2**
- **rolling resistance coefficient = 0.050 Newtons/kg (0.005 lbf/lbm)**
- **transmission efficiency = 90%**

	Effective Frontal Area (C_DA)		Mass (m) Rider & Vehicle		Velocity (mi/hr)		
	m^2	ft^2	kg	lb	up	down	average
	0.05	0.54	114.3	252.0	10.3	19.8	13.6
	0.10	1.07	107.4	236.8	10.5	19.2	13.6
"baseline"----	0.15	1.61	100.0	220.5	10.7	18.6	13.6
	0.20	2.15	92.0	202.8	10.9	18.0	13.6
	0.25	2.69	83.6	184.3	11.1	17.4	13.6
	0.30	3.23	74.4	164.0	11.4	16.8	13.6
"conventional bike"---	0.39	4.20	88.6	195.3	10.2	15.7	12.3

Defining a "Typical HPV"

In this analysis I am interested mainly in the vehicle's mass (m) and its aerodynamic efficiency, which is expressed by its effective frontal area (actual frontal area A times drag coefficient C_D). Other parameters are fixed at what are hoped to be typical values. The efficiency of power transmission (i.e., chain, bearings, etc.) is set to 90 percent. Moulton reports three-speed efficiencies of 80 percent, 85 percent, and 90 percent in low gear, high gear, and direct drive, respectively (Ref. 1). The coefficient of rolling resistance is set to 0.05 Newtons/kg (0.005 pounds-force/pound-mass). This value is given by Whitt and Wilson for a bicycle with 27-inch wheels (Ref. 2, p. 123).

The trade-off is calculated by varying the initial values of mass and aerodynamic drag. For the initial value of mass, assuming the rider weighs 77.3 kg (170 lbs) and the vehicle weighs 22.7 kg (50 lbs, the Vector's approximate weight), the total mass is (our round-number goal now revealed) 100 kg (220 lbs).

Although there are few published data for area and drag coefficient for HPVs, Kyle lists values (Ref. 3) which yield an AC_D product ranging from 0.39 m^2 (4.20 ft^2, for a conventional bicycle) to 0.07 m^2 (0.75 ft^2, for a fully-faired upright bicycle) to 0.06 m^2 (0.65 ft^2, for a prone quadricycle with full fairing). Since practical HPVs might sacrifice some of the performance (and contortionistic requirements) of these single-purpose racing designs, I set AC_D equal to 0.15 m^2 (1.61 ft^2) as an initial value. This could be achieved by a vehicle with frontal area $A = 0.75$ m^2 (8.07 ft^2) and a drag coefficient $C_D = 0.20$.

"Typical" Riding Conditions

I specified riding conditions (i.e., rider

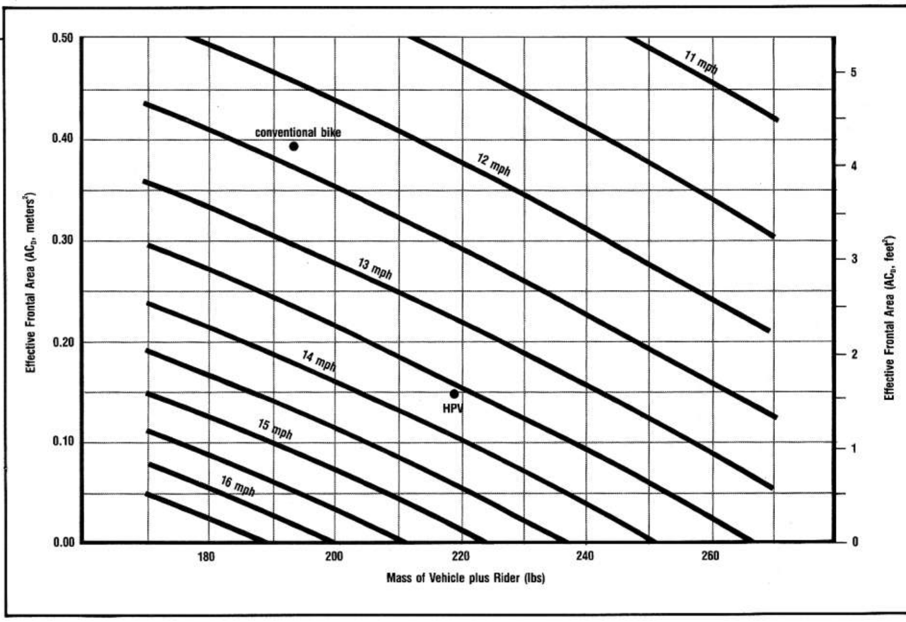


Figure 1a: Lines of constant average velocity with rider power = 100 watts (1 percent slope and 2 stops/mile).

George Retseck

Rider Power (P_{in}) watts	hp	Slope (s)	Stops per mile	Velocity (mi/hr) for "baseline" design: AC_D = 0.150 m², vehicle mass = 22.7 kg: up	down	average	Trade-off parameter ($\%\Delta AC_D$ / $\%\Delta$vehicle mass)
100	0.13	0%	1	16.1	16.1	16.1	0.35
100	0.13	0%	2	14.7	14.7	14.7	0.58
100	0.13	0%	4	12.8	12.8	12.8	1.10
100	0.13	1%	0	11.6	25.6	15.9	0.47
100	0.13	2%	0	7.8	32.7	12.6	1.23
100	0.13	5%	0	3.7	49.8	6.9	5.40
100	0.13	1%	2	10.7	18.6	13.6	0.98
100	0.13	2%	2	7.6	22.0	11.3	2.29
100	0.13	5%	2	3.7	30.6	6.6	10.81
200	0.27	0%	1	21.6	21.6	21.6	0.26
200	0.27	0%	2	19.5	19.5	19.5	0.49
200	0.27	0%	4	16.8	16.8	16.8	1.00
200	0.27	1%	0	18.7	30.5	23.2	0.20
200	0.27	2%	0	14.1	36.4	20.4	0.50
200	0.27	5%	0	7.3	51.7	12.8	2.39
200	0.27	1%	2	16.3	22.6	18.9	0.62
200	0.27	2%	2	13.2	25.4	17.3	1.05
200	0.27	5%	2	7.2	33.1	11.9	4.45
300	0.40	0%	1	25.3	25.3	25.3	0.23
300	0.40	0%	2	22.8	22.8	22.8	0.45
300	0.40	0%	4	19.6	19.6	19.6	0.93
300	0.40	1%	0	23.6	34.1	27.9	0.12
300	0.40	2%	0	19.1	39.3	25.7	0.29
300	0.40	5%	0	10.7	53.4	17.9	1.42
300	0.40	1%	2	20.0	25.5	22.4	0.52
300	0.40	2%	2	17.1	28.0	21.3	0.73
300	0.40	5%	2	10.5	34.7	16.1	2.70

Table 2: Tradeoffs between aerodynamic efficiency and vehicle mass while keeping average velocity constant. The tradeoff parameter tells you the percentage change in aero efficiency (AC_D) that produces the same effect as a 1% change in vehicle mass. Example: for a rider power level of 100 watts, on a closed course with 5% slopes and no stops, a 5.40% improvement in aerodynamic efficiency has the same effect (on average velocity) as a 1% reduction in vehicle mass. In general, when the tradeoff parameter is greater than 1.00, reductions in vehicle mass are relatively more important than reductions in air resistance. (All values based on assuming rolling resistance coefficient = 0.050 N/kg and transmission efficiency = 90%.)

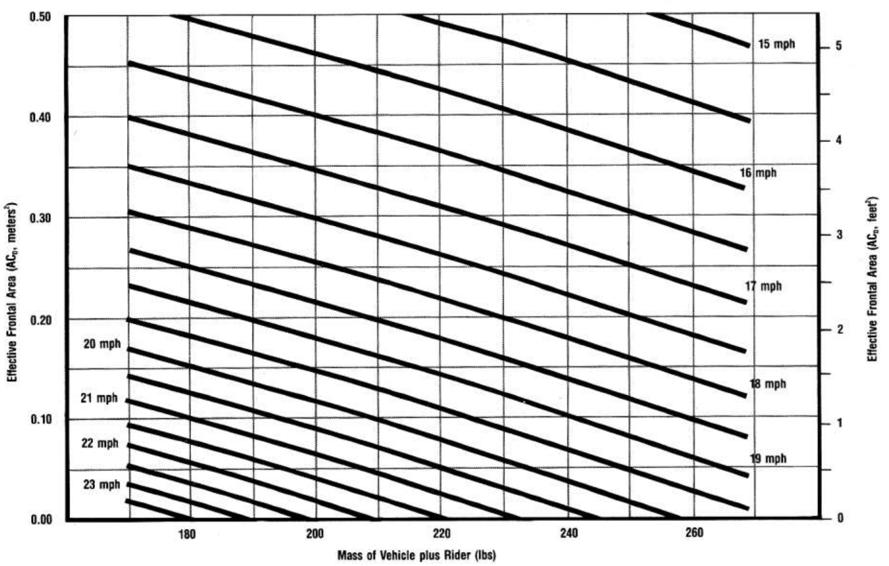
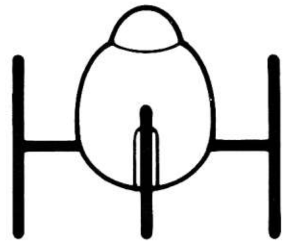


Figure 1b: Lines of constant average velocity with rider power = 200 watts (1 percent slope and 2 stops/mile).

George Retseck



Front View

Body:

$A = 5.4\ ft^2 = 0.5 m^2$

$C_D = 0.2$

Wheels:

$A = (2)(27\ in)(1\ in) + (16\ in)(1\ in)$

$= 70\ in^2 = 0.0452\ m^2$

$C_D = 0.9$

Axle:

$A = (24\ in)(1.5\ in)$

$= 36\ in^2 = 0.0232\ m^2$

$C_{D1} = 0.30$ **(for a cylinder)**

$C_{D2} = 0.06$ **(for an optimal airfoil)–See *Introduction to Fluid Mechanics* by R.W. Fox and A.T. McDonald, John Wiley, 1973, p. 412.**

Calculations:

$AC_{D1} = (.5)(.2) + (.0452)(.9) + (.0232)(.3)$

$= 0.1476\ m^2$

$AC_{D2} = (.5)(.2) + (.0452)(.9) + (.0232)(.06)$

$= 0.1421\ m^2$

Figure 2: Calculation of Effective Frontal Area (AC_D) for a Hypothetical HPV.

power output, frequency of stops, and slopes) which I felt were typical of a day trip or commuter run. For the rider's power output, I chose 100, 200, and 300 watts (0.13, 0.27, and 0.40 hp, respectively). Whitt and Wilson state that an average experienced rider (casual commuter?) can maintain 75 watts (0.1 hp) power output, and a well-trained racer can maintain 200 to 300 watts (0.27 to 0.40 hp) for several hours (Ref. 2, pages 38-43). For hills, I chose slopes of 1 percent, 2 percent, and 5 percent. What passes for level ground in the San Francisco Bay area, including, for example, the so-called "Berkeley Flatlands" near my home, actually has slopes of about 2 percent according to topographic maps. "Level ground" almost always has minor undulations with slope greater than 1 percent. The 5 percent figure represents a substantial but bearable slope. Finally, I chose to examine stops every 1/4 mile, 1/2 mile, 1 mile, and no stops.

Table 1 shows the results of the computer calculation for one set of riding conditions. The line marked "baseline" in Table 1 represents the initial values of aerodynamic drag ($AC_D = 0.15\ m^2 = 1.61\ ft^2$) and mass ($m = 100$ kg = 220 lbs). For these initial values, the table shows that the vehicle achieves an overall average speed of 13.6 mi/hr (10.7 mi/hr uphill and 18.6 mi/hr downhill). The other lines in the table show what happens as the AC_D parameter is varied above and below its baseline value while the overall average velocity (13.6 mi/hr) is constrained to remain the same. These lines were calculated by finding the mass which results in the desired average velocity with the specified AC_D value. Note that for a change in AC_D from 0.15 to either 0.10 or 0.20—a 33.3 percent change—the resulting change in mass (to 107.4 kg or 92.0 kg, respectively) averages 7.7 kg (16.9 lbs)—a 34 percent change in the vehicle's 22.7 kg (50 lb) mass. Thus, changes in vehicle mass are about 0.98 times as important as changes in aerodynamic drag, under the conditions stated in Table 1. Note also that this trade-off between mass and aerodynamic drag is roughly linear over the wide range of AC_D values listed in Table 1 (this linear relationship is also graphed in Figures 1a and 1b).

Table 2 summarizes, for a variety of riding conditions, the relationship between change in vehicle mass and change in aerodynamic efficiency (measured by AC_D) for a constant average velocity. For the baseline conditions ($AC_D = 0.15\ m^2$ and vehicle mass = 50 lbs) and for each combination of power, slope, and stops per mile, Table 2 gives the overall average velocity and, in the rightmost column, what I call the "trade-off parameter," which equals the percentage change in AC_D divided by the percentage change in vehicle mass with the average velocity constrained to remain constant. For example, at a power level of 100 watts, 1 percent slope, and 2 stops/mile, (the same conditions as in Table 1), the trade-off parameter of 0.98 indicates that changes in vehicle mass are 0.98 times as important as changes in aerodynamic efficiency.

Table 2 shows that at even modest rider power levels (100 watts) aerodynamic efficiency is more important than mass (i.e., trade-off parameter is less than 1) for small slopes (1 percent or less) and infrequent stops (2 or less per mile). But with more difficult riding conditions (steeper slopes and more frequent stops) mass becomes more important. At higher rider power levels (and thus at higher speeds), however, aerodynamic efficiency reasserts its importance.

Sample Design Problem

A designer of HPVs might make use of these results in the following way. His or her prospective human powered commuting vehicle might look something like the front view shown in Figure 2. The three-wheeler might have its wheels outside the aerodynamic shell for reasons of stability in cornering and crosswinds, and to keep water out. Is it worthwhile to streamline the axle, given that a fairing for the axle increases weight? Adding an axle fairing would decrease the vehicle's AC_D by 3.7 percent (from $0.1476\ m^2$ to $0.1421\ m^2$), by calculations shown in Figure 2. The designer now selects a trade-off parameter of 1.0 which represents (as shown in Table 2) a typical commuting situation. Thus, the vehicle's mass could be increased by 3.7 percent, or 1.9 lbs for a 50 lb vehicle. Surely an axle fairing weighing less than 1.9 lbs can be built, and this analysis shows that it is certainly worthwhile to

Calculating Average Velocities By Computer

Average velocity over equally long uphill and downhill stretches is derived as follows: The time T to go a total distance D is:

(1) $T = D/2(V_u+V_d)$

where V_u and V_d are the average velocities uphill and downhill respectively. Average velocity over the total course (uphill and downhill) is given by $V=D/T$, and we can substitute T from equation (1) into this expression to obtain:

(2) $V = 2 \cdot V_u \cdot V_d/(V_u+V_d)$

The uphill and downhill velocities are the result of net power input and acceleration between stop signs. Net power is the power available to accelerate the vehicle after deducting power consumed by mechanical losses, aerodynamic drag, rolling resistance, and changes in potential energy (i.e., hill-climbing). Net power input is given by:

(3) $$P_{net} = \eta \cdot P_{in} - q \cdot C_D \cdot A \cdot V^3/2 - m \cdot C_R \cdot V - m \cdot g \cdot s \cdot V$$

where P_{in} = power input by rider (watts)
η = mechanical efficiency of drive train
q = density of air (1.293 kg/m^3)
C_D = coefficient of aerodynamic drag
A = frontal area of vehicle (m^2)
V = velocity (m/s), either uphill or downhill
m = mass of vehicle plus rider (kg)
C_R = coefficient of rolling resistance (Newtons/kg)
g = acceleration due to gravity (9.8 m/s^2)
s = slope (positive for uphill, negative for downhill)

The first term on the right of equation (3) is the power level of the rider adjusted for losses in the chain, bearings, etc. The second term is the power loss due to air resistance. The third term is power lost to rolling resistance. Finally, the last term represents the power consumed in moving the vehicle uphill or gained in moving the vehicle downhill.

Net power acts to accelerate the vehicle. Acceleration as a function of net power and velocity can be derived by differentiating the equation for kinetic energy:

(4) $E = m \cdot V^2/2$

(5) $P_{net} = dE/dt = mV(dV/dt) = mVa$

where $a = dV/dt$ is acceleration.[1] Thus, acceleration is given by:

(6) $a = P_{net}/mV$

The computer simulation uses numerical integration to calculate the time to travel distance D/2 starting from a stop sign. This integration is performed for both uphill and downhill segments.[2]

The heart of the computer program is an algorithm which calculates the velocity and position of the HPV step-by-step through time, starting from a dead stop. At each step, the net power (P_{net}, which depends on the velocity) is calculated using equation (3), and then the acceleration calculated by equation (6) is used to determine the velocity for the next step. A numerical integration scheme is used to estimate velocity steps in a Taylor series expansion with two higher order terms. The accuracy of this program was verified by comparing its results with the analytic integral which can be solved exactly when air drag and rolling resistance are set to zero in equation (3).

The program includes facilities for repeating these calculations for uphill and downhill segments for specified values of the relevant variables (total mass, drag coefficient, rider power, etc.) under the interactive control of the user.[3] Given a value for the air drag term and a target average velocity, the program uses a trial and error procedure to find the vehicle mass that will meet the target average velocity. (That is, if velocity is too low, reduce mass and try again.)

[1]This derivation neglects the power used in increasing the rotational kinetic energy in wheels and cranks. If this were included, the form of equation (5) would remain the same but the term "m" in (5) would then represent an "effective inertial mass" which would be several percent larger than the actual gravitational mass of the vehicle.

[2]There are a few approximations in this analysis. First, I assumed that no time is spent in slowing down to a stop. This approximation results in a slight overestimate of average velocities and thus a slight overestimate of the relative importance of aerodynamic efficiency compared to mass. The analysis also assumes that there are stop signs at the top and bottom of each uphill and downhill segment. Thus momentum is not carried over from one segment to the next. Finally, assuming constant power input from the rider causes a numerical difficulty because acceleration becomes infinite in equation (6) when velocity is zero. This was overcome by using, for the first half-second of acceleration, the analytical expression which ignores power lost to air drag, rolling resistance, etc.

[3]The program is written in MicroSoft BASIC and should run on most microcomputers with minimal modification. Readers may obtain a printed copy of the program by sending a stamped self-addressed envelope to the author at: 1819 Francisco Street (Rear), Berkeley, CA 94703.

add it. In fact, only under very severe riding conditions (slopes of 5 percent) does the axle fairing become a doubtful proposition.

Final Observations

Here are some numbers that should provoke thought among HPV enthusiasts. I sent a conventional upright, unfaired bicycle through the calculation, under the same riding conditions that applied to the HPV in Table 1 (100 watts rider power, 1 percent slope, 2 stops/mile, etc.). For the conventional bike, I assumed a vehicle mass of 11.4 kg (25 lbs) with a 170 lb rider (same as for the HPV), and an AC_D of 0.390 m^2 (4.20 ft^2), based on Kyle's data. The result, plotted in Figure 2, shows that the conventional bike averages only 12.3 mi/hr, which is more than 10 percent slower than the "baseline" HPV's 13.6 mi/hr. Note that the conventional bike is 50 percent lighter than the HPV, but it has 160 percent greater air resistance. Thus the bike's penalty in air drag is far greater than its benefits in lighter weight and, in these conditions, the bike would be a poor choice compared to the "typical" HPV.

One final point: in Figures 1a and 1b, the lines of constant average velocity are very nearly straight and parallel. This indicates that the trade-off parameters listed in Table 2 are applicable over a fairly wide range of vehicle mass and aerodynamic efficiency (AC_D).

Practical HPVs could become popular very quickly. By all indications, modestly trained riders could average nearly 20 mi/hr under typical riding conditions including stops and hills. These speeds should be attractive to bicycle commuters everywhere. But HPV designers have not yet developed a really attractive and practical package. I'd hope to see more compact vehicles in the future. After all, I would hate to include in the calculations of average velocity the time spent walking from the workplace to the HPV's parking space.

1. *"Human Powered Bicycle Considerations," by Alex Moulton, in* The First Human Powered Vehicle Scientific Symposium Proceedings, *A.V. Abbot, ed., IHPVA, 1982, p. 81.*
2. Bicycling Science *(second edition), by F.R. Whitt and D.G. Wilson, The MIT Press, 1982.*
3. *"Predicting Human Powered Vehicle Performance Using Ergometry and Aerodynamic Drag Measurements," by Chester R. Kyle, in* Proceedings of the International Conference on Human Powered Transportation, *San Diego, 1979. Data are reproduced in Bicycling, May 1982, p. 62.*

This article was adapted with permission from the Second International Human Powered Vehicle Scientific Symposium Proceedings, *published by the IHPVA, PO Box 2068, Seal Beach, CA 90740.*

Still Ferdinand

I was delighted to find Claude Genzling's article reprinted in your August issue. Having read the March 1984 "Le Cycle" (no. 99), I would like to bring to the attention of your readers another paper on the same subject. Dr. Jean-Pierre de Mondenard signs this second article entitled: "World Record and Altitude; the Essential Scientific Preparation." The author discusses the pros and cons of choosing Mexico in an attempt to break the record. He points out that, if the lower air density gives the cyclist an advantage of up to 15 percent, the efficiency of the athlete diminishes by 7 percent because of the breathing problems occurring in altitude. This way, Moser would have benefited from an increase of his speed nearing 7 percent. Applying this value to Ferdinand Bracke's old record, Mondenard found that the true hour record holder is . . . Bracke again.

Yves Robert
Poliquin-La Cordée Vélo
Montreal, Quebec, Canada.

Aerodynamic Calculations

I read both aerodynamic articles in the August 1984 *Bike Tech.* The approximations made in "The True Hour Record Holder . . . Is Bracke!" by Claude Genzling are fairly extreme but seem okay. It's Glen Brown's article, "Spoke Drag," that bothers me.

1. The integral (on page five) bothered me for a long time. I calculated it all out and arrived at $\overline{v^2} = {}^2/_3\, V^2$ instead of $\overline{v^2} = {}^5/_6\, V^2$ as was reported. The drag isn't $^5/_6$, but $^2/_3$ of the drag of the same wheel held broadside to the wind of the same speed.

2. How did Brown arrive at 3.2 square-feet for a mounted, fully-crouched rider's drag area? The most streamlined combination in the preceding Genzling article is 3.23 square-feet. Oversimplification to prove a point?

Christopher R. Cleary
Reading, Massachusetts

Glen Brown replies:

I'm happy to clear up the points that you raised.

1. Some of the confusion arises from the fact that several intermediate equations from the original manuscript were omitted from the final printed version because of lack of space. This made the mathematics a bit obscure and my comments on the separability of the translational and rotational effects hard to follow.

Here are some intermediate equations that should help:

$$v^2 = V^2\,[(r/R)^2 + \cos^2\Theta + 2\cos\Theta\; r/R]$$

. . . this becomes the integrand in . . .

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

. . . giving the result . . .

$$\overline{v^2} = ({}^1/_3 + {}^1/_2)\,V^2 = {}^5/_6\,V^2$$

The result is $^5/_6\,V^2$ as published, being the sum of $^1/_3\,V^2$ (translation) and $^1/_2\,V^2$ (rotation). Note that I incorrectly reversed the proportion in the article.

Note also the typographical error in the original article showing the bar denoting an average appearing under the exponent. They should be reversed, as correctly shown above. It is the average value of the square, not the square of the average.

2. The use of 3.2 instead of 3.23 is not an oversimplification and was quite intentional. The way that a number is written implies its accuracy along with its value. The drag value of a rider on a bicycle isn't significant to one percent, especially if his or her identity, clothing, and position on the bicycle aren't specified exactly.

The point of the article is to provide a theoretical justification for the assertion that spoke drag is a highly significant portion of the total drag of a bicycle (one that I found hard to grasp intuitively). Once Chester Kyle publishes his wind tunnel results, including both drag and torque measurements, those values should be used.

3. I'd also like to comment on Genzling's article. His conclusion in terms of athletic performance is, of course, correct. However, those who believe that cycling should remain a purely athletic sport should all ride high wheelers. There is only one hour-record in my mind, and it belongs to Fred Markham in a streamlined Easy Racer recumbent, set September 29, 1984 in Indianapolis. His distance was just over 60 kilometers on a very rough $^5/_8$-mile oval in the wind!

Glen Brown
Zzip Designs
Santa Cruz, California

It's Section Modulus that Counts

I disagree with Doug Roosa's explanation in the April *Bike Tech* article on rim rigidity and strength. He says that the Mavic Model 4 and the Rigida 1320 rims have equal rigidities but unequal strengths because "the Rigida has less material to bear the load, so each bit of material is under a higher level of stress." The Rigida is weaker because it has material at a greater distance from the neutral axis than the Mavic. This material, being more highly stressed than any material in the Mavic rim for the same applied load, may exceed its yield point and cause permanent deformation, while the material in the Mavic rim, lying closer to the neutral axis, remains within its elastic limit.

This can be seen by noting that the bending moment M_s at the onset of permanent deformation is given by:

$$M_s = s_y\,(I/Y)$$

where s_y denotes the material's yield stress, Y is the distance of the farthest element from the neutral axis, and I is the moment of inertia for the rim section. The quantity I/Y is commonly known as the section modulus.

If the rigidity R is defined by the relationship $R = EI$, then the rigidity-to-strength ratio becomes:

$$R/M_s = EY/s_y.$$

Note that the above expression is independent of cross-sectional area. If the rims have approximately the same elastic moduli and yield points, then the rigidity-to-strength ratios are proportional to Y alone.

Using the data supplied in *Bike Tech,* we have for the Rigida rim:

$$(R/M_s)_{Rigida} = 92/43 = 2.14,$$

and for the Mavic rim:

$$(R/M_s)_{Mavic} = 92/61 = 1.51.$$

Consequently, I would suspect that the corresponding Y-values are roughly in the ratio of 2.14 : 1.51.

To sum up, the Rigida and Mavic rims are equally rigid because their moments of inertia are similar; they are unequal in strength because their section moduli differ. This is the only possible explanation if variations in yield strength and elastic modulus are excluded. Differences in cross-sectional area are immaterial.

Raymond Pipkin
Western Springs, Illinois

HPV SPECIAL SECTION

Easy Talker Interview with Gardner Martin

Jim Redcay

Editor's note: The names Easy Racer, Tour Easy, and Gardner Martin will be familiar to those readers who attended the Tenth Annual Human-Powered Speed Championships in Indianapolis this past September. The Easy Racer, with rider Fred Markham, set two new IHPVA World Records (the 4000-meter pursuit and one-hour time trial), and other Martin-designed vehicles also took several high awards. The Tour Easy, a lower-priced, higher-production successor to the Easy Racer, won the IHPVA's Most Practical Vehicle Award in 1983. As one of the few designers of HPV's and recumbents whose products have been successful in the commercial marketplace, Gardner Martin has a unique point of view. In this interview, he explores some technical and practical factors that will influence HPV design in the near future.

Bike Tech: What research have you been doing on partial fairings or streamlining for commuters' protection from the elements? Most of what we've seen in the past seems impractical.

Martin: The practical approach to streamlining for the street is basically a two-fold system. It's a quickly removable windshield, a streamlined windshield such as the standard Zzipper, or the new Super-Zzipper, which is about three times the size of the standard. Then by adding a removable, stretchy body cover that zips onto the big windshield and entirely covers the rider, substantial aerodynamic benefits can be achieved.

For the street, the streamlining should be easily removable if the bike's to be ridden in heavy winds or heavy traffic with trucks or buses — in some situations any large surface area can be highly affected by side winds. You can learn to ride in the wind, and do what sailors call "tacking into the wind," but in a lot of situations, full streamlining is not safe. The large fiberglass bodies that we use on the Easy Racers literally become airborne under adverse wind conditions.

Bike Tech: So you're saying that elastic cloth as opposed to hard laminates would be the way that streamlining will go for bikes, if at all?

Martin: We think that that's the way it will probably go for bicycles in the immediate future, because it's so simple, so light and does significantly reduce wind resistance.

Bike Tech: What are we talking about in miles per hour?

Martin: If a small Zzipper windshield adds one and a half mph to the top speed on level ground, the large Zzipper windshield adds two and a half mph. The large Zzipper windshield with a body stocking will add another two or two and a half mph above the windshield alone. We're talking $4^{1/2}$ to 5 mph hour for the complete setup, maybe even a little more downhill.

Bike Tech: How about the cooling . . .?

Martin: The cloth body stocking probably starts to become a little hot for the rider, with temperatures above 65. There are things we're going to try with different ventilation points. We've found that without good ventilation your hands get awfully hot and sweaty, and control could become a problem. Although I haven't investigated what percentage of heat is lost through the hands, I think that could be one of the more important places to ventilate the rider.

Bike Tech: Have your HPV's been ventilated?

Martin: We haven't done any ventilation yet on our experimental body stockings. Don Licht, in Ohio, is working with his own version of a cloth body stocking made out of silver reflective nylon. It completely covers the windshield so sunlight cannot get through. (The rider's field of vision is above the windshield.) He says the big windshield acts like the greenhouse effect. His bike has three separate panels of the cloth. One goes over the head and shoulde[s, and just that will keep the sun off of you. And he says that it's cooler to ride with the cloth than without it on hot sunny days. So maybe he's onto something.

Bike Tech: Are any major manufacturers interested in taking advantage of the pioneering work that's been done on recumbents?

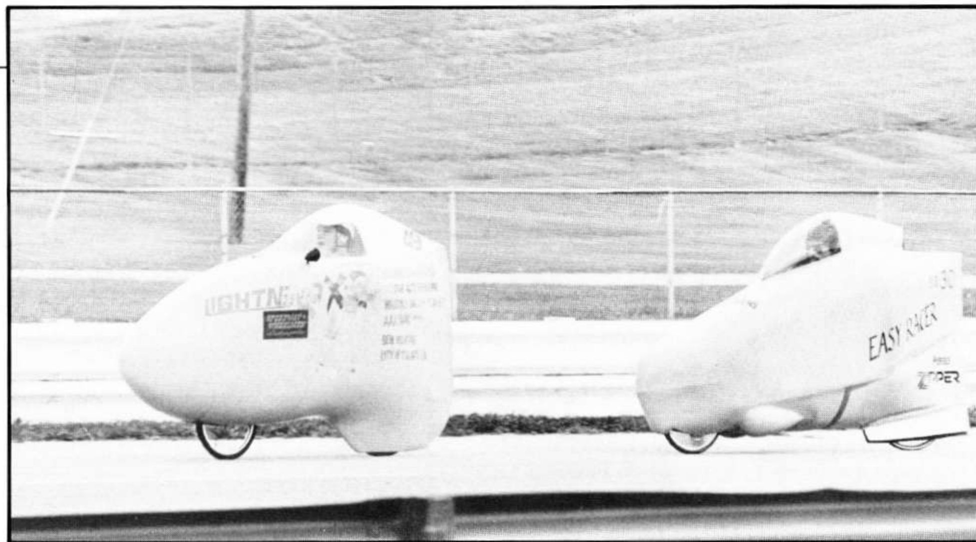
Martin: The manufacturers are interested. The boom in mountain bikes was very easy for all of the manufacturers to jump into. Because they are so similar to the conventional diamond frame road bike, any manufacturer could tool up in a week to build a mountain bike. But the manufacturers are unsure how big the market may be for recumbents and they know less whose recumbent to copy. So, until they know, I may have some breathing room.

Bike Tech: You said you won't sell to bike manufacturers; have you tried negotiating a joint venture or licensing agreement? Is there anything to license?

Martin: Certainly, our name. I'll tell you, almost all of the current recumbents on the market have features that were patented 75 or 80 years ago. The patents are all expired, and the Patent Office considers none of the recumbents patentable, even though they might combine many of the features into a different overall package. That doesn't mean

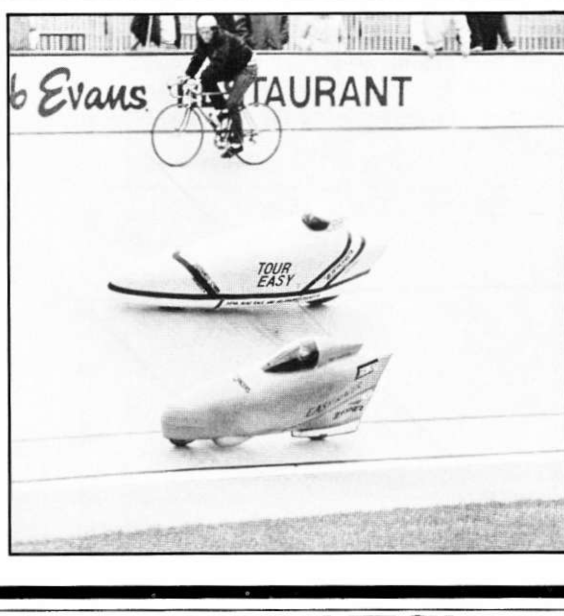
Photos by Michael Chritton

Gardner Martin, at right, helps rider Greg Miller to get the Easy Racer I **rolling. The** Tour Easy **name appearing on the vehicle is not its name, but an advertisement for Martin's mass-produced recumbent bike. (All photos accompanying this interview were taken at the 10th Interntional HPV Speed Championships, September 1984.)**



Two views of rider Fred Markham in the Easy Racer II: **in hot pursuit of** Lightning X-2, **and moments after setting a new world record in the One Hour Time Trial.**

The "street legal" version: this is the standard production model Tour Easy, **shown here with the small** Zzipper **fairing snapped to the handlebars. The owner has added his own fairings on the wheels and rear fender.**



The two Easy **vehicles: I (background) and II (foreground). The changes in design from I to II are significant: shorter wheelbase, more compact body, larger windshield for visibility, and greater ground clearance.**

that somebody couldn't invent something that is patentable. But so far the ones that work best have been tinkered with for the past 100 years. Until just five years ago nobody really built a recumbent to sell that had the bugs worked out of it well enough to be a viable alternative.

Bike Tech: Do safety and comfort concern a much greater segment of the population than most cyclists realize?

Martin: Yes, I really believe so. Our first 100 buyers received questionnaires in which we asked them, "Which is most important to you? Comfort, safety, or speed?" and I was surprised that most of our buyers bought the bikes for comfort and safety.

Knowing you can put your feet down on the ground at a moment's notice makes the bike, perhaps, more user-friendly to some folks than conventional bicycles are. It takes a little bit of learning, but in five minutes in a flat parking lot, anybody can learn to ride a recumbent. They'll be smiling and saying, "Wow, this is neat!" Now there are a few people that it's not going to be better for. The trained racer that knows how to get the maximum out of his bike in both braking and performance — well, our bike is maybe not his cup of tea.

Bike Tech: Do you feel your bike is more stable in high-speed situations, like descending hills, than a conventional bike?

Martin: There is certainly less tendency with the long wheelbase recumbent to be as upset by bumps into a speed wobble, or to be knocked off the course. The ultimate limit of whether you skid off the road or not is probably going to be as close with a good recumbent bicycle as with a good upright bicycle.

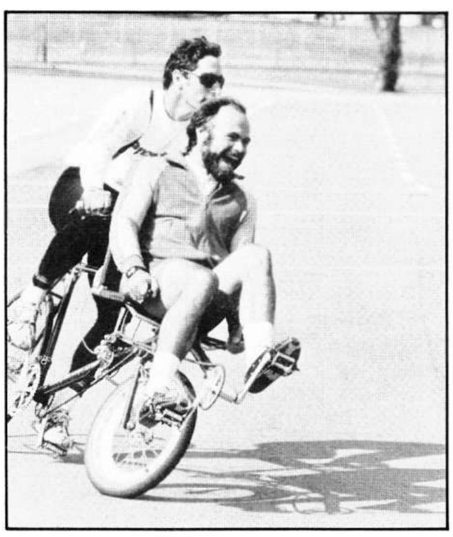
Bike Tech: One of our more accomplished test riders, who has been commuting on the Tour Easy for more than three months, claims that although he "loves" it and he's now used to the handling differences between your recumbent and an upright bike, he finds a certain quickness in the Tour Easy's steering that requires more concentration for straight line riding. Don't you

HPV SPECIAL SECTION

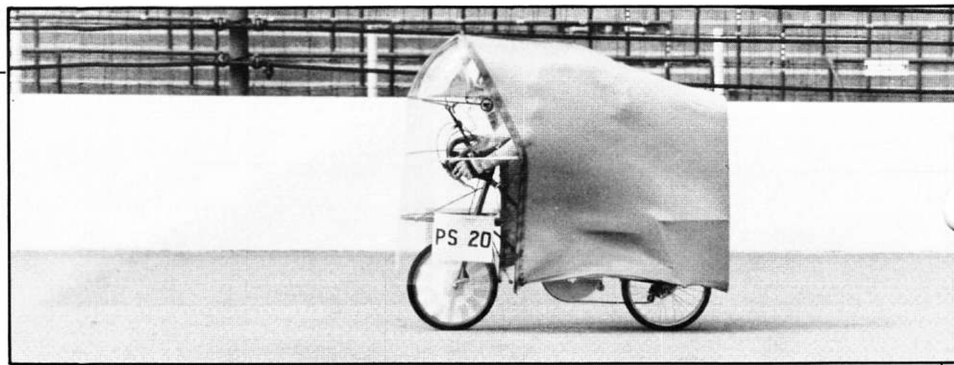
Another Record Year

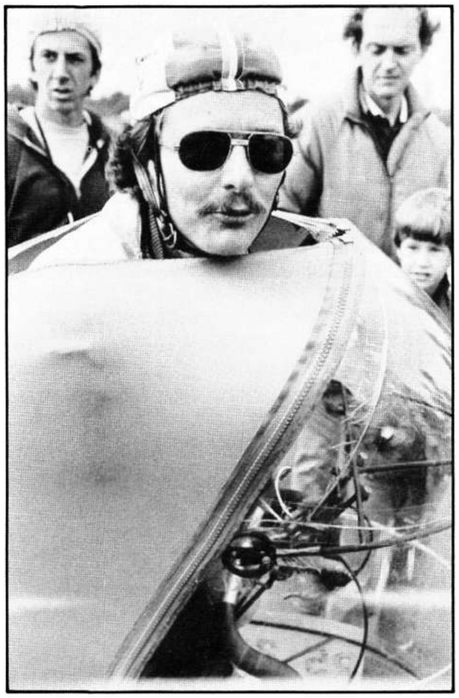
Robert G. Flower

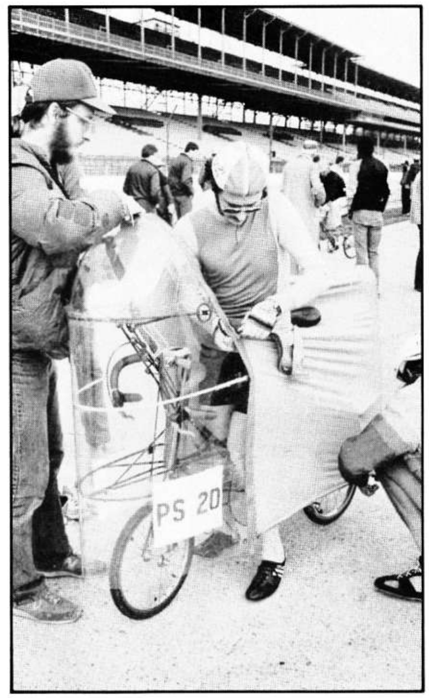
"It's only a matter of time before local HPV races become as common as USCF-sanctioned bike races." That was one of the prevailing opinions among HPV enthusiasts at the 1984 International Human Powered Vehicle (IHPVA) Speed Championships, held in Indianapolis this past September. It might be a few years before that prediction comes true, but there certainly is a lot of momentum in that direction. Three new world speed records were set this year, and a record number of entrants (84, up from 62 last year) participated. There were 8 separate speed contests and one competition for practical "commuter" vehicles; top finishers are listed on page 14 of this issue. And thanks to the tireless efforts of volunteers from the IHPVA Indy Chapter, plus the world-class racing facilities at Indianapolis, the three-day affair ran smoothly. The event was definitely international: there were 5 Canadian entrants and 5 from Great Britain. And there is a good chance that the 1986 HPV Speed Championships will be held at the International Transportation EXPO in Vancouver, British Columbia, to the accompaniment of worldwide publicity and interest. Technical highlights of this year's competition are shown here and on the next few pages of Bike Tech.



Photos by Michael Chritton

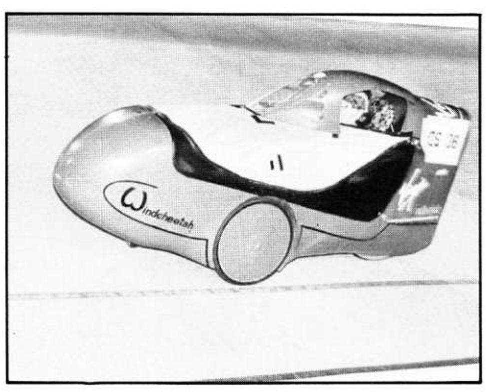


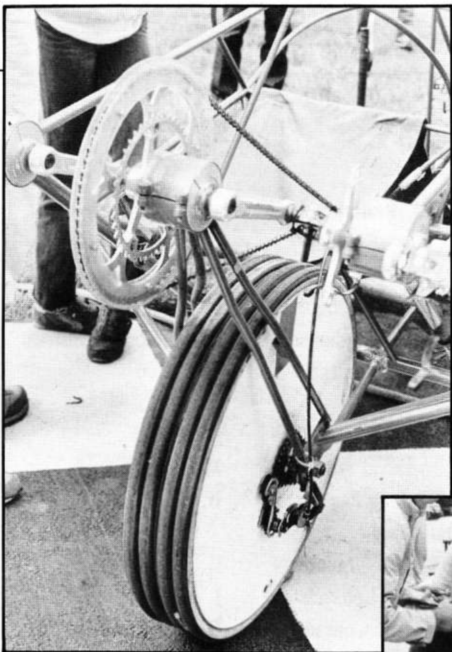


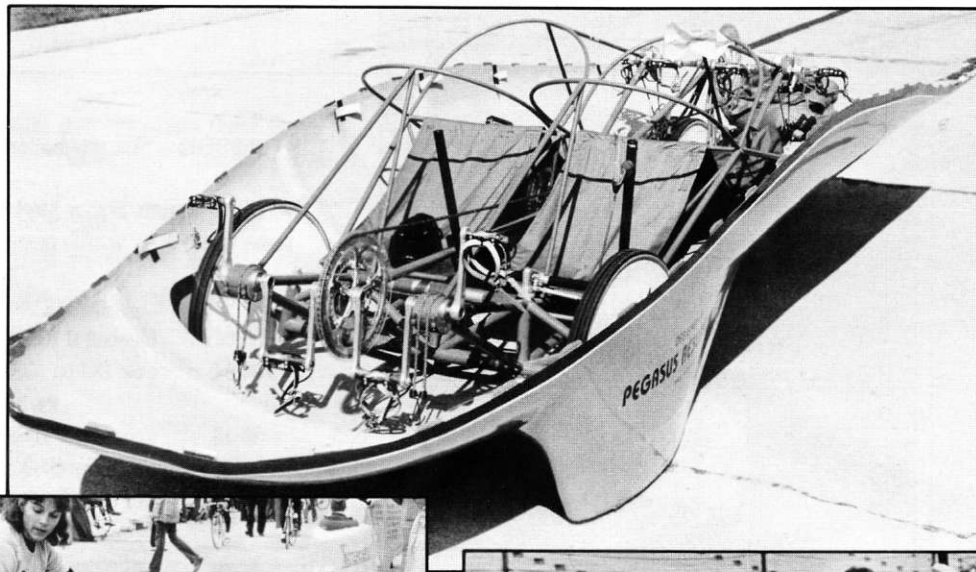


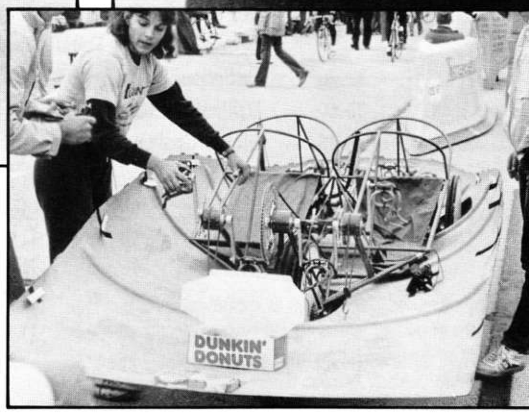
Hybrid Vigor: The Aero-Moulton **shown here uses an elastic fabric ("Spandex") fairing fastened by zipper to a clear Lexan plastic windshield. Hybrid fairings (part-rigid, part-flexible) are starting to be seen as solutions to some of the difficulties inherent in all-rigid designs. At 29 lbs., the** Aero-Moulton **is among the lightest one-rider HPV's with almost full fairing coverage.**

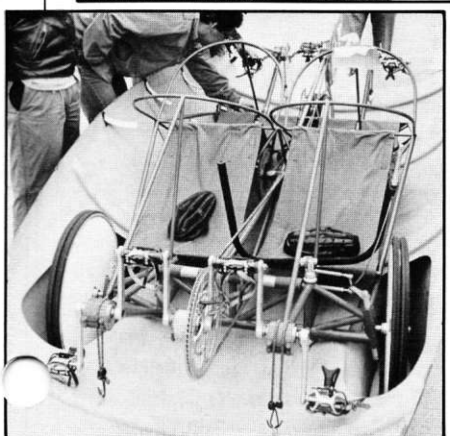
Winner of the "Commuter" Vehicle Competition: Windcheetah, **designed by Mike Burrows and ridden by Andy Pegg, both of Norwich, England. The main frame is 2 inch by 1/8 inch steel tubing, assembled by gluing in cast alloy lugs. The main fairing is fiberglass, with a stretch fabric top and removable wind screen.**





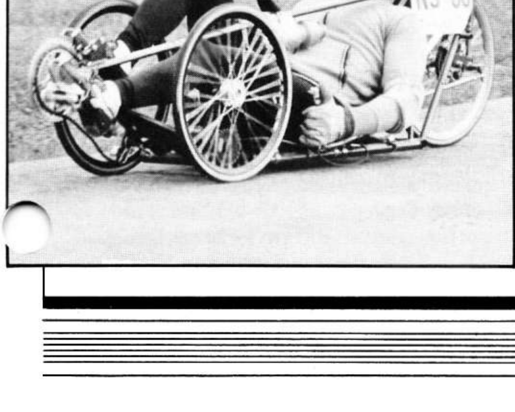


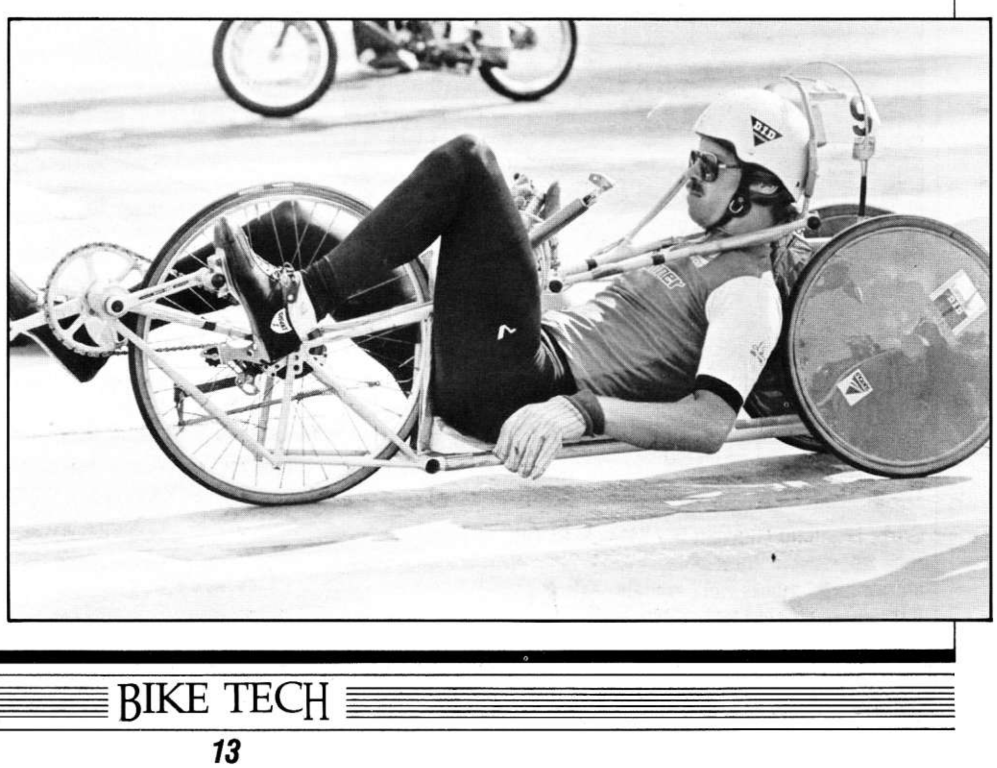




The four-rider *Fusion* vehicle seen here, designed by Leisha Peterson and Kelly Londry of Pegasus Research, demonstrates a number of novel and well-integrated design ideas. A descendent of the four-wheeled *Pegasus* vehicle that competed last year, *Fusion* is three-wheeled, with two front (unpowered) steering wheels plus one rear driving wheel. A total of eight tires (two per rider), obtained by assembling conventional bicycle wheels *in tandem*, are used; thus the rear "wheel" actually has four tires (see photo). The frame is now a light (36 lb.) "spaceframe" structure made of many small diameter steel tubes. The body shell was made by conventional automotive fiberglass techniques, and is based on a design from the Pininfarina Research Labs (known for their Ferrari automotive designs). Light fabric seats are suspended directly from the frame tubing.

Spaceframes: One of the designer's biggest challenges is to create an efficient structural frame. Here *Biotec Challenger* (NS9, two rear wheels) and *Biotec Vision* (NS66, one rear wheel) show two variations by one design team (Eric Conrad and Gil Linde) on the theme of a well-triangulated 3-dimensional spaceframe. Both designs include a main frame tube which passes over the rider; on *Biotec Challenger* this tube also carries the steering control levers. Both vehicles are steered from the rear wheels and are powered from the front.





Full-body fairings present a classic design dilemma: how to open the shell for the rider to enter/exit the vehicle. Shown here: the "landing gear" doors on Lightning X-2, which open so the rider can put down his feet when stopping, and the side door on Dust Devil with rider Tom Cochran emerging.

Cross-breeding between the BMX, ATB, and HPV species is inevitable. Power Mills I, shown here, sports BMX wheels, ATB gearing, and is steered by leaning.

TOP FINISHERS—10th International Human Powered Speed Championships

200 Meter Sprints Flying Start (9/28/84, Indianapolis Motor Speedway)

Place	Speed (mph)	Vehicle (number)	Rider	Designer/Builder
		Open Class		
*1	57.39	Lightning X2 (OS49)	Carl Sundquist	T. Brummer
2	55.04	Bluebell II (OS56)	Doug Adamson	D. Henden/S. Mettam
3	54.94	Cole Dalton (OS10)	Cole Dalton	C. Dalton
		Partially Faired Class		
1	35.13	Econogator (PS97)	Stu Krebs	J. Lebsack
2	34.36	No Name (PS92)	David Wilson	M. Bannan
3	34.09	DeFelice (PS78)	Tony Peyton	B. DeFelice
		Non-Faired Class		
1	37.20	DeFelice (NS79)	Dan Griesmer	B. DeFelice
2	35.50	Navigator (NS44)	Jon Lebsack	J. Lebsack
3	35.43	Biotec Challenger (NS09)	Eric Conrad	E. Conrad/G. Linde G. Mosser/B. Boston
		Multiple Rider Class		
1	53.90	Fusion (OM99)	J. Gross, K. Nowakowsi, H. Peterson, D. Stanley	R. K. Londru L. Peterson
		(OM40)		
2	40.03	Counterpoint Opus		
		(NM40)		
3	39.24	Counterpoint Opus	Tom McDonald,	J. Weaver Ken Yu

4000 Meter Individual Pursuits (9/28/84, Major Taylor Velodrome)

Place	Time (m:s)	Approximate Speed (mph)	Vehicle (#)	Rider	Designer/ Builder
*1	3:43.79	40.09	Easy Racer (OS30)	Fred Markham	G. Martin
2	4:01.07	37.10	Lightning X-2 (OS49)	Carl Sundquist	T. Brummer
3	4:07.52	36.15	Easy Racer (OS31)	Greg Miller	G. Martin

20 K Lemans Start Road Race (9/29/84, Indianapolis Raceway Park)

Place	Time (m:s)	Vehicle (#)	Rider	Designer/Builder
1	26:23	Lightning X-2 (OS49)	Carl Sundquist	T. Brumm
2	26:32	Moby/Infinity 2C (OS51)	Murray Wilmerding	T. Hreno
3	29:03	Easy Racer (OS30)	Fred Markham	G. Martin

34K Paced Start Road Race - 12 laps, 20 miles (9/29/84, Indianapolis Raceway Park)

Place	Time (m:s)	Vehicle (#)	Rider	Designer/Builder
1	36:34	Easy Racer (OS30)	Fred Markham	G. Martin
2	38:44	Lightning X-2 (OS49)	Carl Sundquist	T. Brummer
3	38:59	Easy Racer (OS31)	Greg Miller	G. Martin

One Hour Time Trial - Standing Start (9/29/84, Indianapolis Raceway Park)

Place	Distance (miles)	Vehicle (#)	Rider	Designer/Builder
*1	37.50	Easy Racer (OS30)	Fred Markham	G. Martin
2	35.73	Easy Racer (OS31)	Greg Miller	G. Martin
3	29.62	Bluebell II (OS56)	Doug Adamson	D. Henden/S. Mettam

8K Lemans Start Road Race - Approx. 12 Miles (9/30/84, Eagle Creek Park)

Place	Time (m:s)	Vehicle (#)	Rider	Designer/Builder
1	13:29	Lightning X2 (OS49)	Carl Sundquist	T. Brummer
2	13:45	Moby/Infinity 2C (OS51)	Murray Wilmerding	T. Hreno
3	14:15	Windcheetah (OS36)	Andy Pegg	M. Burrows

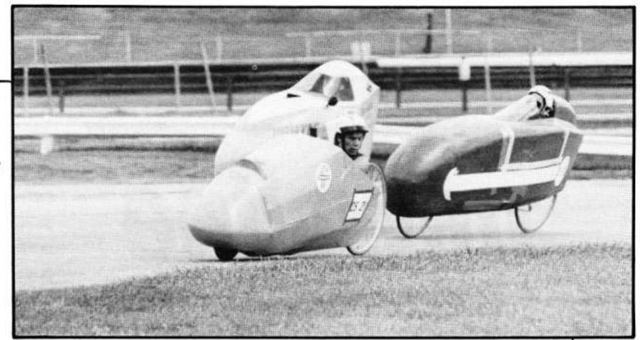
*denotes new World Record

32K Paced Start Road Race - Approx. 20 Miles (9/30/84, Eagle Creek Park)

Place	Time (m:s)	Vehicle (#)	Rider	Designer/Builder
1	40:39	Easy Racer (OS30)	Fred Markham	G. Martin
2	41:55	Lightning X-2 (OS49)	Carl Sundquist	T. Brummer
3	42:55	Moby/Infinity 2C (OS51)	Murray Wilmerding	T. Hreno

Commuter Vehicle Road Rally (9/30/84, Eagle Creek Park)

Place	Vehicle (#), Country	Rider	Designer/Builder
1	Windcheetah (OS36), GB	Andy Pegg	M. Burrows
2	Camp Carrier (OS22), US	Stephen Delaire	S. Delaire
3	Hon Folder (NS57), US	Adam Englund	H. Hon
4	Joyrider (PS17), CAN	Morgan Lemen	D. Messenger/L. Robert
5	Moulton Aero (PS20), US	Jim Glover	A. Moulton/D. Milliken/ P. Milliken
6	Sneaker (OS95), US	Dan Valatka	D. Valatka



Close quarters in this turn highlight a tough design problem: the need for better control at high speeds. Designers this year generally agreed that "good handling" is almost as important as good aerodynamics in a winning HPV. Seven vehicles in the 32K road race crashed in this turn; contributing factors were steering instabilities, low ground clearance, and road roughness. Fortunately, fairings protected the riders from serious injuries.

Gardner Martin Interview

Continued from page 11

think the five-minute learning period is a little brief?

Martin: Nobody can be fully adapted, certainly, in even five hours. To get all the nuances might even take six months.

Bike Tech: Do you think it's easier for people who have trouble with the standard upright position or who are not as used to that position . . . ?

Martin: Or have never ridden a bicycle before. An example: my wife's best friend from college wants to get into some kind of exercise. She has never ridden a bicycle, although about seven years ago, my wife and I tried to teach her how to ride a regular bicycle. Well, she fell about three times, hard, and gave up. A week ago, I took her out again on a recumbent, and inside of 15 minutes I had her riding the recumbent by herself.

Bike Tech: If somebody wanted to try racing on a recumbent, would they handle the same?

Martin: Oh, yes. Our racer, Greg Miller, from the Los Angeles area, can ride our bike in most situations with any kind of racing bicycle.

Bike Tech: Does that long frame necessarily mean that the bicycle is going to feel whippy or like you're losing power when you're pedaling it?

Martin: Some recumbents do feel whippy. Ours is very well triangulated and has no whippiness whatsoever, so far as side-to-side motion is concerned. There is some flex in up-and-down motion because your weight is concentrated between the long wheelbase, and as you go over the bumps, you can get a little springiness in the frame.

But the frame is very well braced against any bottom bracket twisting because most of the force on a recumbent is going in the direction of the major frame tubes. And on a recumbent you generally sit more still than you would on a regular bike; by not actually shifting your weight so far from one side to another, there'll probably be much less flex.

Bike Tech: Do you think major innovators in the market place are somewhat penalized by the buying public? That if parents go to buy a bike for little Johnny they want a bike that looks just like the one they had?

Martin: Could be. It certainly could be overcome. Look at the Marx BigWheel, the first successful recumbent sold to the masses. (It) has saved probably millions of dollars in dental bills for parents all over the country. The Marx BigWheel might be the most successful children's toy in the past 20 years. It's much safer for kids in most respects.

Bike Tech: Do you think that the general enthusiasm of the public for genuine improvement would carry the day?

Martin: Oh, I think it would. I think the serious cyclist market is perhaps the most resistant to change in that direction. They have so much ego as well as money invested in what they have perceived as the best possible bicycle they could get.

But I would say perhaps as much as 50 percent of our buyers are not serious cyclists. They may be serious engineers, serious about their exercise, but they're not serious about the brand name of their derailleur. They just want it to function. This is the kind of thing that recumbent bicycles and the Human Powered Vehicle Association are stimulating more than any other thing has stimulated bicycling in the past 100 years. So we are going to see a revolution in bicycling and human powered developments. It's showing some definite progress toward finding out if there are alternate ways to pedal.

STATEMENT OF THE OWNERSHIP, MANAGEMENT AND CIRCULATION OF *Bike Tech* REQUIRED BY ACT OF OCTOBER 23, 1962: SECTION 4369, TITLE 39, UNITED STATES CODE, FILED, *October 1, 1984*. *Bike Tech* is published *bimonthly* at 33 East Minor Street, Emmaus, PA, 18049, publication and general business offices.

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newsline

THE PRACTICAL VEHICLE AND COMPONENT COMPETITION, sponsored by *Bicycling* Magazine and the International Human Powered Vehicle Association, has already received over 100 inquiries. The purpose of this competition is to encourage development of more practical human powered vehicles for use on existing streets and roadways. While speed can sometimes be an important design element, there are countless practical features that, if incorporated into HPVs, could make them much more useful for various purposes besides racing.

There are actually two separate competitions: The Practical Vehicle Competition and the Practical Component Competition. The Practical Vehicle should be better than a conventional bicycle by providing better protection, superior lighting, greater cargo capacity, and easy maintenance. Judges will be looking for designs that have high pedaling efficiency and are safe on existing roads in automobile traffic. The Practical Component designs could include improved brakes, signaling systems, weather ponchos, energy storage devices, or any other design that is a practical improvement over existing components.

Designs will be judged at the 11th International Human Powered Speed Championships on September 26-29, 1985. Winners for overall vehicle and better component design will receive $5,000 and $2,500 respectively. Information packets for entrants are now being prepared. For entry information, write to *Bicycling* Magazine, PVC Competition, 33 E. Minor Street, Emmaus, PA 18049. Participants do not have to be a citizen of the United States.

REFLECTORS ARE NOT ENOUGH, SAYS NHTSA

The National Highway Traffic Safety Administration (NHTSA) has published its long-awaited study of the ability of overtaking motorists to detect bicyclists at night. The final report concludes that it is essential for bicyclists to use an "active source" of light in addition to the legally-required standard CPSC reflectors, and suggests that "those who ride regularly at night" should use "one of the available high intensity lighting systems."

The study included field tests of how easily an overtaking motorist could detect and recognize various commercially-available lights and reflector devices; here is a partial summary of results:

Device	Detection Distance	Recognition Distance
Highway barricade (7 inch dia. flashing amber light on a 3 ft by 3 ft barricade with amber/white diagonal reflective stripes)	1119 feet	617 feet
Bike w/ standard reflectors plus leg light on rider	1303	481
Bike w/standard reflectors plus fanny bumper on rider	957	469
Bike w/standard reflectors	844	439
Stationary strobe light (Honeywell "Strobolight," 3 inch by 2 inch white lens)	1201	396
Flashlight carried by pedestrian	1379	316
Belt Beacon on static bicycle (no rider)	1341	24

Only the highway barricade had a recognition distance greater than 550 feet, the distance which the Institute of Traffic Engineers specifies as the stopping distance for a car traveling 55 mph.

On the question of whether reflectors provide a safe level of conspicuity, the NHTSA study made this comment: "Significant doubt must still exist concerning the efficacy of the basic reflectors as required by the CPSC and used on all bicycles in this study. The most complete accident investigation of bicycle/motor-vehicle accidents in the literature (Cross and Fisher, 1977) indicates that most bicyclists struck at night had their required rear reflectors in place. Hence, something in the driver/bicyclist system is likely negating the inherent conspicuity of these reflectors as measured in this experiment. Driver intoxication, particularly at night, is certainly a major factor in nullifying the standard reflectors, but other influences, such as the possible confusing meaning of the single, bright, red rear reflector, must also be considered." The report also commented that active light sources performed significantly better in the tests than "passive" reflector devices, and noted that motorists are becoming familiar with the distinctive "signature" of the cyclic up-and-down motion of pedal reflectors used with a leg lamp.

The work was performed as part of a three-year study on bicycle conspicuity, and was funded by NHTSA for $250,000. Copies of the report, titled *Conspicuity for Pedestrians and Bicyclists: Definition of the Problem, Development and Test of Countermeasures* (Report No. DOT-HS-806-563, April 1984), by R.D. Blomberg, A. Hale, and D.F. Preusser (Dunlap and Associates East, Norwalk, CT) may be purchased through the National Technical Information Service (NTIS), Port Royal Road, Springfield, VA 22161. Also available from NTIS is a companion report, *Review of the Literature and Programs for Pedestrian and Bicyclist Conspicuity* (Report No. DOT-HS-806-564, April 1984), which provides an annotated bibliography and a summary of bicycle conspicuity programs in various states and foreign countries.