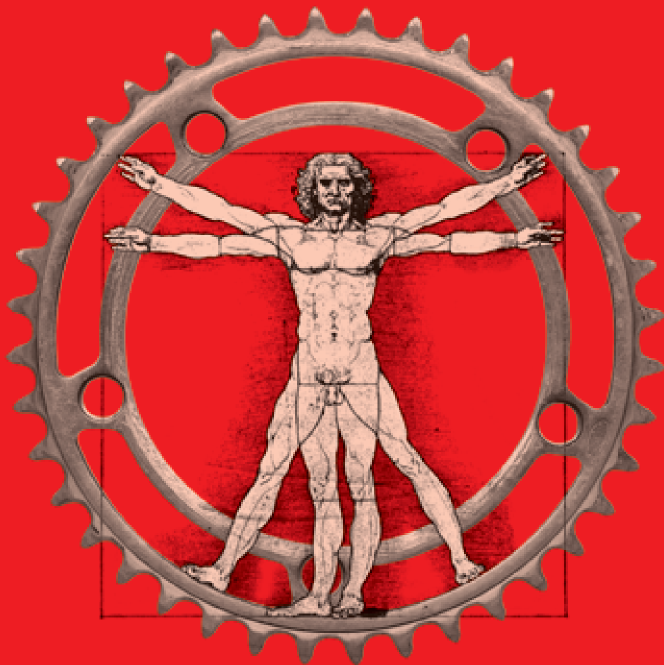

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IN THIS ISSUE

Propeller simulation with PropSim
Our IHPVA chair Theo Schmidt explains how he came to write a propeller-simulation program, what one can do with it, and how it works. He does so in a delightfully modest way, but it is a very useful program for all but out-and-out racers, and even then some races have been won with propellers designed with Prop-Sim. Theo makes it available to IHPVA members.

The Bodysail: improved bicycle sailing
There is now general agreement that fully faired HPVs are dangerous in crosswinds. Therefore Peter Sharp's report of the development in Canada of a seemingly huge sail carried high up on an "upright" bicycle and operated safely (though not in streets!) in high winds is stimulating and stunning.

Simple approximations for the effects of tire resistance, wind, weight and slope
Jim Papadopoulos provides rule-of-thumb (but mathematically derived) methods of estimating the effects of changes in the roadway, weather and HPV on speed. The rules often go against accepted beliefs.

MiniCal: an introductory spreadsheet for determination of power use while cycling
John Tetz's treatment of the power equation for vehicle propulsion is the complement to Jim Papadopoulos' approximations that can be used while riding. The spreadsheet MiniCal is used on a computer to generate points or lines or complete plots, often quite beautiful, showing the effects of various changes (in road slope, vehicle conditions, and wind) on the power input required. It is something that would therefore be used in the design stage of a new vehicle or in the analysis of the performance of existing vehicles.

MiniCal itself is not given here, but is available at cost to members as a separately published (and edited) monograph.

Optimum body shapes for bicyclists
Mark Drela, Jim Papadopoulos and Doug Milliken discuss the effects of body shape on bicycling performance in this short technical note.

Aerodynamic advantage from using fewer spokes, and Optimum pilot for a human-powered helicopter
These are two more technical notes by Mark Drela, who produces elegant simple and, for your editor, irresistible models of interesting aspects of HPV performance.

Crank-arm length
Danny Too updates his article in the last issue and gives a great deal of information on the optimum crank length for different circmstances.

A tandem recumbent design
Charles Brown gives sketches of his own tandems and discusses reasons for design choices.

Crashworthiness analysis of ultralight metal structures
This technical note is an abstract of an MIT doctoral thesis by Sigit P. Santosa.

Transmission efficiencies.
Your editor has constructed a technical note from various contributions giving sometimes varying measurements and estimations of the efficiencies of different transmissions.

Review: Chasing ricksaws
Carl Etnier, who is a pedicab driver, reviews an intriguing new book.

Editorials
Your editor has contributed a piece on HPVs, health and spinning. A longer and very interesting guest editorial on non-circular drives is by Dave Larrington, editor of the newsletter of the British Human Power Club.

Letters
Letters are on the effects of pedalling on wind resistance; on bottom-bracket height; on suspension specifications; and on a correction to the new numbering system introduced in the last issue (which becomes issue 47).

Propeller simulation with PropSim

by Theo Schmidt

ABSTRACT
PropSim is an easy-to-use propeller simulation program for evaluating cruising propellers. The basic functions and usefulness are described in a first part, and the way it works in a second, where emphasis is given to explaining propeller physics in simple terms.

INTRODUCTION

Many years ago I needed a propeller for my first human-powered boat. The power-boat propellers available seemed unsuitable, so I wrote to Gene Larrabee, then a professor at MIT, who had designed propellers for Paul MacCready's Gossamer human-powered aircraft and had published various articles on optimal propeller design [1], [2]. Larrabee designed two propellers for me; the experimental models I made from his designs worked very well. However, I really wanted to know more about the topic and to design propellers myself.

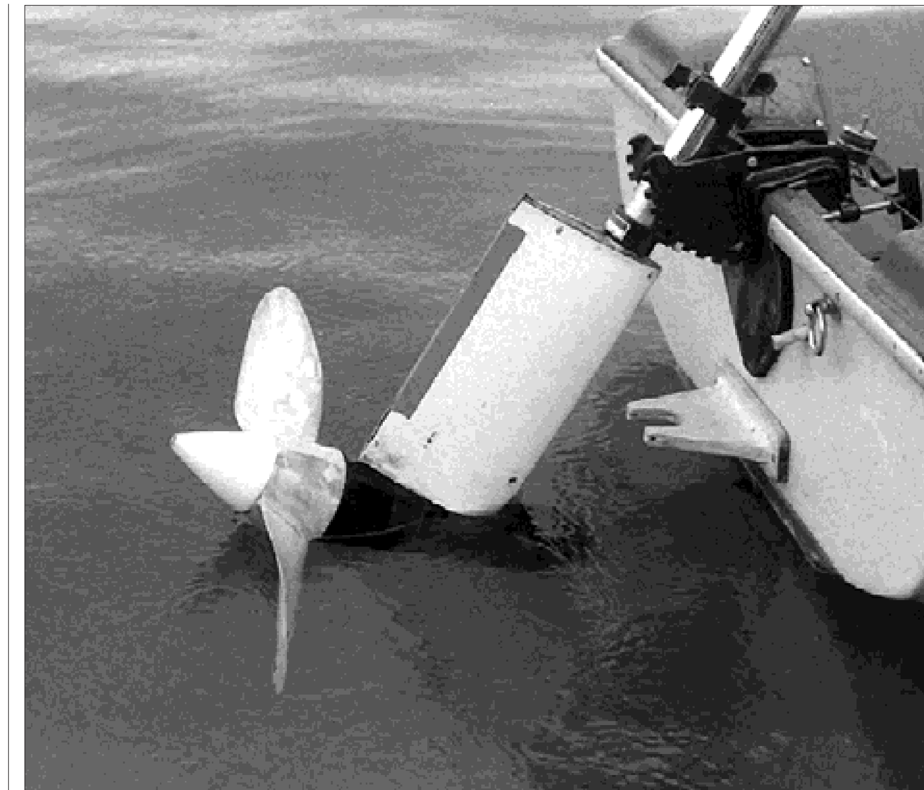


Figure 1. Two-bladed propeller.

Larrabee's minimum-induced-drag design method had two disadvantages. The main one was that I couldn't sufficiently understand the calculus involved! The other was that his method designed an optimum for a specified operating point. This is great for records and racing, where you do have a specific operating point (i.e., power required at a certain speed) and can change propellers for different events. I was more interested in cruising, where wind and waves dictate quite different loads at different times. I wanted a good compromise over a large operating range. The Larrabee designs had high-efficiency peaks of over 90% at their design points, but tended to stall (suddenly lose lift) when overloaded, thus losing efficiency. This is typical for the slender, aeronautical-type blades with high pitch/diameter ratios.

On the other hand, traditional boat or ship propellers don't stall, but don't reach very high efficiencies anywhere in their operating range. They are designed for relatively high loadings in order to minimise the craft's draft, and often for high speeds. Such propellers have wide, sometimes even overlapping blades, resulting in both considerable wetted-surface drag and tip-losses. Typical stall-proof boat propellers also have relatively low pitch/diameter ratios. They must thus turn quite fast, again resulting in much wetted-surface drag. Another consideration important with today's high-powered boats is cavitation, which occurs when the local pressure on the propeller blade's convex surface becomes so low that a cavity occurs, or more accurately, that the water begins to boil even at ambient tempera-

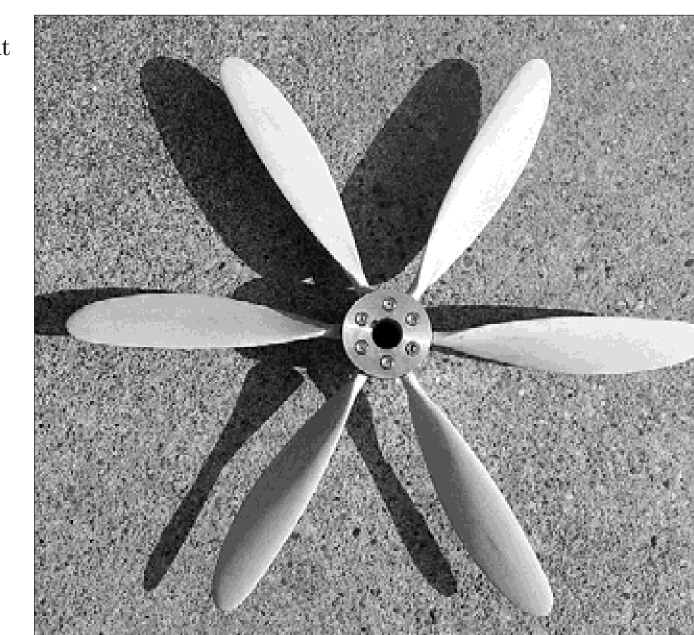

The two propellers shown here in two solar-boat applications, were developed for human-powered boats. The two-bladed prop is my standard cruising propeller which has been used for numerous boats powered by one or two persons, and in one case, twelve persons. The six-bladed prop can be pitch-adjusted and also used with fewer blades. Jochen Ewert's flying hydrofoil craft uses a two-bladed version of this propeller. Photos, Theo Schmidt.

Figure 2. Six-bladed propeller.

ture. With human-powered propellers and hydrofoils this is not a problem so far.

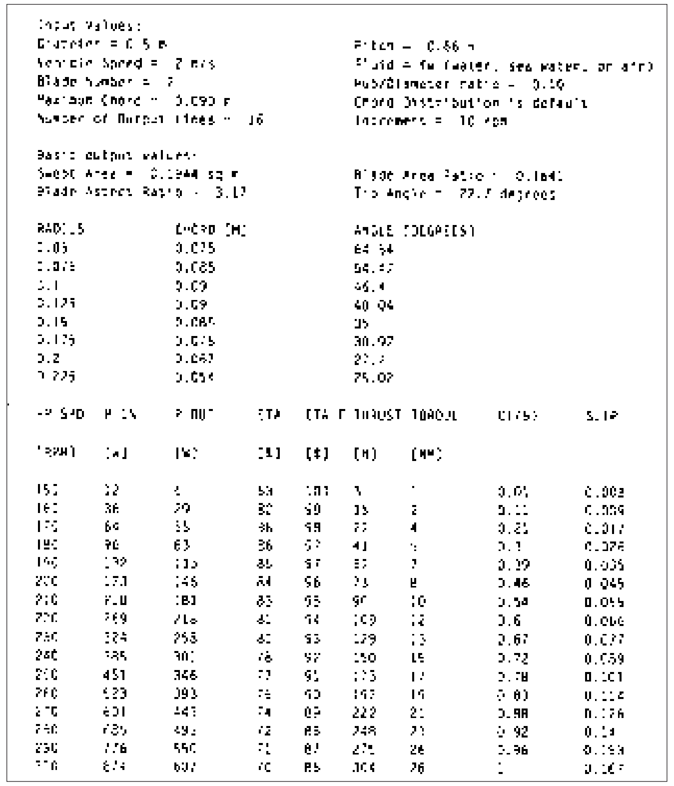
It is plausible that good all-round cruising propellers for low-powered boats are somewhere between the aeronautical-type and the ship-type propellers. In order to evaluate designs suitable for cruising I really needed a simulation rather than a design program. With nothing available to me to run on my "toy" computer, I wrote my own program PropSim in the BASIC programming language and published the first version in *Human Power* [3], along with a specification for a good cruising propeller which has since been made about 50 times. Quite a few people inquired and got copies of the program. Some, like Christian Meyer, expanded and improved it. Many people weren't able to use the program because their computer's understanding of BASIC was different to mine: there are quite a few dialects about. Therefore I've rewritten the program in a new version with some improvements and am making it available as a stand-alone application, at present only for the Macintosh PowerPC, free to *Human Power* subscribers.

PART 1: WHAT CAN YOU DO WITH IT?

PropSim is suitable for studying the behaviour of air or water propellers used for human-powered craft. It can also be used for power applications as long as reasonable speeds and blade loadings are not exceeded. Although a simulation program, PropSim does calculate and output suitable chord and twist-angle values of the blades when given the maximum chord (i.e., blade width), propeller diameter, and pitch. You further specify the boat speed, medium (fresh water, sea water, or air), and number of blades.

The following (fig. 3) is a PropSim input/output page with data corresponding nearly to the propeller in fig. 1.

You thus get a table of output values (e.g., power and efficiency) for a suitable range of propeller speeds starting at that speed at which the propeller freewheels, i.e., produces no thrust. If your input values are reasonable, one propeller speed (i.e., output line) will correspond to the values of a particular craft and situation. The better you know the thrust required for your boat at a particular speed, the better you can optimise the propeller by



```
Input Values:
Diameter = 0.5 m                     Pitch = 0.66 m
Vehicle Speed = 2 m/s                Fluid = fw (water, sea water, or air)
Blade Number = 2                     Hub/Diameter ratio = 0.10
Maximum Chord = 0.090 m              Chord Distribution is default
Number of Output lines = 16          Increment = 10 rpm

Basic output values:
Swept Area = 0.1944 sq m             Blade Area Ratio = 0.1[illegible]
Blade Aspect Ratio = 3.17            Tip Angle = 22.7 degrees

RADIUS        CHORD [M]       ANGLE [DEGREES]
0.05          0.075           64.54
0.075         0.085           54.47
0.1           0.09            46.4
0.125         0.09            40.04
0.15          0.085           35
0.175         0.075           30.97
0.2           0.067           27.7
0.225         0.054           25.07
```

PROP SPD [RPM]	P IN [W]	P OUT [W]	ETA [%]	ETA F [%]	THRUST [N]	TORQUE [NM]	C(75)	SLIP
150	12	4	53	101	3	1	0.05	0.002
160	36	29	82	99	15	2	0.11	0.009
170	64	55	86	98	27	4	0.21	0.017
180	96	83	86	97	41	5	0.3	0.026
190	132	115	86	97	57	7	0.39	0.035
200	173	146	84	96	73	8	0.46	0.045
210	220	183	83	95	90	10	0.54	0.055
220	269	216	81	94	109	12	0.6	0.066
230	324	255	80	93	129	13	0.67	0.077
240	385	301	78	92	150	15	0.72	0.089
250	451	346	77	91	173	17	0.78	0.101
260	523	393	75	90	197	19	0.83	0.114
270	601	443	74	89	222	21	0.88	0.126
280	685	492	72	88	248	23	0.92	0.14
290	776	550	71	87	275	26	0.96	0.153
300	874	607	70	85	304	28	1	0.167

Figure 3. Input/output page from PropSim.

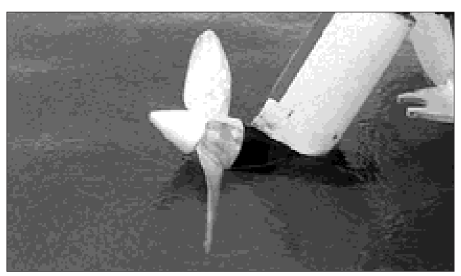


Figure 1 (smaller version from page 3). Two-bladed propeller.

adjusting the input values until you get the efficiency peak where you want it.

For a racing propeller you would then concentrate on varying several values in order to maximise the peak efficiency.

For a cruising propeller, you would examine the behaviour at several boat speeds and come to a compromise which best suits the intended use.

Figure 4 is the graphical output of a spreadsheet program, corresponding to the data in figure 3. A good overall efficiency requires a Froude efficiency (ETA F in fig. 3) between 95% and 99%

ASSUMPTIONS AND LIMITATIONS

- PropSim assumes and outputs unshrouded straight blades with correct twist for light loading. Things like simple flat plates or variable-pitch props used off the design point cannot presently be modelled, nor can highly skewed or bent blades ("weedless" designs) or props used in rings or tubes (e.g., bow thrusters, Kort nozzles).
- You have a choice of four reasonable blade planforms without knowing which of these (if any) is the optimal one. Yet the program assumes an optimal planform, so in a general case the output values will be slightly optimistic. (This is the consequence of not using Larrabee's minimum-induced-drag method.)
- The program uses a basic medium-thickness flat-bottomed round-nosed foil section (Clark-Y). Strongly differing sections are not correctly modelled.
- Reverse thrust (braking or turbine modes) is not modelled.
- Overloaded (stalled) blades are only approximately modelled.
- Hubs are modelled, but the larger the hub, the less accurate the result.
- No solid-material properties are assumed. Thus it is possible to design highly efficient propellers that are not buildable in practice (although a little common sense or carbon fiber goes a long way!).
- Present program versions (e.g., 4/99) run only on Power Macintosh computers (probably any type). Sorry Bill Gates!
- Present program versions can print, but do not save files or copy to the clipboard. To draw diagrams, you would have to enter the data into an appropriate program manually or with an OCR program. The program remembers your input values only as long as it is kept running.
- The only unit system presently supported is metric SI with propeller speed in revolutions per minute and angles in degrees.

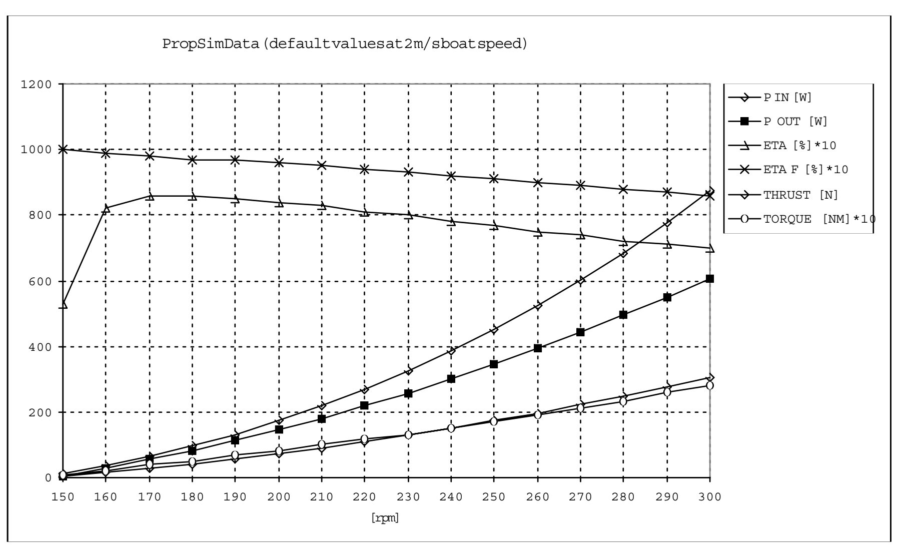


Figure 4. This is the graphical output of PropSim via a spreadsheet program, corresponding approximately to the two-bladed Prop in Fig. 1. A good overall efficiency requires a Froude efficiency (ETA F in figure 3) between 95% and 99%.

WHAT GOOD IS IT?

With all these limitations you might wonder what good the whole thing is. It turns out that most of the limitations are not very important unless you are after extreme achievements, where only the very best is good enough. For most practical purposes it is the overall performance which counts, and this is highly dependent on the basic geometry: diameter, pitch, and blade area. Planforms and sections are then of secondary importance. Using the program, you can for example determine a geometry which "feels" good at low speeds, which may result in a more popular product than some, or indeed avoid really poor mismatches sometimes proposed even by large companies or knowledgable people. The program is now easy to use and runs quite fast, so the missing graphical and filing capabilities are not catastrophic.

VALIDATION

Although there is good agreement [3] between some results from PropSim and those of more sophisticated programs, this is probably a coincidence. As I have never accurately measured any propellers and there are so many influencing factors, it is very difficult to say how accurate the results are. My cruising propellers designed by this method have been very successful and some have even won races, but I think the program is probably not good enough to make propellers suitable for breaking present hydrofoil and aircraft speed and distance records.

WHERE CAN YOU GET IT?

The new versions of PropSim are available free of charge to members of HPV or research organisations for personal use. In order to get PropSim, e-mail me at <tschmidt@ihpva.org>. For those without online access, disks may be made available subject to a postage and handling charge. Programmers interested in improving the program or porting it to other operating systems than Power Macintosh should ask for the BASIC source code.

PART 2: HOW PROPSIM WORKS

As PropSim is not a design program, but rather a simulation program, you have to enter some halfway-sensible parameters to begin with and the program will behave like a virtual test tank. It is mathematically inelegant, using no calculus, but only simple theory and numerical methods to arrive at solutions, a task well suited to fast number-crunching computers. The underlying method used is the *actuator-disc theory*, which describes the behaviour of a "perfect propulsor" acting continuously on a "disc" perpendicular to the direction of fluid flow. One could say it "couples" perfectly to a disc of fluid which is continuously replaced. This theory is valid for any fluid and for our purposes there is no difference between air and water as long as the physical characteristics are correctly modelled. Secondary effects like cavitation or supersonic flow also have no bearing on the type of propulsor we are interested in and are not modelled.

Because there are many misconceptions on the way a propeller (or indeed any propulsor) works, the remainder of this article attempts to explain propulsion physics in simple terms, using PropSim's inner working as an example. The source code is available from the author or the essential parts can be found in [3].

BASICS BEHIND THE ACTUATOR-DISC THEORY

In order to produce any propulsive force on any craft, you must have matter to react against. In the case of land vehicles using wheels (or barges using poles) to push against the ground, this matter is the ground underneath the vehicle, and as the ground is very stiff, you are pushing against the whole earth, a huge mass which moves backward a tiny amount as you move forward. (This is action = reaction, Newton's third law.) As the earth has very much more mass than have you and your vehicle, you move forward almost the full amount defined by your wheel rotation and the earth moves back only an imperceptible amount. Discounting the small amount of tire slip, you have a propulsive efficiency of practically 100%.

Instead of reacting against the earth by turning the wheels with pedal cranks, you could instead throw bricks out the back and propel your vehicle this way. This is the principle by which rockets move but it is really the same thing: you are reacting against the mass of the brick as you throw it, producing a propulsive force equal to its mass times its acceleration (i.e., the applied speed increase), Newton's second law. The faster you can throw the brick, the lighter it can be for the same effect, and the more bricks you can carry (space-propulsion systems emitting ions at nearly the speed of light can operate for years while expending very little mass). In order that you don't run out of bricks, you could previously lay them out to pick up on the way; this would then be the principle by which jets or indeed propellers operate: they intercept the fluid at rest along their path and act on it. To propel a boat or airplane you are thus "picking up" and "throwing" parcels of water or air out the back; it is exactly the same as with bricks and has little to do with the Victorian notion that a propeller pulls its way through a medium like a screw through a block of wood, although the geometrical concept of "pitch" (distance between two screw threads) is useful in the special case where there is no "slip": at each turn the screw advances by the distance equal to the pitch.

Now you can produce the required force either by intercepting a large parcel of fluid with a large propeller and speeding this up only slightly, or by using a small propeller or even a ducted impeller in order to speed up a small parcel a greater amount, right up to a high-speed jet. The propulsive force is created at the point where the fluid is accelerated (impeller and nozzle) and not because the jet pushes against the water or air, as many people think. Thus the jets of many motorised water craft actually exit above the water line. Now these small units would be really neat except for one thing: each parcel of of fluid also carries a kinetic energy equal to 1/2 times its mass times its velocity squared, energy which is lost to the propulsive system. Doubling the jet speed you need only half the mass, but you get twice the energy loss, i.e., the inherent loss of a propulsive system is a linear function of the jet velocity.

Therefore it is clear that we must strive towards a small velocity increase and a large mass. The mass available per second is equal to the fluid density times the volume acted upon by the actuator per second. This is equal to the distance travelled per second times the actuator area (perpendicular to the direction of motion). A high propulsive efficiency thus requires either a high vehicle speed or a large actuator disc, whether this is a propeller, oar, or paddle wheel. Boats with relatively small propellers or narrow high-speed jets have very poor propulsive efficiencies at low speeds. At high speeds, the situation improves, as more parcels of water are presented to the propeller or jet drive and thus must be accelerated only slightly in order to achieve the high mass-per-second throughput desired. Thus high-speed craft can use smaller propellers or even jets somewhat efficiently when planing or hydrofoiling, whereas slow craft or heavily loaded craft need the largest propellers that are practically possible.

The state of affairs above is described in simple equations by the actuator-disc theory. The efficiency of an ideal propulsor worked out this way is called the *Froude efficiency* and is a natural limit which cannot be exeeded by any device, no matter how good it is. Any propulsor which has virtually zero slip in the water, whether this is a very large propeller or a huge drag device, approaches 100% Froude efficiency. The essence of the actuator-disc theory is that if the slip is defined as the ratio of fluid velocity increase to vehicle velocity, the Froude efficiency is 1/(slip + 1).

BLADE FORCES

The second component needed to calculate propellers is simple foil theory applied to propeller blades. A blade or wing moved through a fluid generates a force by the very act of accelerating and redirecting fluid as described above. This force can be resolved into components which are perpendicular or parallel to the blade movement, called lift and drag, respectively, or into another pair of components perpendicular or parallel to the direction of vehicle motion, called torque (when multiplied by the local radius) and thrust, in the case of a screw propeller. This is very similar to a wing except that the blade is twisted, so that the blade must be divided into several segments which are treated separately.

Using the propeller diameter, pitch, and rotational speed and also the boat speed, PropSim first works out the angles of ten segments corresponding to an *ideal helix* which would slide through the water with no disturbance at all if rotated at exactly the specified speed. A thin foil shape corresponding to this helix would generate no lift and no thrust at the propeller speed which corresponds to one revolution in the time it advances a length equal to its pitch, i.e., the pitch per time unit must equal the boat speed, giving a zero angle of incidence on the blades. When the propeller is turned faster, this angle increases and propeller theory predicts a lift force increasing in proportion. At angles under 10 degrees this is the same for all usual wing shapes and agrees closely with what is measured in practice. Other values affecting lift are the surface area and the aspect ratio of the complete three-dimensional blade: short fat blades have considerable pressure loss around the tips whereas long narrow blades are less affected this way. (This is where the first propeller designers erred: thinking in terms of the wood-screw model they thought they would get minimal slip with wood-screw-like propellers. However these have tremendous pressure losses around the edges and were thus particularly bad, until one day one got broken accidently and performed much better, leading the way to modern propellers!)

Now PropSim must determine the drag force of each blade segment. At this point, propeller theory becomes very complicated or unknown and we must make use of values measured in tanks or wind tunnels, which are available as tables or diagrams for a great variety of different wing sections. Sets of measurements are always similar if the so-called *Reynolds number* is the same, no matter what the size or speed of the blade or whether the fluid is water or air. All that is needed are tabular data at various Reynolds numbers and the size, speed, and angle of the blade. PropSim looks up the drag data in a table as a function of lift and Reynolds number, using data for the Clark-Y foil section, which is similar to the Eppler 193g.u. and Eppler 205 profiles, and is relatively easy to define and make because of the flat bottom surface. The segment forces are resolved into thrust and torque components and added up. A further correction is also applied in order to compensate for the tip losses mentioned above. This is called induced drag. In calculating this, PropSim deducts only the theoretical minimum loss, i.e., it assumes that the blade's planform is optimal. This corresponds to an elliptical lift distribution of an untwisted blade. In the case of a screw propeller, Larrabee has shown that the requirement for minimum induced drag is a uniform wash velocity, i.e., the same local slip values for all segments [1]. It is therefore planned to add a numerical optimisation routine into PropSim to adjust the segments' chord dimension in order to arrive at this condition. Until this is implemented, PropSim simulations will be slightly optimistic for blade planforms which do not happen to be exactly correct.

NUMERICAL SOLVING

Now we must marry the two sets of calculations described above. The propeller blades sweep out a virtual disc in the fluid: this is our actuator disc. Initially we assumed no slip for the force calculations although physically there must be some if the prescribed force is to be generated. Using the first results, we can work out the slip or stream velocity required to produce the same forces in the actuator disc. The force calculations are now repeated using the new slip value, and they will be seen to have changed a bit. PropSim keeps doing this until the values no longer change. These calculations almost always quickly converge towards a solution unless wild, nonsense, values are used as inputs. The result is thus a numerical solution for the velocity of the slip stream and all corresponding forces, whence total power and efficiency values can be derived. Once this solution has been found, all desired values are printed, giving a single line of output corresponding to a single operating point.

The propeller speed is now increased by a specified increment and everything repeated until we have a table of propeller values as a function of propeller speed, which is shown on the screen or printed out, as shown above in Part 1. Now the boat speed could be increased and the whole procedure repeated, so the end result is a set of tables describing the propeller behaviour over a wide operating range. It is important to develop a feeling for the physical parameters and not to go outside sensible boundaries. I hope that future versions of PropSim will draw fancy diagrams or at least prepare files suitable for drawing diagrams in other programs. Far-future versions may even have some optimisation routines, but I would be delighted if some of you gentle readers accomplish these improvements before I do!

ACKNOWLEDGEMENTS

Thanks to Gene Larrabee for having started it all, and Dave Wilson, Bob Stuart and Michael Lampi for reviewing this article and giving suggestions.

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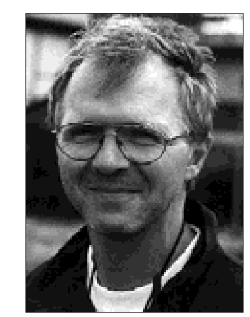
Theodor B. Schmidt, 44, has lived in the USA, Wales, England and now in Switzerland, married, in an old farmhouse full of HPV and boat components (but few actually functioning vehicles!). Degree: BSc (Wales) in Physical Oceanography and Electronic Engineering. Occupations have included domestic solar installation, kite-sail, solar hydrogen, vehicle safety, and Stirling-engine research (with the Center of Appropriate Technology in Langenbruck, Switzerland), and now mainly human- and solar-boat consulting. HPVs and boats are a vocation, as is lobbying for and trying to live in, a car-free healthy environment. Vehicles built and owned include several solar/human-powered tricycles and electric bicycles, scooters, and boats powered by just about everything except internal-combustion engines. Theo is current chairman of the IHPVA.



LETTER

AN OVERLOOKED ISSUE

I enjoyed the latest issue of *Human Power*, as always. I read it cover to cover when it first arrived. This morning, desperate for an HPV fix, I looked back through it, hoping I had missed something. It turns out that I had not gone over the renumbering list. I came across a few things you will want to know.

1. Issue no. 4 is labelled fall 1979 (issues 2, 3 and 4 have no volume or issue numbers).
2. Issue no. 5 is labelled Spring 1980.
3. Alas, you missed volume 2 number 2 …as a consequence, issue 46 is actually no. 47.

Hope I haven't spoiled your day.

—Warren A. Berger

Mea culpa! You did spoil my day, Warren! However, it is good to have errors corrected early rather than late. A corrected table (see p. 23 of the last issue, now no. 47), will be printed in the next issue.

—Dave Wilson

CONTRIBUTIONS TO HUMAN POWER

The editor and associate editors (you may choose with whom to correspond; see mast, p. 2) welcome contributions to *Human Power*. They should be of long-term technical interest (notices and reports of meetings, results of races and record attempts. Articles in the style of "Building my HPV" should be sent to *HPV News*). Contributions should also 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 or 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-affiliate members and nonmembers.

The Bodysail: improved bicycle sailing

by Peter A. Sharp

Previous attempts to combine bicycles and sails have achieved very limited success. They have included conventional triangular sails mounted on a mast held outboard on one side of the bicycle or mounted on a mast extending up from the head tube, or small sails mounted behind the rider. Typically, they have been difficult to control if they were large enough to provide significant power.

But Linc Alexander, an airplane pilot living (or once living) in British Columbia, Canada, devised a sophisticated and powerful sailing system for bicycles (fig. 2) that has proven to be remarkably stable. The system is called the "Bodysail" (also called "Bodisail") because it is worn like a back pack with shoulder straps and a hip harness. It achieves a considerable degree of control over the sail and is well suited for recreational use with either bicycles or a specially designed scooter (with a high platform and handle bars that can be quickly adjusted to the size of the rider, plus other amenities; fig. 1). [We haven't been able to locate Alexander, so that we don't know the current state of the system. —*Editor*]

The rectangular, symmetrical sail is surprisingly large to be used on a bicycle—3 meters tall by 1 meter wide. An even larger, advanced model is 3.5 meters tall. (That sail area is about the same as for small, racing sand sailers, which place their rear wheels about 2 meters apart in order to keep from tipping over.) The Bodysail rider achieves stability by leaning against the wind, somewhat like a sail boarder or a sail skater on ice. The sail, itself, is always rotated around the back of the rider to the windward side. The sail and its controls are sophisticated yet simple to use.

The sail is mounted on pivot arms which extend from lockable hinges mounted on the rider's "back pack", and the middle of the sail is at about the height of the rider's shoulders. Viewed from above with the sail vertical, the sail forms the concave top of a "T", the pivot arms form the body of the "T", and the lockable hinges are located at the bottom of the "T". A cloth sail is stretched between two vertical masts at (and perpendicular to) the tips of the "T" top, one of which contains a reefer roller (spring loaded) used to store the sail. The masts are spread apart by pairs of upper and lower support arms, which are adjustable for width using an hydraulic hand pump. That enables the rider to decrease the area of the sail by any amount in strong winds. The camber, or belly, of the sail can be adjusted as well by first locking the reefer roller and then moving the masts closer together or farther apart.

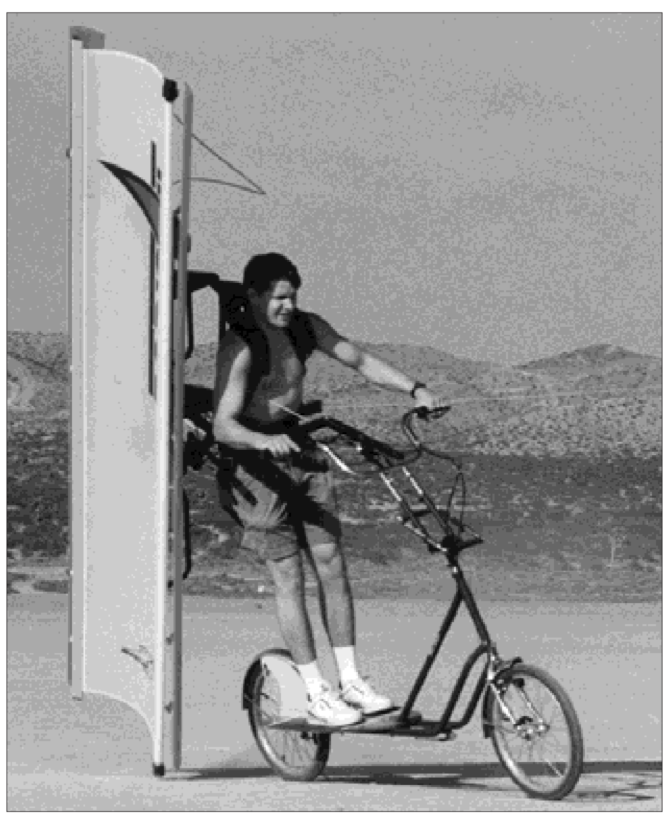


Figure 1. "Bodisailor" on a specially-designed scooter.
—Photos provided by author

While riding, the rider uses one hand periodically to adjust the control arm, which rotates the sail and also locks it into position. The control arm is fixed to the pivot arms, and then curves around the right side of the rider, under the rider's armpit. The control arm also has an elbow joint that permits the handle of the control arm to be moved sideways into a comfortable position. The handle of the control arm has three release controls: a hinge-release lever, a roller-reefer release lever, and an hydraulic-pressure release button.

Simultaneously pressing the hydraulic-release button, while squeezing the release handle for the roller reefer, collapses and reefs the sail completely within two seconds. The handle of the hydraulic hand pump is located close to the rider's right hip. It also has a release for the roller reefer. In addition to these controls, the rider can lean forward so as to change the position of the sail from near vertical to near horizontal. This technique reduces drag when turning into the wind and when rotating the sail to the other side.

The weight of the last-known version of the Bodysail system is about 15 kg. (32 lbm), but the use of advanced materials is expected to reduce that weight. The comfortable "back pack" arrangement enables the rider to feel the wind as a firm but gentle push against his back. While the Bodysail is not recommended for street riding—because the large, tall sail is leaned to the side, and because the wind force of the sail can be reduced quickly but not instantly—experienced riders have occasionally ridden on appropriate roads. Control of the sail, and the necessary balancing techniques, are learned quickly. This is apparently due to the simplicity of the controls and the feedback provided by "wearing" the sail.

Falls are rare. An experienced rider can steer straight ahead, without swerving, even in strong winds. For instance, Alexander has ridden in winds gusting as high as 18 m/s (40 mph), and he notes that he has never fallen. The top speed of the Bodysail is about twice the speed of the wind, as compared to low (recumbent) "sand sailer" tricycles which achieve about two and a half times the speed of the wind. This difference is due to the higher aerodynamic drag produced by the upright position of the Bodysail rider. But it is the higher center of gravity of the Bodysail rider which helps to provide good stability on two wheels. Balancing the Bodysail seems to be very similar to balancing a bicycle. When riding the special scooter, the rider's center of gravity is even higher than when riding a bicycle, thus increasing stability. On the other hand, when using human power, it is easier to propel a bicycle than a scooter. The scooter's main advantage is that it further improves the fun factor by increasing the rider's freedom to shift, twist, and bend his body while sailing.

Figure 2. Bodysail worn by bicyclist.

Because the Bodysail is rotated out to the windward side of the rider, the aerodynamic force vector of the sail usually passes through, or in front of, the rider's center of gravity, even though the center of pressure of the sail is somewhat to the rear of the rider. This may partially explain the good stability of the Bodysail. (Placing the sail's center of pressure directly to the rear of the rider might otherwise be expected to cause instabilities.)

The Bodysail is convenient to transport and to store when folded and installed in its carrying bags. It therefore has the potential to become the basis of a new cycle-sailing sport. It is also likely to encourage other inventors to explore the combination of sails and cycles. If so, the Bodysail could turn out to be an important step toward eventually transforming the nature of cycling in the 21st century. Bodisail Systems Corp. was formerly at 1830 Kingsway Ave., Port Coquitlam, B.C., Canada V3C 1S5, but is no longer there. If anyone knows about its present whereabouts or that of Linc Alexander, would she/he please let me know? Peter A. Sharp, 2786 Bellaire Place, Oakland, CA 94601 USA; <sharpencil@pipeline.com>

Peter Sharp is a self-employed craftsman and an amateur inventor. He rides a Tour-Easy, a folding Dahon, and a 30-year-old Schwinn with a fat seat. His long-range goal is to make HPV the primary mode of transportation around the world. In two upcoming articles he outlines how that might be done.

LETTERS

Wind resistance pedaling vs. coasting

While on a six-day bike tour in Colorado recently I had several occasions to notice that I had significantly greater wind resistance while pedaling compared to while coasting. This was on my Speed Ross SWB recumbent, with an air speed of 25–30 mph (11–13.5 m/s). The typical situation would be when I was coasting down a gentle grade at 20 mph (9 m/s) with a 10-mph (4.5 m/s) headwind (resulting in a 30-mph (13.5 m/s) airspeed). If I started to pedal my speed would instantly decrease to 18.5 mph (8.3 m/s) and it would take at least five seconds of vigorous pedaling to regain the 20-mph speed that I had while coasting, and my speed would level out a about 21 mph. So I would generally quit pedaling since the effort didn't result in much speed increase.

I'll have to do more experiments at home with my new Fiberglass/Vivak nosecone. I presume that pedaling will have less effect on my wind resistance when my legs are sheltered by the nosecone.

–Wayne Estes, Mundelein, IL, USA
<Wayne_Estes@css.mot.com>

BOTTOM-BRACKET HEIGHT

This concerns the letters about bottom-bracket height and climbing, particularly Zach Kaplan's in HP 45 (13/3). My experience is about the opposite, I climb better with high-bottom-bracket machines. Zach and I have very different morphology (significant height difference), so there might be an anatomic reason to explain the difference….

Watching someone go through a "FitKit™" on a "headfirst" bike, one possibility came to mind: Response to bottom-bracket heights may be the recumbent equivalent of the "knee over pedal spindle" (KOPS) adjustment for which the saddle of an upright has slides, albeit in a much less convenient form. The KOPS adjustment is supposed to affect "efficiency".

On uprights, I would slide the saddle to its rear limit (long upper leg, much shorter lower). Zach, how about you?

–Jeff DelPapa <dp@world.std.com>

Zach Kaplan responded as follows.

Back when I rode uprights I'd generally slide the saddle to or close to its forward limit. However I think I did this because my upper body is short in relation to my leg length and it made it easier to reach the handlebars.

Interestingly, the same cycling podiatrist who told me the low-bottom-bracket recumbent position was better for me than the high-bottom-bracket position had previously told me when he fitted my upright that I should have the seat farther back.

–Zach Kaplan <zakaplan@earthlink.net>

Simple approximations for the effects of tire resistance, wind, weight and slope

by Jim Papadopoulos

The power required from the rider of a human-powered vehicle (or the engine of an automobile) moving at steady speed is given by

$P = V*[D*(V+V_{hw})^2 + M*G*(C_{rr} + S)]$

P is rider power

V is HPV speed relative to the road

V_{hw} is headwind velocity relative to the road

D is the aerodynamic-drag factor (half the product of frontal area A, drag coefficient C_d, and the air density ρ), with a value around 0.2 kg/m for an upright cyclist.

$M*G$ (kg $*$ 9.81) is the total system weight in N (that is, pounds $*$ 4.448)

C_{rr} is coefficient of rolling resistance with a value between 0.002 and 0.006

S defines the uphill slope in terms of the sine of the angle (negative if downhill). Since the sine and tangent are virtually equal for real-world hills (S up to 0.25), S is also approximately the 'percent slope' divided by 100.

This equation should not be considered 'exact', inasmuch as each term involves approximations. But as far as I know it is a reasonable approximation to the most important drag phenomena.

The power equation can be used to predict effects on speed in any circumstance (e.g., hill, headwind, etc.), due to changes in any quantity (e.g., aero drag, total weight, rolling resistance) for a fixed power level. But to do so generally requires numerical solution of the cubic equation for V, which very few of us can do routinely. (And cubic equations don't lend themselves to "thinking about" magnitudes.) It turns out that one can still make progress in many cases, by deriving simplified laws which give a good idea of some effects of practical interest. In fact they are simple enough to be remembered, and used mentally during a ride, if the user is so inclined.

These simplified laws are presented here. Their actual derivations, which are somewhat complicated, may possibly be found in the next, third, edition of *Bicycling Science*, in preparation, and/or in a future issue of *Human Power*.

1. EFFECT OF ROLLING RESISTANCE ON LEVEL SPEED

Let's look at rolling resistance. Define an ideal 'no rolling resistance' level speed, V_{nr}, as the speed reached for a given power output when rolling resistance is absent. Then we can calculate the approximate fractional speed decrease (at that same power level) due to any amount of rolling resistance:

$\Delta V/V_{nr} = [(M*G*C_{rr})/(D*V_{nr}^2)]/3$, or

$\Delta V = (M*G*C_{rr})/(3*D*V_{nr})$

It can be seen that the speed loss ΔV is proportional to C_{rr}, and inversely proportional to the ideal or no-rolling-drag speed, which is very close to the actual speed. (Note: here and subsequently, a 'Δ' in front of a quantity represents a 'delta' or 'change' in that quantity.)

In the fraction in square brackets in the first of these two equations, the numerator is the rolling drag (typically a little more than 2N or 0.5 pound), and the denominator is the aerodynamic drag at full speed (typically 15N or 3.5 pounds if unfaired, or half that if well streamlined). Ordinary rolling resistance therefore causes a decrease from ideal speed of about 5% in the first case, and 10% in the second.

As a specific example, suppose that on a level road on a day without wind you find that you can ride at 11 m/s, 24.6 mph, with tires that have a C_{rr} of 0.003, and you would like to know how much additional speed you could expect if you invested in expensive tires that had a C_{rr} of 0.002. You and the machine total 80 kg, and your value of D is 0.18 kg/m (from C_d = 0.9, A = 0.33 m^2, and air density = 1.2 kg/m^3). Using the second equation and inserting as an approximation your actual speed instead of your ideal speed, you calculate the full loss of speed due to rolling resistance as $(80*9.81*0.003)/(3*0.18*11)$, or about 0.4 m/s. (If you wish, you can then make a closer approximation to V_{nr} as 11.4 m/s, and make a closer calculation of the speed loss as 0.38 m/s or 0.86 mph.) The gain in speed from reducing the tire rolling resistance by a third is, then, one-third of this, about 0.13 m/s or 0.29 mph. This is a 1.2% speed increase, which would lead to a time saving of 43 seconds in an hour. (For a recreational rider this may not seem to be much, but for a racer it is enough to clinch a race decisively.)

This equation also shows the effect of system weight on level speed. But whereas rolling resistance can be reduced by (say) 33% relatively easily, it is hard to reduce total weight by more than two or three percent. Thus the time savings in an hour would be four seconds or less. If someone goes noticeably faster with lighter components, it is probably because of being psyched up!

2. EFFECT OF WEIGHT ON UPHILL CLIMBING SPEED

Now let's look at the effect of reducing rider-plus-vehicle weight, $W = M*G$, on the speed of climbing any size of hill. A one-percent reduction in system weight will increase speed for fixed power output, but by how much? Most people would assume a speed increase of one percent, but they would be wrong. The result depends strongly on the slope of the hill and the aerodynamic drag.

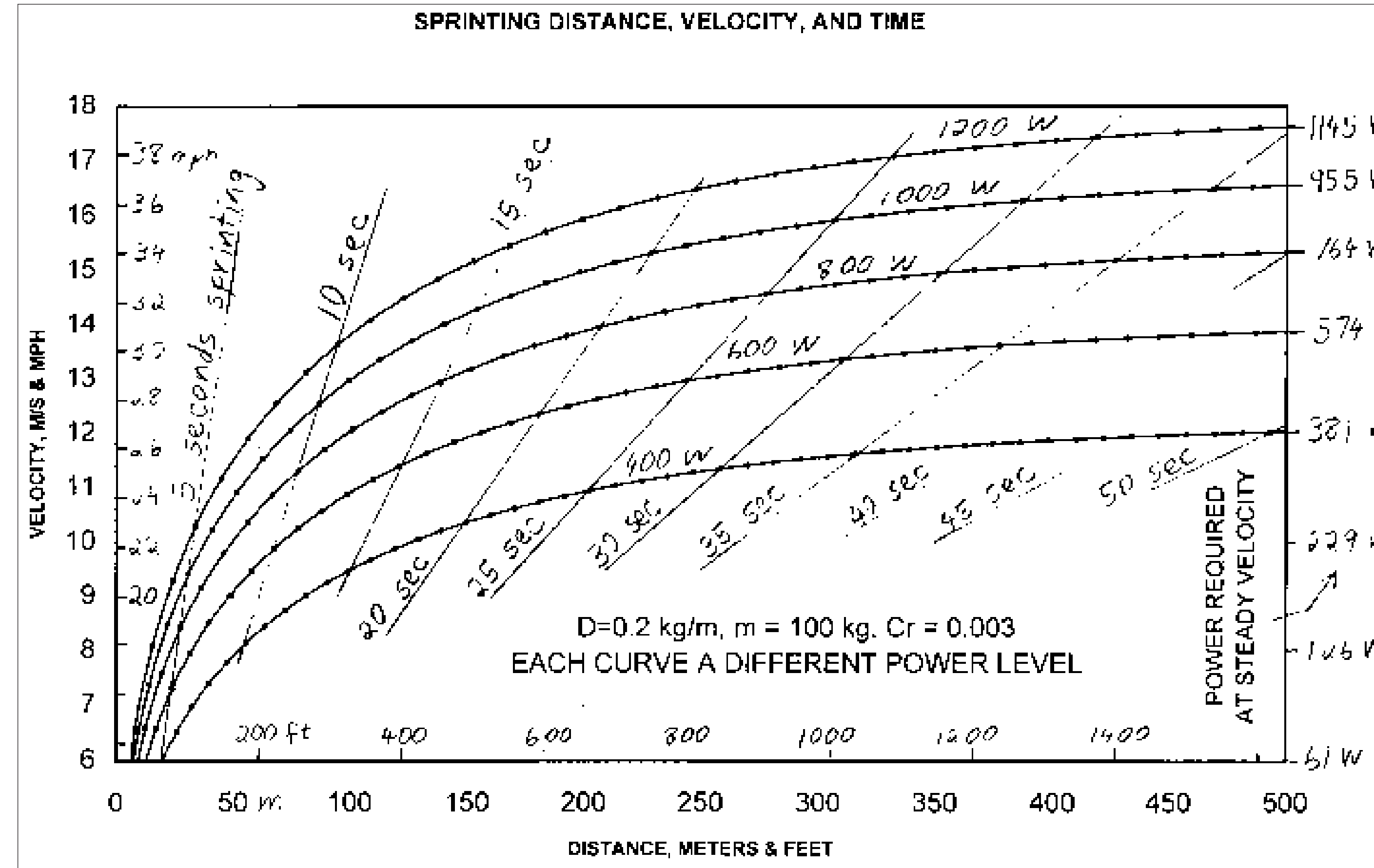
Luckily, one needn't solve the cubic power/velocity equation to find the effect of mass. The general rule is:

$$\frac{(\Delta V_{hl}/V_{hl})}{-(\Delta W/W)} = \frac{1-(V_{hl}/V_{nr})^3}{1+2*(V_{hl}/V_{nr})^3}$$

On the left is the ratio of expected percentage increase in hill speed V_{hl}, to percentage weight or mass decrease. In the expression on the right, (V_{hl}/V_{nr}) is the ratio of speed on any given hill, to ideal (no rolling resistance) speed on the flat, assuming the same rider power. As a good first approximation, instead of V_{nr} you could use V_{fl}, your actual speed on the flat.

To apply this relation, you have to know your own ratio for your own hill—does it slow you to half your normal speed? or to one third? On a hill which slows you to half your normal speed for constant power output, $(V_{hl}/V_{fl})^3$ is 1/8, and the right side of the equation is 0.7. Therefore a one-percent weight savings will lead to a 0.7-percent speed increase.

The main problem in using this simple 'hill speed ratio' is that many riders 'attack' short hills, and ride up them at much higher power than on the level. You might want to monitor your heart rate, to



make sure that your power level remains approximately constant.

3. EFFECT OF WIND ON OUT-AND-BACK LEVEL RIDING

When you ride out and back with a constant wind parallel to the road of velocity V_{hw}, it slows you in one direction, and speeds you in the other. Unfortunately you spend more time going slowly than going fast, so the average speed drops somewhat.

There is a tendency to assume that a headwind simply subtracts its value from your speed, but this is not true. Normally you have to solve the cubic equation to find what actually happens. Thankfully, the main effect can be approximated.

The general rule is: $\Delta V/V_{fl} = (V_{hw}/V_{fl})^2/3$

On the left is the decrease in average speed, as a fraction of the no-wind riding speed V_{fl}.* On the right is the ratio of the wind's speed to the no-wind speed, squared and divided by three. For example, if the wind's speed is half the rider's no-wind speed, that wind will reduce average out-and-back velocity by $(0.5^2)/3$ or 0.08, that is 8%.

When the rider is unfaired, a steady crosswind causes just half the average-speed reduction found for a parallel wind.

4. EFFECT OF MASS ON SPRINTING

Now for another effect of rider-plus-bicycle weight: sprinting. How far ahead would you be at the end of a sprint if you could shed one percent of your total mass? There is a tendency to assume that the gain would be one percent of the sprint distance, but this is never the case. A longer sprint involves more time at a virtually constant speed (top speed), when there can be little effect of mass because acceleration has virtually ended. So the gain in distance approaches a fixed value.

For this situation the power equation must be modified, by adding an acceleration term $(\Delta V/\Delta t)*M_{ef}$ inside the square brackets. (Effective mass M_{ef} is just slightly greater than M, by approximately the mass of two tires and two rims. This difference will be ignored here.)

This is a much harder problem to solve, because it involves a differential equation which must be integrated through hundreds or thousands of time steps. Of course this can be done by computer, but such results are in no way general.

The general rule is:

Distance gained = $(\Delta M/M)*\{V_{ex}*t-L\}$

On the right is $(\Delta M/M)$, your fractional or percentage decrease in mass or weight. The quantity in curly brackets is a *fictitious distance*, computed from sprinting time t, velocity reached (when exiting the sprinting zone) V_{ex}, and sprint distance L. The constant-power sprint can be assumed to start either at rest or at any initial velocity—the value of initial velocity does not enter the expression directly. Fictitious distance is always less than the actual sprint distance, typically about a quarter or third the value in the case of one or two hundred meters.

In words: the distance gained in a t-second sprint is the percent decrease in mass, times the difference between the sprint

** This was derived assuming no rolling resistance, in which case $V_{fl} = V_{nr}$. The inaccuracy introduced when rolling resistance is present has not been investigated.*

distance, and the distance that could have been covered in t seconds at the final velocity V_{ex}.

The trouble is, many of us have never had a constant-power sprint timed and measured. So a graph has been provided, to illustrate the concept. It was constructed to depict one slightly heavy individual, with system mass M=100 kg, aerodynamic-drag factor D=0.2 kg/m, and C_{rr}=0.003.

The main curves reflect sprinting at constant power, starting at rest. They show how speed rises as more distance is covered, and are marked with time since the start. But most sprints do not start from a dead stop. With this graph, you can pick an initial speed, and determine elapsed time and elapsed distance as velocity increases.

Consider the rider travelling at 22 mph (9.9 m/s), with a steady power output of 223 W. Power is suddenly raised to 600 W for a long sprint of 300 m. Enter the 600 W curve at 9.9 m/s, which gives a starting distance of 68 m and a starting time of 10 s. Adding 300 m for a final distance of 368 m reveals a final time of 34.3 s (elapsed time 24.3), and a final speed of 13.6 m/s (30.2 mph). The 'fictitious distance' is then 13.6∗24.3–300 = 30.5 m. Reducing system mass by 1% (1 kg) will move the rider forward by 305 mm, or almost exactly one foot—about 0.1% of the total sprint distance.

When I consider reducing the weight of my bicycle, I put it in the context of how much I care about gaining a distance advantage like this. If I was routinely losing by more, there'd hardly be any point.

Note that the graph can also be used to estimate power output for those individuals whom it 'fits'. Consider a rider travelling at 7 m/s (15.5 mph) in an 80" gear (6.38 m per crank revolution). Suppose that ten crank revolutions at maximum effort brings the speed up to 12 m/s or 26.7 mph. The graph is used by seeking the proper power curve, which shows a speed increase from 7 to 12 m/s, over the elapsed distance of 64 m. I marked the speed increase and distance increase on tracing paper, and concluded that they would have fallen on a 920-W curve, if such had been present. (A final speed of 25.8 mph would have indicated 800 W, and 27.3 mph would have implied 1000 W.)

5. IMPROVEMENT IN LEVEL SPEED PREDICTED FROM INCREASE IN COASTING SPEED

If you find a long, constant-slope hill, your steady-state coasting speed is just such that your total drag equals the hill slope S times your weight. (Beware—achieving steady state is slow. A small deviation from the true steady-state speed is reduced by about 63% only after coasting a distance of M/(2∗D), or about 200 m. You may have to pedal to the expected coasting speed, then watch your speedometer to see if you guessed correctly.)

One trouble with this experiment is that it requires measuring the hill's slope. Another is that your total drag force , alone, is not enough information to determine your aerodynamic drag factor.

Something simple you can do, however, is to coast the same hill with two different riding positions. (Or two different bikes, as long as the wheels are switched to keep rolling resistance constant!) If one has less aerodynamic drag, you will coast faster. It might take an afternoon of trials (good training!), but if there is no wind, you ought to be able to establish the speed advantage quantitatively.

That information can be used to infer the improvement in aerodynamic drag factor: $-\Delta D/D = 2 * \Delta V_{cs}/V_{cs}$
(Here, V_{cs} is steady coasting speed on a downslope. Delta V_{cs} refers to the difference in speed for the two cases.)

The percentage decrease in the drag coefficient is twice the observed percentage increase in speed on the slope.

In turn, the decrease in drag factor can be used to predict the improvement to V_{fl}, your speed on the flat:
$\Delta V_{fl}/V_{fl} = (-\Delta D/D) / [3+(W*C_{rr})/(D*V_{fl}^2)]$
$= 2*(\Delta V_{cs}/V_{cs}) / [3 +(W*C_{rr})/(D*V_{fl}^2)]$

As shown in the first section, the second term in the square brackets varies between 0.14 and 0.29 at high speeds for unfaired and faired bicycles, respectively. Ignoring this small contribution for simplicity, we can say that the percentage speed increase on the level, expected due to reduced aerodynamic drag, is approximately 2/3 the percentage speed increase noted in coasting down any slope.

6. ESTIMATING THE AERODYNAMIC DRAG FACTOR, D

D is almost always important to HPV speed. Apart from measuring it accurately with coast-down instrumentation or a wind tunnel, how can you get a rough idea of its value for your size, riding position, and typical clothing?

Find a very level stretch of road several hundred meters long. You also need to find a time (like dawn) when the air is calm—wind will badly upset the results. Finally, you need a speedometer which responds quickly, and measures in tenths of mph or kph.

Mark off a test distance, 50 m or 100 m. (Once you've calibrated your gearing, you can do this in terms of a certain number of pedal revolutions. Also, telephone poles in my neighborhood are 51.5 m apart.) The essence of the test will be to coast through that measured distance, starting at 9–11 m/s (20–25 mph), and note the speed at the beginning (V_{in}) and the end (V_{ex}). Then the average drag force in Newtons is $M*V_{av}*(V_{in}-V_{ex})/L$, where M is the total mass in kg, the velocities are given in m/s (note that m/s is mph ∗ 0.45), and L is the distance in m. V_{av} is defined as $(V_{in}+V_{ex})/2$.

Where you proceed from there depends on the accuracy you seek. If you do this test multiple times, in both directions, at roughly the same initial speed, you may be able to cancel the effect of slope, and average the effects of bumps and wind.

At high speeds, at which the drag force is almost all aerodynamic, you could approximate D as F/V_{av}^2, or $M*(V_{in}-V_{ex})/(L*V_{av})$. Coasting at various initial speeds should give roughly the same value for D.

If you wished to try your hand at determining values for D, C_{rr}, and S, you would plot the computed force versus V_{av}^2 for each of your trials. *Ideally* the points would fall on two straight, parallel lines. The slope of the lines would be D. The average of the two intercepts would be $M*G*C_{rr}$. The difference of the two intercepts would be M∗G∗2∗S. (But I would be very surprised if real-world conditions allowed you to succeed at this! You might well find a large cloud of

Continued from page 12

data points, which hint at something physically impossible, such as negative rolling resistance….)

BIOGRAPHICAL NOTE

Jim Papadopoulos <papadopoulos@alum.mit.edu> is a registered engineer who has spent much of his life trying to understand bicycles. He has a Ph.D. in the areas of mechanics and materials, and works in R&D at Rexnord, a manufacturer of power transmission hardware in Milwaukee, Wisconsin. In previous years he co-founded the Cornell University Bicycle Research Project, offered bicycle engineering services as BICYCLE R&D, and started the internet mailing list Hardcore Bicycle Science. He has been engineering editor for the journal Cycling Science, and chair of the ASTM task group on handlebar testing. He has patents on single-pass computerized wheel truing, and on a regenerative brake for bicycles.

Minical: an introductory spreadsheet for determination of power use while cycling

by John Tetz

Quite often we hear generalized statements being made concerning issues such as the amount of power required to climb a hill, or determining the amount of power an individual is capable of delivering or the difference in power demand of adding or subtracting a weight on a vehicle, or determining aerodynamic drag (C_dA) and rolling resistance (C_{rr}) and many other interesting matters. With the resources available today we can be more specific than are general statements.

The now ubiquitous personal computer allows an individual to leave behind mere generalizations and guesses in favor of more specific and objective answers to the power question. Spreadsheet programs, which merchants often provide as part of the software package of a new computer, manipulate equations with the same ease as the more familiar word processors manipulate text.

The simple spreadsheet layout described herein, Mini Calculator or MiniCal, introduces the first of a series of increasingly sophisticated spreadsheets which Jo-el Sanders, John Snyder and I developed to explore a range of different cycling situations. The tentative names of some give a hint as to their function: Power, Gearing, %Power Distribution; Power Comparitor; Coastdown Calculator; C_dA Estimator. These will be made available to the public as completed digital files via the HPVA web site, and other sources as demand so warrants.

Spreadsheets are only as good as the quality of their formulation. The formulations used in the series are meticulously detailed in "A Primer on Bicycle Mechanics, with a Spreadsheet for Power Calculations" by Joël Sanders, published as a separate monograph and available, at cost, through the HPVA. The primer explains the expressions according to "first principles." It is recommended reading for anyone who wishes to study the empirical basis of these spreadsheets.

The primer's primary audience represents the "artisan" community—those who design and build human-powered vehicles, yet who do not possess a formal engineering background. The primer, along with MiniCal and the rest of the series, provides these individuals with access to a cycling-focused treatment of mechanical theory via a versatile computational instrument, such that they may better understand and quantify their efforts. The spreadsheet as an integral part of any tool collection enables a designer to isolate effects, to study patterns, and to predict outcomes with reasonable accuracy, greatly lessening the trial-and-error phase demanded by unguided experimentation. MiniCal's intended audience also includes avid cyclists who wish to begin exploring the relationships of inter-related parameters, thus increasing their riding skill and greatly expanding what they can accomplish aboard a bicycle, and to make more meaningful comparisons to different vehicles and riders.

MAKING A MINICAL SPREADSHEET

The following will describe how an individual may build MiniCal from scratch. The spreadsheet layouts have been designed around user inputs which are a set of variables defining the bicycle, the environmental conditions, and the intended or observed riding technique used. The equations in turn reveal the corresponding power value in either horsepower or watts.

The velocity, grades, weights and other user inputs depicted in the example will serve to confirm the accuracy of your replication. Later, new values will be used to illustrate how the spreadsheet can be utilized. It is assumed that the reader will posses some familiarity with the spreadsheet program that he or she chooses to use. If you do not have sufficient experience it pays to find a friend to help with the initial set up. Building your own spreadsheet gives the advantage of a deeper understanding of the workings and operations.

MiniCal may be generated in either of two basic versions: feet per sec units (U.S.) or SI metric units. The authors used the spreadsheet application Microsoft Excel™ to create these instructions. Other computer programs will function in a similar manner, although the user should consult relevant documentation for her/his specific application.

Start by opening a spreadsheet program and checking to see if you have up to L columns. If not you can change the % size in the zoom control in the right side of the tool bar. Cell letters represent horizontal rows, whereas numbers represent columns. Cells may contain text, numerical values, or mathematical equations as entered by the user. Start in cell A1 and type in MINICAL in 14 pt bold text. This will be the name of the spreadsheet.

The sections below provide the cell address (A3:, B3:, etc.) followed by the information typed into the cells. These are titles for which I generally capitalize the first letter and use **bold** to differentiate these labels from other cells. Titles will be in 10-pt. text size. Our first row will be row 3. The same titles are used for both U.S. and metric.

U.S. and Metric

A3: CdA	G3: Total Wt
B3: Crr	H3: Wind
C3: Eff	I3: Grade +/–
D3: Rider Wt	J3: Velocity
E3: Vehicle Wt	K3: Power
F3: Cargo Wt	L3: Power (used only for U.S. version)

UNITS

Now is the time to make a choice to configure MiniCal as either in U.S. or in S.I. metric units. Type in the words, or letters and symbols, exactly as they appear below, including the parentheses, excluding the cell location and colon as before.

U.S.

A4: (C_d x ft^2)	G4: (lbm)
B4: (no units)	H4: (mph)
C4: (percent)	I4: (percent)
D4: (lbm)	J4: (mph)
E4: (lbm)	K4: (hp)
F4: (lbm)	L4: (watts)

Metric

A4: (C_d x m^2)	G4: (kg)
B4: (no units)	H4: (km/h)
C4: (percent)	I4: (percent)
D4: (kg)	J4: (km/h)
E4: (kg)	L4: (watts)
F4: (kg)	

Note: (lbm) denotes pounds-mass to differentiate it from pounds-force. The coefficient of rolling resistance is a ratio and is thus dimensionless.

INPUT PARAMETERS

For the moment type in the user inputs as they appear below. Set the number of decimal places to appear the same as shown if possible (adjust by clicking on the decimal buttons on Excel's tool bar to increase or decrease). You may also wish to use the underline feature to differentiate user inputs from calculated values. Notice in the following listing certain cells do not have a suggested input. These cells (G5, K5 and I5) will be filled in later with formulas.

U.S.		**Metric**	
A5: 2.4	G5:	A5: .223	G5:
B5: .0070	H5: 2.0	B5: .0070	H5: 3.2
C5: 95	I5: 2.0	C5: 95	I5: 2.0
D5: 160	J5: 10.0	D5: 72.6	J5:16.2
E5: 27	K5:	E5: 12.3	K5:
F5: 5.0	I5:	F5: 2.27	

FORMULAS

Now begin to carefully enter the formulas. The first will add the Vehicle, Rider and Cargo weights to provide Total Weight.

U.S. and Metric

G5: =D5+E5+F5

In Microsoft Excel™ an equals (=) sign signals to the program that an equation follows. After typing in the formula press the "enter" key. A value of 192 lbm (87.1 kg) should appear in cell G5. If not, recheck your entries ensuring they have been typed exactly as shown above. The formula will appear in the tool bar if you click on cell G5. There you may correct any errors.

In cell K5 you will enter the Power equation which consists of the sum of separate formulas representing Gravity, Aerodynamic Drag, and Rolling Resistance, divided by Drive-Train Efficiency. (More sophisticated formulations in the series account for additional terms including; air temperature and pressure, wheel weight, acceleration, etc.) MiniCal's power equation appears below as a group of three lines, but it must be typed in as one continuous string (no carriage returns or spaces).

U.S. formula for cell K5:

```
= ((2.667*10^(–3)*G5*J5*I5/100) +
(6.67*10^(–6) *A5*(J5+H5)^2*J5) +
(2.667*10^(–3) *B5*G5*J5))/(C5/100)
```

Cell L5:

L5 =K5*746

SI metric formula for cell K5:

```
= ((2.706*G5*J5*I5/100)+
(1.247*10^(–2)*A5*(J5+H5)^2*J5)+
(2.706*B5*G5*J5))/C5/100)
```

Press the "enter" key.

If everything has been entered correctly the power output will read 0.170 hp and/or 127 watts. Caution: after the spreadsheet is checked to be working properly make a copy because if in the future you make a mistake by typing something in a formula cell you will erase that formula.

EXPLORING

One of the most useful features, of a computerized spreadsheet like MiniCal is an ability to explore an array of numerical patterns with "drop and drag". Drop and drag allows the computer user to repeat the spreadsheet layout numerous times without having to re-type the entries.

Click on cell A5. While hovering over the cell the cursor appears as a white "plus" sign but when moved to the lower right corner of the cell it will change to a dark + sign. With the cursor appearing as a dark plus sign hold down the mouse button while dragging across to the last cell to the right, L5, and release the mouse button. All the cells in row 5 should now be selected as indicated by a darkened background. Without touching the mouse buttons move the cursor to the lower right corner of L5 until the cursor changes into a dark cross. Click and drag down at least ten rows. Drag down farther if you so wish. All of the data and formulas will have been repeated ten times or more.

Select cell J5: velocity. Change its present value to 1. Move down one cell to J6 typing in the formula. =J5+1. Press the "enter" key. The value "2" will appear. Once again select J6. Using the black + cursor, click the mouse while drawing the cursor down the row from J6 to J16. Release the button: you will observe the displayed velocity increasing in 1 mph (or km/h) increments in each sequential row, with the corresponding changes in power appearing in column "L". You can do this operation to all user-defined non-formula cells such as weights, grade, wind, C_dA and C_{rr}. You may also insert single values in user-defined cells to compare the parameter sets of different bicycles or different riding conditions. Enjoy exploring!

POWER CAPABILITY

An exciting prospect in conjunction with MiniCal's ability to quantify power is the capability of almost any bicycle to have the potential of functioning as an ergometer. With our ability to calculate power over a wide range of speeds and grades we have the first step in finding your power capability. The second step is in locating a suitable hill of uniform slope and steepness—enough that you can raise your effort near your normal higher-level hill-climbing ability. You could also climb at your more typical effort to learn what that power level is. So look for a rather steep hill (5%–6%) long enough to get a good workout.

Suggestion: because you have to climb this hill several times you may want to use this same hill to determine your C_dA value as you go back down. To compare the aerodynamic efficiency of your vehicle against others it is necessary only to compare effective frontal area (C_dA), which is the product of the aerodynamic drag coefficient (C_d) and the frontal area (A). You can determine your actual C_dA by using our Coasting Calculator. The Coaster also is used to determine your C_{rr} (rolling resistance) on flat ground at low speeds.

I happen to have a hill that has a continuous climb of 67 m in 1125 m, which is a 6% grade (I measured the elevation gain on a geodetic map). On a map look for elevation lines that are evenly spaced meaning the grade is reasonably uniform. Four-lane center-divided highways generally have more uniform grades (and wider shoulders). Try to get an accurate value for percentage grade because this will affect your results. The more accurately you set up the test the more accurate your results can become.

Now comes the physical work. Do your tests on a relatively calm day. You could put in a wind condition ([–] for wind coming behind—or from passing cars) but wind is generally not steady. Also our calculators are set up for wind in line with the vehicle travel. You should make your calibration run as "pure" as you can. Later you can play with wind values on your spreadsheet and see what that does.

Do a warm-up ride, then go to the bottom of the hill, zero your odometer/bike/computer/timer. Ideally this computer should read in tenths of km/h or mph and have a manual start-stop button. A heart-rate monitor would be handy but not absolutely necessary. Now take a breath and start climbing. Try to climb at a uniformly hard rate and when you arrive at a known stopping (distance/altitude) point write down the time, distance, average speed. Do at least three to four runs or until your body says "enough". Go home. Have a cool drink and get out your spreadsheet programs.

At this point you will have to estimate your C_dA and C_{rr} (see table 1). C_dA and C_{rr} values have a small impact for this test if the grade is steep and the speed is low. Then insert your test values in the proper cells and read your power. For instance, for four runs on a 6% grade, with an average heart rate of 140 beats per min with a peak of 154 beats (close to my max) my average speed is 6.5 mph. This converts to 0.233 hp (176 watts) with other user inputs of, E_{ff} 95, C_dA 2.4, C_{rr} 0.007, Bike 27 lbm, rider 160 lbm.

By climbing more slowly my more typical power level is 0.150 hp, 110 watts. (If these power levels seems very low wait till you get to my age!). If you type in a 3-mph (4.9 km/h) speed into MiniCal it will show a surprisingly low power level. Do a few more runs on different days to see how uniform your data are and also to calibrate your internal sensation of what various power levels feels like.

Table 1. Representative parameter values

Parameter	Value (ft^2)	Value (m^2)	Ref	Comment
(C_dA)	6.0	0.56	2	Upright roadster
	4.3	0.40		Touring (arms straight)
	3.4	0.32		Racing (full crouch)
	2.9	0.27		Open recumbent
	1.9	0.18		Racing (draft position)
	1.4	0.13		Typical faired recumbent
	0.5	0.046		Streamlined faired recumbent
C_r	0.005–0.010		1	1-1/4-inch tire; good road
	0.002–0.007			Various tires; smooth surface
	0.002–0.005		3	Various tires; smooth surface
η	0.95		1	~2% chain, 1% bearings, 2% limbs

OTHER INVESTIGATIONS

As a way to understand the effects of different variables, reduce the grade to zero, increase the speed to 15 mph and change the value of C_dA to zero. For my bike the value of 0.055 hp is the power required to overcome a C_{rr} of 0.007. Then reduce C_{rr} to zero and insert C_dA of 2.4 and you will see that C_dA and C_{rr} are about equal at the surprising speed of 15 mph. As C_dA drops to 1.0 the speed climbs to 23 mph for them to be equal. So you can see the contribution of these two terms. Some of our spreadsheets include an automatic percentage power distribution of gravity, air resistance, rolling resistance, acceleration, efficiency, pedal power, and power to the wheel.

LIMITATIONS AND STRENGTHS

The accuracy of any calculation is only as good as the quality of its input values. How well defined the inputs are depends on such external factors as: one's ability to determine grade, the accuracy of weight scales, the calibration of the cycle computer used to determine velocity, the accuracy of reported wind speed, etc. Additionally MiniCal must be recognized as an entry-level tool. Many potentially defined variables such as air density, wind direction, and nuances in the calculation of grade were omitted for the sake of brevity. In the same spirit the tables accompanying this article are to be regarded as guidelines.

CONCLUSION

The MiniCal is simple enough to build from scratch allowing a user to become acquainted with the formulation and operation of computerized spreadsheets. It is compact enough to be used in such bicycle-portable computer platforms as personal digital assistants and palmtops.

Other spreadsheet layouts in the series will serve as a set of tools which give a user the ability to determine customized parameter sets for her/his individual vehicle. Quantifiable differences among vehicles are finally possible.

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3. Whitt, Frank Rowland, and David Gordon Wilson. *Bicycling Science*, 2nd ed. ISBN 026273060X, MIT Press, 1982.
4. HPVA web site: www.ihpva.org/hpva/

John Tetz <jgtetz@worldnet.att.net> is an engineer who retired after 38 years at Bell Laboratories, and who is having the time of his life. He writes that he just cannot stay off his bikes.

TECH NOTES

OPTIMUM BODIES FOR BICYCLISTS

by Mark Drela and Doug Milliken

Mark Drela wrote: Jim Papadopoulos gave some data on the physiology of bicyclists and the speeds they attained. I tried to correlate speed with power/drag-area. I define an average diameter "d" by representing the rider as a circular cylinder of the same mass and height. This then defines a body aspect ratio a = h/d and a frontal area A = hd. If all riders are geometrically similar, then A should be proportional to the frontal and drag areas, and P/A should be proportional to V^3.

When I plot P/A versus V^3 from the columns below, I don't get a straight line. It's more like a meatball! This implies that rider shape is a critical parameter. This is evident in the last column C_D', which is a modified drag coefficient based on the equivalent area rather than the true frontal area. C_D' should be a constant for all riders of the same shape. Clearly, shape matters.

Looks like rider A had rather streamlined parents.

—*Mark Drela (drela@orville.mit.edu)*

Doug Milliken commented, with regard to Mark Drela's statement, "Looks like rider A had rather streamlined parents."

Or maybe he/she spent some time in the wind tunnel, working on position. Small adjustments in riding position can make relatively large changes in drag.

$M = \rho_b\, h\, d^2\ \pi/4$ (body density: $\rho_b = 1000\ kg/m^3$)

$A = hd = \sqrt{4Mh/\pi\rho_b}$

$C_D' = (P/A)/(0.5\ \rho V^3)$ (air density: $\rho = 1.225\ kg/m^3$)

Rider	t (min)	M (kg)	h (cm)	LT (l/min)	P(1hr) (W)	a	A (m^2)	P/A (W/m^2)	V^3 (m^3/s^3)	C_D'
A	51.0	79.8	1.8700	4.40	376.	8.022	0.1900	1978.9	2233.7	1.4465
B	52.5	71.3	1.8150	4.14	359.	8.115	0.1648	2178.8	2047.6	1.7373
C	52.5	74.8	1.8450	4.40	363.	8.121	0.1757	2065.8	2047.6	1.6472
D	54.0	80.7	1.8300	3.94	357.	7.723	0.1880	1898.6	1881.7	1.6473
E	54.0	70.2	1.7400	3.71	336.	7.677	0.1555	2160.4	1881.7	1.8745
F	54.0	77.1	1.8500	4.08	360.	8.031	0.1816	1982.3	1881.7	1.7199
G	55.5	62.4	1.7480	3.50	307.	8.199	0.1389	2210.6	1733.2	2.0823
H	55.5	66.4	1.7400	3.93	331.	7.894	0.1471	2250.1	1733.2	2.1196
I	56.0	73.2	1.7750	3.77	325.	7.746	0.1654	1964.6	1687.2	1.9011
J	57.0	73.9	1.7850	3.95	335.	7.775	0.1680	1994.6	1599.9	2.0354
K	59.0	81.2	1.8380	4.23	336.	7.750	0.1900	1768.2	1442.7	2.0010
L	59.0	69.0	1.7330	3.84	326.	7.697	0.1523	2141.2	1442.7	2.4232
M	61.0	75.7	1.8080	3.60	313.	7.831	0.1743	1796.1	1305.4	2.2465
N	59.0	62.1	1.7300	3.34	300.	8.092	0.1368	2193.2	1442.7	2.4820
O	65.0	62.0	1.6850	3.08	256.	7.785	0.1330	1924.6	1078.9	2.9124

John Cobb (Bicycle Sports) has been part of the Texas A & M tests run by Steve Hed (Hed Wheels) for the last 10+ years. John has a great deal of experience working in the tunnel with different athletes, and has established a real "eye" for putting riders in positions for low drag, without any major changes in rider power output. There are a lot of subtle changes that can make significant differences, in the context of racing.

—*Doug Milliken*
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[This was extracted, with permission, from the HBS mail list. For subscription information on the HBS (Hardcore bicycle science) list, see: www.sheldonbrown.com/hbs.html
For frequently-asked questions, see: www.sheldonbrown.com/glossary.html]

AERODYNAMIC ADVANTAGE FROM USING FEWER SPOKES

by Mark Drela

Ted Bennett writes:

"My question is two-part: is there any significant aerodynamic gain in using fewer spokes, especially on front wheels which are in the slipstream, unfaired by forks or the rider's body?"

The spoke drag should be very nearly proportional to the total frontal area of all the spokes. If you hold the spoke stress fixed, then the frontal area varies as the square-root of the number of spokes. So going from 36 2-mm spokes to nine 4-mm spokes should theoretically cut the spoke drag in half. Other considerations obviously preclude using nine spokes.

Theoretically, the effective drag of a spoke rotating in a wheel is roughly the same as if the spoke were held fixed at right angles to the airflow, as shown below. Assuming a coefficient of drag, C_d, of 1.2 for a cylinder, the C_dA of 36 spokes is about 0.025 m^2. The actual C_dA will be considerably less than this, since the spokes draft the rim and each other to various extents depending on the crosswind.

[Jim Papadopoulos points out that there isn't much drafting effect from one spoke to another because they are not close enough: see the technical note on interference drag by Jim and Mark in issue 46.]

The whole bike + rider in a crouch has $C_dA = 0.3\ m^2$ (from *Bicycling Science*, Whitt & Wilson), so the worst-case spoke drag for one wheel is about 0.025/0.3 = 8% of the total for the bike. Maybe 3-5% might be the actual number—I can't say.

The rest of this technical note gives the reasoning behind the statement on the theoretical effective drag of a spoke. The wind resistance force on a spoke is mostly at right angles to the spoke, since this is the direction that pressure forces must act regardless of the wind direction. Only friction forces can act along the length of the spoke, and these are quite small by comparison. The drag coefficients are, approximately,

$C_d = 1.0$ for round cylinder in subcritical flow at a Reynolds number of about 3000.

and

$C_d = 0.044$ for very thin 3-mm-long airfoils at 13.5 m/s (30 mph).

Mainly pressure forces act on the cylinder, and only friction forces act on the thin airfoil. The latter should be representative of the friction forces on the spoke. Furthermore, the pressure forces depend only on the flow velocity component normal to the spoke—the flow component along the spoke has little influence. This is why jets have swept wings: the wing airfoils think they are flying at the Mach number times the cosine of the sweep angle, instead of at the full flight Mach.

The normal velocity component V_n seen by the spoke at some radius r (in a wheel of radius R) for bike speed V is

$V_n = V[r/R + \cos(\theta)]$ where θ is the angle of the spoke from the vertical.

The force on the spoke (diameter D) at some radius r, pointing mostly normal to the spoke, is therefore:

$dF = 0.5\ \rho\ [V_n]^2\ C_d\ D\ dr$

(ρ is the air density)

The power required to drive the spoke against this force is $dP = dF\ V_n$

Also,

dP = horizontal-force ∗ V + torque ∗ ω

which gives the same result. The bottom line is $dP = 0.5\ \rho\ |V_n|^3\ C_d\ D\ dr$

The quantity r/R + cos(θ) relating V and V_n is negative only for a circular region joining the ground-contact point to the hub, [the circle is the set of points where dP = 0] and is never less than –1. So leaving it out has little effect on the final result for P.

dP is easily integrated, first over 0..r..R to get the power of one spoke at angle theta, and then averaged over 0..θ..2π to determine its typical contribution over the entire wheel revolution.

2∗π∗P =

(∨ from 0 to R) (integral from 0 –2 π) 0.5 ρ $|V|^3$ [r/R + cos(θ)] C_d D dr dθ

This gives:

$P = 0.5\ \rho\ V^3\ C_d\ A$

where A = DR is the frontal area of spoke. Therefore the average power needed to drive a single spoke around on the wheel is the same as if it were simply held in the breeze at right angles to the flow (e.g., attached upright to the bike frame).

—*Mark Drela <drela@orville.mit.edu>*

[This was taken, with permission, from some correspondence on the HBS mail list. —Ed.]

CRANK-ARM LENGTH

by Danny Too

Since there has been quite a bit of interest and discussion about crank-arm lengths, and since I referred to my study in the last issue of *Human Power*, I thought I would provide some additional comments.

First, it is very difficult to generalize a specific crank length that would be optimal for everyone, and it is just as difficult to specify an exact crank length that would be optimal for any one person. The reason? There are many factors that would affect the "optimal" crank-arm length to maximize/optimize cycling performance, and there is a complex interaction among these factors. These factors include: height of the person, total leg length, thigh-length-to-leg-length ratio, seat-to-pedal distance, type of recumbent position, load/resistance/gear ratio used, pedalling rate, type of measurement (peak power, average power), training effect or familiarity with a specific crank-arm length, inter-individual variability, and intra-individual variability.

Second, it may not be the crank-arm length that is as important in determining cycling performance as the crank-arm length that optimizes the hip and knee joint angles (which then optimizes muscle lengths) to maximize force and torque to the pedals in cycling performance. For instance, the optimal crank-arm length for one individual 6 ft (1.80 m) tall (who has short legs but a long trunk) may not be optimal for another 6 ft individual (who has long legs but a short trunk). I would suspect that the joint angles (hip, knee, ankle) and joint kinematics during a pedal cycle between the two 6-ft individuals would be different and how/where force, torque, and power are produced (and during which part of the pedal cycle it is produced) would also be different and could have a major effect on cycling performance. By the same token, two 6-ft individuals with the same total leg lengths may have different ratios of the lengths of thigh to lower leg. One individual may have a very long thigh and a very short lower leg, and the other one may have a very short thigh and very long lower leg. Therefore, what is an optimal crank-arm length for one person may not be optimal for another (since the joint angles and kinematics of the hip and knee are probably more important variables to consider than crank-arm length).

Third, on the assumption that a certain crank-arm length is found that maximizes cycling performance on a recumbent for an individual (with a certain height, total leg length, thigh-to-lower-leg-length ratio, etc.), would this crank-arm length also maximize cycling performance for a second individual having the same anthropometric characteristics? The answer could be yes and no. It depends. If the second individual has had significant experience on a recumbent (while the first individual did not) and this second individual consistently trained and cycled with some given crank-arm length, then based on the principle of specificity of training, I would suspect performance to be better with the crank-arm length that the second individual had been training with. Using the "optimal" crank-arm length (found for the first individual) by the second individual would, initially, result in a decrement in performance before there is an adaptation and training effect (and an increase in performance that may surpass that the other crank-arm length originally trained with).

Fourth, there are also inter-individual variability and intra-individual variability in cycling performance. Individuals (even elite cyclists) will vary in performance from one day to another (intra-individual variability) and some individuals will have greater day-to-day variability than others (inter-individual variability). A certain crank-arm length that maximizes performance on one day may result in an increased or decreased performance on another day. Multiple trials would be needed to determine what would be considered to be an average performance value for that particular crank-arm length. This is on the assumption that there is no longer a learning curve or training effect due to repeated trials. The exact same procedure would have be repeated for different crank-arm lengths and statistical analysis undertaken to determine if differences in average performance between different crank-arm lengths are attributed to chance (random performance variability from a day-to-day basis), or truly to differences in crank-arm length. (Note: there are other factors to consider, such as randomizing the crank-arm length experimental conditions and trials). This is obviously a lengthy and tedious process since it becomes a research-oriented project with controls implemented to remove any confounding variable(s) that may affect the results. But controls of this sort would reveal (to the person willing to undertake this task) what is the crank-arm length that would maximize his/her performance for that particular test protocol. There are also other difficulties encountered, such as what crank-arm lengths to examine. Based on my research, 35-mm changes in crank-arm length will clearly (although not always statistically significantly) affect performance. Changes of this magnitude will significantly alter hip and knee angles. However, will there be a difference in cycling performance between cranks that differ by 10 mm (e.g., 170 vs. 180 mm) or 5 mm (e.g., 145 vs. 150 mm) or 2 mm or 1 mm? And will the differences be equally applicable to individuals of different heights and leg lengths? I don't know. There are a lot of questions, but very few answers because very few people are involved in this area of human-powered-vehicle research (especially for recumbents). In addition, if the test protocol involves a maximum-endurance test, or a test for an extended period of time, then motivation becomes a very large and important factor and could confound the results. This is the reason why I have used a 30-second all-out power test for my experiments (since it has been determined to be extremely simple, reliable, consistent, and a robust test with minimal intra-individual variability from day to day. But then the training effect has to be accounted for if the same individual is tested repeatedly over time). To illustrate the effect of motivation on performance, a simple example will be used. If you were to cycle the same route (e.g., 42 km, 26 miles) each day with maximal effort, but on some days you were chased by a fairly large dog for several miles at various parts of the route, I would predict faster/shorter average times during those days being chased (although maximal effort is given on all days). This information was provided by a friend of mine (a "regular" cyclist) who, with his friends, noted that their cycling times were significantly faster on days they were chased by farm dogs in open areas along their route.

Fifth, to add to the confusion, there is an interaction among crank-arm length, pedalling rate, and load/resistance gear ratio. This would suggest that there may not be one optimal crank-arm length that maximizes power production, but several (depending on the load and pedalling rate selected). The optimal crank-arm length to maximize power would be dependent on the load and pedalling rate, and could be one where the maximum pedalling rate is maintained with the largest load that can be applied (without a decrement in maximum pedalling rate). Based on the muscle force-velocity-power relationship, for a given power output, optimal pedalling rate would decrease with increasing load (with a constant crank-arm length). This would suggest that increased loads to maximize power, resulting in a decreased pedal rate, would favour longer crank-arm lengths. For a given power output, it is possible that a shorter crank-arm length with a higher pedalling rate and lower resistance would be equally effective when compared to a longer crank-arm length pedalling at a lower rate (but higher resistance). I have collected data examining these interactions, but have not had the opportunity to crunch and analyze them. I have been too busy making revisions to reviewers' comments to a manuscript submitted to the Journal of Sports Sciences (on how changes in crank-arm length affect power production in upright-cycle ergometry. With the same load, it appears the effects on power production are different in recumbent-cycle ergometry. But this is a paper that is currently in review for publication in *Ergonomics*).

To conclude, and to perhaps, shed some light on the "optimal" crank-arm length to maximize power, I will provide a brief summary of my paper currently in review. In that paper, a recumbent position was used to test crank-arm lengths of 110, 145, 180, 235, and 265 mm. Nineteen untrained males (most have never cycled a recumbent) were each tested in all five crank-arm-length conditions according to a different randomized test sequence for each subject. The test was a 30-second all-out power test with a load of 85 gm/kg of body mass. The results on power production showed a parabolic (inverted U-shape) curve with increment in crank-arm lengths from 110 to 265 mm. Peak power (highest power produced in any five-second interval during the 30-second test) was found with the 145-mm crank. The largest mean power (average power produced for the entire 30-second test) was found with the 180-mm crank, and the largest minimum power (power produced during the last five seconds of the test) was found with the 230-mm crank. What does all this mean? It means that there is an interaction among crank-arm length, load, pedalling rate, and power output. For the load selected (85 g/kg of body mass), a crank-arm length of 145 mm (based on the five cranks used in this study) with a pedalling rate of approximately 170 rpm will produce the largest power output. However, as fatigue sets in during the latter part of the test (especially during the last five seconds), the pedalling rate decreases to somewhere between 82 rpm (for the 110-mm crank) and 93 (for the 230-mm crank). This decrement in pedal rate with the load selected favours the use of a longer crank (230 mm for this study). The 180-mm crank happened to be most advantageous if the power production (mean power) over the entire 30-second test was considered. Would the results have been different if a significantly greater or lesser load/resistance was used (or if trained recumbent cyclists were used)? I would think so. Care must be take in interpreting the results of any study. It should be noted that the average leg lengths of the subjects were: total leg length measured from the greater trochanter to the floor was 941 mm; upper leg length measure from the greater trochanter to the knee center of rotation was 409 mm, and the lower leg length was 534 mm. Not all subjects showed the same parabolic trend in power production with increments in crank-arm length from 110 to 265 mm. This is attributed to intra- and inter-individual variability and the a training effect with repeated testing. However, this training effect was accounted for in the research design of the study where each subject was tested with a different crank-arm length test sequence. Finally, this study was a power test for 30 seconds and not an extended aerobic study. Again, I have other data sets collected, but have not had the time to analyze them.

To provide information about cranks that is a little bit more substantial, I have run regression analyses on the data set for that particular study. The results from the regression equations obtained predict that peak power (5-second interval) would be obtained with 124-mm cranks, highest average power (for 30 seconds) would be obtained with 175-mm cranks, and the largest minimum power (last 5 seconds in an 30-second all-out power test) would be obtained with a 215-mm crank.

One final caveat. These predicted crank-arm lengths are limited to the subjects, recumbent position, and test protocol used in that particular experiment. It is not necessarily true or applicable for experienced recumbent cyclists or their particular recumbent position.

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A TANDEM RECUMBENT DESIGN
by Charles Brown

I'd like to share some thoughts with you on the design of recumbent tandems. Reliability tends to be a problem on tandems, whether upright or recumbent, because they often take parts which were designed for single bikes and impose on them twice the load.

Wheel strength varies with size, so a wheel that's half the diameter needs approximately half the spokes for equivalent strength. 20" (500 mm) rims and hubs are readily available with 36-hole drilling; these should provide a sturdy foundation for your steed. Further, the drive wheel(s) could be built up without dish, the off-center rims being compensated for by moving both dropouts 1/3" (10 mm) to the right. This gives even more strength without added weight, just from using the materials more intelligently.

Drive-train troubles caused by having the power of two applied to parts made for one could be reduced if the front rider powered the front wheel, and the rear rider powered the back. Independent pedalling cadence comes automatically with such a design. This may incur a slight weight gain, but you get the advantage of not having to transfer the front rider's power seven feet (2 m) to the back wheel, with the attendant losses of power.

The two-stage drive train often used in fixed-boom front-wheel-drive designs would gear up the front wheel nicely. A good design is one made from an old back hub. A cog is attached to one of the hub flanges, and a chain runs down from this to a single sprocket on the front wheel. The 'power' or highly-tensioned side of the chain should be nearly parallel to the steering axis. One end or the other of this chain can be a little closer to the steering axis than the other, but looking at it from the top, the chain must be going straight out from the steering axis. This is so that the forces from pedalling do not try to turn the steering a little bit to the right or left with each pedal stroke, which forces must be resisted by the rider's hands on the handlebars.

A freewheel or cassette would be attached to this upper hub, allowing gear changes, and a second chain would run forwards from this to the crankset. Proper positioning of the intermediate gear unit is essential, and the builder might want to allow for some fore-and-aft adjustment of it. This would allow fine-tuning of the chain line of the final drive if someone wants to change the gearing there. The mountings for this intermediate drive must be made very strong and rigid.

Gearing up the 20" (500-mm) rear wheel is more problematical. A rear cassette with an 11-tooth top cog combined with a 60-tooth big chain ring would give a 110-inch top gear. Alternatively, an internally-geared hub could step up the gearing.

Bicycles are controlled by a combination of balancing and steering; it helps the captain pilot the machine if she or he has firm control over the balance. A stoker moving around unexpectedly can make a tandem hard to manage. To improve this, the captain's center of gravity should be up higher than the stoker's, so the person steering the craft also has more leverage, and thus more control, over the balance. This is particularly important if the lighter person does the steering.

In the sketch I've placed the captain's seat over the stoker's pedals to make the bike more compact: other arrangements are of course possible. Note that on tandem recumbents with a 20" front wheel under the captain's knees, like this one, a 20" back wheel makes for a better front-to-rear balance than the more usual 20" front-, 27" rear-wheel sizes, improving the ride and handling. In the drawing, adjustment for different-size riders is by moving the rear seat forwards and back, and by moving the front pair of pedals fore and aft. This will probably require changing the length of the chain, but the captain's will probably not require adjustment very often.

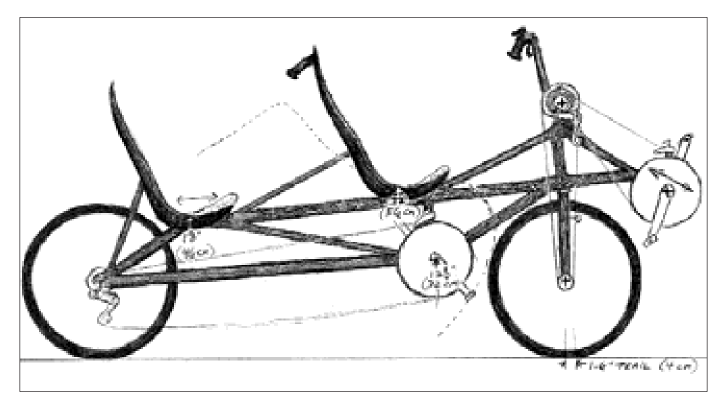

Figure 1. Side view. Drawings provided by author.

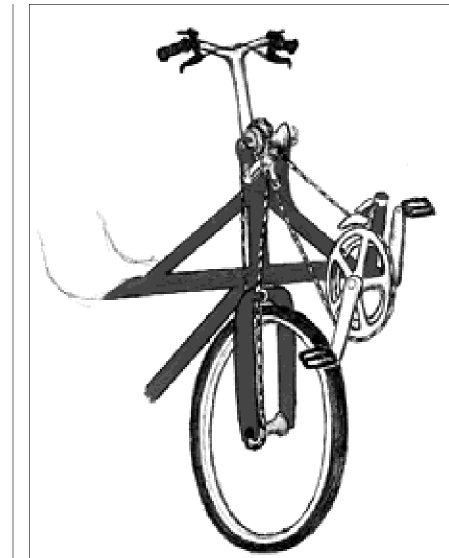

Figure 2. Front details.

—Charles Brown
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[Editor's note: chain management takes on critical significance in front-wheel-drive systems. There must be no chance that the chain could come off and lock the front wheel. Dave Wilson.]

CRASHWORTHINESS ANALYSIS OF ULTRALIGHT METAL STRUCTURES
by Sigit P. Santosa

Abstract of a doctoral thesis presented at MIT, May 1999.

In the design of lightweight crashworthy metal structures, thin-walled prismatic components have been widely used in aircraft, high-speed trains, fast ships, and automobiles. Two new types of such components are proposed, both of which consist of a thin-walled member and an ultralight metal core such as an aluminum

honeycomb or a closed-cell aluminum foam. The first type is the thin-walled member filled with the ultralight metal core, while the second type is a double-walled member with the ultralight metal core sandwiched between the two walls. This research is [aimed at studying] the crushing behavior of the ultralight metal core and at determining the crashworthiness and the weight saving of structures composed of the two types of reinforced components, each of which may undergo axial crushing, bending, or twisting. Numerical expressions are developed to predict the crushing behavior of the new type of components.

The first task of the research is to study the crushing behavior of closed-cell aluminum foams. A new model of a truncated cube, which captures the basic folding mechanism of an array of cells, is developed. The model consists of a system of collapsing cruciform and pyramidal sections. Theoretical analysis is based on energy considerations in conjunction with the minimum postulate in plasticity. The assumed kinematic model for the crushing mechanism of the truncated cube cell gives a good agreement with the deformation mechanism obtained from the numerical simulation. Analytical formulation for the crushing resistance of the truncated cube cell is shown to correlate very accurately with the numerical results. Closed-form solutions for crushing resistance of closed-cell aluminum foam in terms of relative density are developed. The formulas are compared with the experimental results and give an excellent agreement.

The double-walled sandwich columns appear to have the highest crushing resistance. It is found that during progressive crushing, debonding of the core-face plate is localized only at the corner portion of the column, while the web portion remains intact and dissipates most of the external work. In the case of filled columns, a significant increase of the mean crushing force is also obtained by filling the thin-walled columns with aluminum foam. It is found that the increase of the mean crushing force of a foam-filled column has a linear dependence with the foam compressive resistance and cross-sectional area of the column with a proportionality constant equal to 1.8. The proposed solution is well correlated with the experimental data for wide range of column geometries, materials, and foam strengths.

The ultralight metal filler also provides a higher bending resistance by retarding inward fold formation at the compression flange. In the case of aluminum-foam filling, the presence of the foam filler changes the crushing mode from single stationary fold to a multiple propagating fold. The progressive crush prevents the drop in load-carrying capacity due to sectional collapse. This phenomenon is captured from both experiment and numerical simulation. It was found that partially foam-filled beams still offer high bending resistance, and the concept of the effective foam length is developed.

Two distinct crushing states are observed in the torsional deformation of filled thin-walled bars, namely the initial torsional resistance and the stabilized torsional crushing mechanisms. The ultralight metal filler provides a stabilizing effect on the torsional crushing process. The inward fold collapse of the column wall is restricted due to the filler, and the plastic resisting mechanism is increased through the formation of outward-diagonal shear bands.

The structural weight saving is assessed through a concept of specific energy absorption, defined as the external work absorbed divided by the total component weight. For the same total component weight, the filled and double-walled sandwich members give higher specific energy absorption compared to the empty thin-walled column. A 40–60 % weight saving can be achieved by the double-walled sandwich components, while 25–45 % weight saving can be achieved by filled thin-walled components. It has been proven that the proposed components are attractive structural elements for weight-efficient crashworthy design.

Doctoral Committee:

Prof. L. Anand (Chairman)
Prof. F. A. McClintock (member)
Prof. T. Wierzbicki (Thesis advisor)

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OPTIMUM PILOT FOR A HUMAN-POWERED HELICOPTER

by Mark Drela

Jim Papadopoulos writes: "The tangible result is that, just as for bicycles where power and aero drag figure prominently, there is a technically definable approach to defining the 'best' pilot (assuming helicopter weight depends on pilot weight).

If you have an envelope of 'best athlete' weight and anaerobic power numbers, then your best point on that curve will satisfy a slope relation based on the power and system weight at that point."

Yes, but it's interesting that once the vehicle is built, the criterion for the "best athlete" changes totally.

For the case of an already-built flying vehicle, the minimum cruise-power/weight required from the powerplant occurs when the powerplant is twice the empty vehicle weight. This is true for any vehicle which uses only dynamic lift against a fluid (i.e., no buoyancy or road contact).

For the case of a conceptual flying vehicle being designed, the lightest pilot is best because the airframe weight decreases faster than the pilot weight due to the cube-square law.

For buoyancy or road vehicles, the picture is very different. A given bike or rowing shell wants the largest-possible rider to "dilute" the vehicle's weight and drag. The shell's air (water) drag does go up with larger weight, but much more slowly than for an airplane or helicopter.

I ran into these issues firsthand in the MIT hydrofoil project in the early 1990's. Big riders could motor along pretty well on the pontoons, but had a very hard time coming up on the wings. But this was relatively easy for small riders. This was not surprising, since 41 kg (90 lb.) was the ideal pilot weight for cruising the 20-kg (45 lb.) hydrofoil.

—Mark Drela <drela@orville.mit.ed>

TRANSMISSION EFFICIENCIES

by Dave Wilson

Keen readers will have noticed that I have a fascination with transmission efficiencies. In *Human Power* (vol. 13 no. 1, p. 17), I reviewed a paper by Ron Shepherd, stating that the efficiency of chain transmission is normally over 99%, and in

the following issue (no. 44, 13/2) I reported some of his comments in a response to a letter I wrote him. Angus Cameron's paper on measuring drive-train efficiency was published in issue 46: his static experiments led him to believe that derailleur efficiencies in a new six-speed system ranged from 92.1 to 98.4%, and the efficiencies of a three-speed hub gear had a low of 87.3 and a high of 97.9 over the same power range (50–400W). A direct one-speed chain drive had efficiencies from 96.0 to 99.0%, the higher values occurring at the 400W power level. (These predictions are, I believe, based on an assumption that the losses change little as the power is increased.)

In this note I am giving results obtained and reported by others, in most cases in the Hardcore Bicycle Science (HBS) mail list started by Jim Papadopoulos and now moderated by Sheldon Brown. I am not going to attempt to interpret them or to comment on areas of agreement or disagreement, except to re-state the need for further work. There are, nevertheless, valuable data reported here.

Giles Puckett <giles@research.canon.com.au> reported Angus Cameron's efficiency figures for a seven-speed hub, a Shimano Nexus.

"Columns 1 and 2 represent the bare hub without the roller brake attached. Column 3 includes the brake.

"The input load on the sprocket was about 110 N, using an 11.0-kg weight, which represents a recreational rider producing about 75 W (1 hup) at 60 rpm or 125 W (1.7 hup) at 100 rpm. I'll try other values in the future but I doubt it will make much difference.

gear	ratio	#1	#2	#3
1	0.632	0.91	0.91	0.91
2	0.741	0.94	0.93	0.92
3	0.843	0.87	0.87	0.86
4	0.989	0.86	0.89	0.89
5	1.145	0.86	0.87	0.87
6	1.335	0.92	0.93	0.93
7	1.545	0.91	0.91	0.91

"Some observations include this: that gears 3, 4, and 5 showed lowest efficiencies. Presumably the drive is passing through both sets of planetaries. Guess which gears I use most often? :(

"And again for some other hubs, courtesy of Jan Verhoeven, the following are for a Shimano Deore LX derailleur system, with 28" wheels."

Shimano Deore LX derailleur system, 28" wheels

No. of teeth front	rear	Efficiency (%) 200 watt	100 watt	Metres per crank revolution
22	28	98.5	99	1.7
22	24	98	98	2.0
22	21	98.5	96	2.2
22	18	96.5	96	2.7
32	21	95	93.5	3.1
32	18	94.5	93.5	3.8
32	16	94	94	4.2
32	14	93.5	94.5	5
42	16	93	93	5.7
42	14	91.5	91.5	6.6
42	12	91.5	89.5	7.6
42	11	91.5	88	8.3

Sachs elan 12-speed hub gear, 28" wheels

Gear	Efficiency (%) @ 200 watt	@ 100 watt	Metres per revolution
1	92	87	2.2
2	95	91	2.7
3	96.5	92.5	3.2
4	91	90	3.8
5	91	90	4.3
6	90.5	90.5	4.8
7	88	88	5.1
8	88	88	5.7
9	88	88	6.1
10	86	85.5	6.6
11	87	86.5	7.1
12	88	86	8.5

All numbers, except the gear numbers, have been estimated visually.

Shimano inter 7-speed hub gear, 28" wheels

Gear	Efficiency % @ 200 watt	@ 100 watt	Crank revolution
1	90	87	2.9
2	91.5	90	3.3
3	>> 75	76	3.8
4	87	86.5	4.4
5	83	82	5.2
6	92	92	6.0
7	91	91	7.0

—Giles Puckett

ML Bruce <adlc@jbic.com> reported that he had posted the following results from some testing he did with a Sachs Super 7 hub.

"In short, the speed lost in going to the 7-speed hub from a derailleur system was 2–3%. For those of you counting seconds on short rides or minutes on long rides, that few percent can be significant. We did two experiment sets using our Linear 'Co-pilot' tandem.

"#1. 240-m hill climb with 32-kg (70-lb) daughter as a non-pedalling stoker heart rate 160 to 170 bpm

Sachs 7-speed hub: av. speed = 6.96 mph 4% relative standard deviation Suntour derailleur: av. speed = 7.19 mph 4% relative standard deviation

"The Sachs hub is about 3% slower with a 4% RSD, thus the difference is not statistically significant.

#2. 4.5-km flat loop with 25-kg daughter as a non-pedalling stoker, heart rate 150 to 160 bpm

Sachs 7-speed hub: av. speed = 14.1 mph, 0.7% relative standard deviation

Suntour derailleur: av. speed = 14.4 mph, 0.5% relative standard deviation

"The Sachs hub is about 2% slower with a 0.6% RSD, thus the difference is statistically significant.

"The speed loss attributable to the Sachs 7-speed hub is small under these conditions and not a problem for our intended touring."

—Mark Bruce

Andy MacGee contributed a note to the HBS mailing list on the friction occurring when bicycle chains pass from one sprocket to another that is not perfectly aligned.

"Someone mentioned a test run by Cyclo-Pedia (in the 17th edition of its catalog, 1993) to evaluate drive-train friction resulting from "bad" chain alignment. Because the writeup is fairly short, I thought the verbatim text might clear up some questions.

"We set up a bike with an eight-speed freewheel with 16-tooth cogs in every position. We drove the crankset with an electric motor which was connected with a watt-meter. We installed felt brakepads and a screw adjustment on the brake. We even installed three chainwheels of the same number of teeth (34) so that we could try the chain at all angles.

"The test was started, the motor allowed to warm up, and the brake was adjusted to provide drag on the rim. The warm-up lasted for five minutes. We then shifted the chain to each of the possible chain positions (24 in all). At each position we recorded the number of watts the motor required. The recording procedure was done six times and results were averaged and are provided in the chart below. All components were new at the start of the test: after break-in some of these watt figures may decline. However a couple of our highest readings were taken in the last test series.

Chainwheels

	A	B	C
1	20.0	21.0	23.0
2	18.8	19.9	20.3
3	17.7	18.6	19.7
4	16.5	17.0	18.5
5	16.7	16.2	17.3
6	17.4	16.3	16.5
7	19.3	17.3	16.6
8	22.6	18.9	17.7

"We feel that this simple test shows that people who use 8-speed drive cogs are paying a severe penalty for the added gears that they get. Taking the lowest watt reading (16.2) and dividing it into the highest (23.0) you will find that the drive-train friction is taking a considerable amount of added energy (23.0/16.2= 1.42). In our test, using the outermost chainwheel and innermost cog costs the rider over 40% in added drive-train friction. It really does pay to keep that drive chain asaligned as possible."

"The illustration shows that cog 1 and chainwheel "A" are closest to the bicycle centerline. Presumably the best alignment is between the middle chainwheel and drive-cog number 5.

"Some of my questions about this test are: how consistent is friction from a screw-adjusted felt pad? –how scattered are the data from all six tests?"

—Andy MacGee
<phillip_the_ambidexterous@yahoo.com>

I asked Jim Papadopoulos to comment on chain losses: "I would like to see losses 'resolved' into (1) those on the drive side of the chain loop; (2) those on the non-drive side of the chain loop (mostly for derailleur losses); and (3) the hub losses (mostly for internal gearing).

"I wish there were a facility with a proven, calibrated efficiency tester so that we could be certain of the accuracy of the losses and efficiencies reported above!"

—Jim Papadopoulos
<papadopoulos@alum.mit.edu>

–Dave Wilson <dgwilson@mediaone.net>

REVIEWS

Pedalling for a living

Chasing Rickshaws
by Tony Wheeler and Richard l'Anson
Lonely Planet Publications, Hawthorn, Australia. 1998. 191 pp.
ISBN 0 86442 640 2

Accompanying photos are from the book, by permission of Lonely Planet.

I was unprepared for the reaction of the visiting foreign colleague whom I once surprised with a pedicab tour of Oslo. She was captivated by the unconventional tour of the city, but also commented that she had resolved never to take human-powered transportation again after a visit to Nepal. There she had felt like a Great White Exploiter, sitting passively while being pedalled around by poor, thin Asians. In my delight at how environmentally friendly my new hobby was in Oslo, my thoughts had passed lightly over the status of full-time colleagues in Asia.

Are pedicabs a form of exploitation, or are they an ideal form of transportation—a decent job for those who earn their living transporting others, and a low-pollution, low-noise means of moving people and goods through the city? Are they an environmental boon, or a health hazard for the operators? Many of my fellow pedicab chauffeurs in Europe and North America would join me in characterizing a well-run pedicab business as good, honorable work that also is good for the environment. What about in Asia, where pedicabs are more firmly established and much more often provide serious transportation? A definitive answer is not to be found in the painstakingly researched and beautifully photographed *Chasing Rickshaws*, but between its two large covers is a wonderfully detailed and vibrant visual record of pedicab transportation in twelve Asian cities, and here there are many clues.

Variations on four rickshaw designs

While it's not clear how author Tony Wheeler and photographer Richard l'Anson selected the places visited, the twelve cities represent quite a range of pedicab design and use. The delta trike, with passengers behind the driver, is used in Agra (India), Beijing, Dhaka (Bangladesh), and Macau (a Portuguese province over which China will take control in 1999). The reverse design, with passengers between two wheels in front

Nino Quilon's full stereo sidecar in Manila also has a built-in light show. The equipment weighs as much as a passenger and makes the vehicle more costly to rent, but the wired sidecars attract more business.

of the driver, is found in Hanoi, Penang (Malaysia), and Yogyakarta (Indonesia).

Ordinary bicycles furnished with sidecars fill the pedicab niche in Manila, Rangoon, and Singapore. The original jinrickshaw, basically a horse cart with a human runner replacing the horse, is also still used, both in Hong Kong and Calcutta.

Within these four major pedicab types, each city has its own distinctive model, which usually has remained unchanged for decades. Schematic drawings of each vehicle in profile and top view convey the most important lines of the design, while accompanying tables give information about size and weight, in both British Imperial and French (metric) units, plus an estimate of the number of such pedicabs in the city.

All the pedicabs portrayed are "rather horrible examples of design," says Wheeler, with excessive weight one common fault. Most range from 90 to 125 kg. While Hong Kong's pull rickshaws are the lightest vehicle documented (60 kg), their 500-mm-wide seats seem too narrow for anything more than one adult. The Sai Kaas (a corruption of "side cars") of Rangoon are the lightest pedalled pedicab (73 kg with the bike), but with one passenger seat facing forward and one facing backward, they weigh significantly less per passenger than their Hong Kong cousins.

While heavy, these machines are generally very solidly constructed. This robustness, together with the ingenuity of the rickshaw mechanics, means that an individual pedicab can last many decades. While this longevity sounds like an environmentalist's ideal, the very opposite of planned obsolescence, it also is probably one factor discouraging design improvements that could make them more friendly to the driver, like lower vehicle weight. "The new pedicabs I build have to compete for sales with every other pedicab that has ever been constructed," commented Steve Meier, of Main Street Pedicabs in Denver, Colorado. So it is in Asia, as well. Even in cities where tens of thousands of pedicabs are operated, no single manufacturer puts out more than some dozens of new vehicles each year. It is easy to see how the smallness of the market for new vehicles, together with a risk aversion surely merited where a single design has dominated for decades, could work together to keep a lid on innovation.

The pedicabs' capabilities vary greatly with the design. The Singapore sidecar cannot take more than one adult, whether of European or Asian dimensions. A striking photo shows thirteen pedicabs in a row on a street in Singapore, each of the drivers highly visible in a yellow shirt, and but one passenger in each. Judging from the selection of photos for this book, at least, it seems that the delta trishaws of Agra are the real pack horses. They are shown chock full with kids in school uniforms and festooned with hanging school bags, or full with a 2.5-m-high load of metal containers, securely tied down.

The social impact of pedicabs

"Socialism can arrive only on a bicycle," predicted José Antonio Viera-Gallo, Assistant Secretary of Justice in the government of Salvador Allende.[1]

"Beijing's tricycles are less than elegant, especially with their ragged winter hoods. The Qianmen Gate at the southern end of Tiananmen Square is in the background."

But what if the bicycle (or tricycle) is pedalled by someone else? The social and economic lot of the rickshaw operator varies considerably from place to place, as we see clearly in the portraits that Wheeler and l'Anson provide.

"With more than a third of a million rickshaws filling the streets of Dhaka, traffic jams like this one in Old Dhaka are commonplace."

Explaining why 95% of his colleagues are bachelors, 54-year-old Penang rickshaw operator David Kok says that it is too dangerous an occupation for him to get married. In Dhaka, in a crowded "rickshaw dormitory," all the floor space is used for sleeping; everything else in the operators' lives must go on outside. Breaking into a rare editorial comment about the operators, Wheeler says of Calcutta, "[I]f there is a job anywhere on earth which reeks of exploitation and indignity it must be pulling a Calcutta rickshaw."

On the other hand, tourists and the night life provide a large part of the income for the 300 trishaws in Singapore, and it seems that pay there is better. As in North America and Europe, some of the Singapore operators are university students, financing their studies. The riders in Macau cater exclusively to tourists, taking $20 an hour for a tour; they are less than aggressive about drumming up business, however, so it is hard to know how many of these $20 hours they actually work each week. Beijing trishaw operators apparently get good pay, and generally view their jobs as better than working in a factory.

In the end , it seems difficult to make generalizations about the level of exploitation of pedicab operators. The lot of those who make their living by pedalling may be more closely tied to the general condition of workers in a given society than it is to the trade itself. It seems that even where socialism has arrived on its bicycle, there is room for some people to make their living by pedalling others around.

I would have liked to know more about the function of the pedicabs in the cities portrayed. My passengers in Oslo are mostly people who hop aboard on a whim, treating the pedicab as an amusement-park ride appearing in the middle of the city, or, occasionally, those who pre-order the pedicab for ceremonies like weddings. But what is the niche that pedicabs fill in the twelve cities Wheeler and l'Anson portray? It is clear that the tourist trade dominates the business in Hong Kong and Macau, and that the pedicab is a school bus in Agra and Penang, but a lot more information about the pedicabs as practical transformation would be interesting. What other sorts of transportation are pedicabs' chief competition, and how do they compare in price, comfort, and speed? Where pedicabs are used as practical transportation, why do the passengers choose to pay to ride instead of using their own bicycles?

Whatever the details of the passenger profile, it is clear that there is quite a demand for exactly the services that rickshaws provide. This is brought to life both by all the photos—I especially like the ones of broad streets caught in rickshaw gridlock—and the interesting history of the rickshaw at the end of the book. The many stories of regulations hostile to pedicabs show that they often have to be

"A pint-sized rickshaw wallah takes a well-earned rickshaw rest." (Dhaka)

forced out of business before they disappear, and sometimes are resilient enough that the pedicabs outlive the regulations against them. (Although the history unfortunately ignores all use of pedicabs in Europe and the Americas, there, too, are many stories of motorized taxis and other interests colluding to keep the streets free of pedicabs.)

Wheeler and l'Anson set out partly "to record a fascinating means of transport and human activity before it disappeared," and also to create a book which "celebrates rickshaws." They do a lovely job of both recording and celebrating, but it's not at all clear that rickshaws are on the way to disappearing. Pedicabs can pop up in large numbers many places, both where demanded by economic necessity, as in wartime Poland and post-Soviet Cuba, and for other reasons, as in Berlin, Denver and many other wealthy cities. Many of the readers of this book could find themselves chasing rickshaws closer to home than they thought possible.

ENDNOTES

[1] Quoted in Illich, Ivan, *Energy and equity*, London: Calder & Boyars, 1974.

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LETTER

SUSPENSION SPECIFICATIONS

Discussion of wimpy recumbent suspension led me to a few observations and a couple of questions about recumbent suspension. Of course, I expect most people to disagree with at least one thing that I've written, so feel free to comment and correct me. No flames, please—I'm a sensitive new age kinda guy, and my feelings hurt easily.

Observation #1 – Front suspension is a "must have" feature on a SWB. This is because there's a lot more weight on the front wheel than on a LWB or even a CLWB, and the small-diameter front wheel is more susceptible to road-surface irregularities (nice term for potholes, eh?). Because you steer with the front wheel (we'll leave rear-wheel-steered vehicles out of this for now), any jolts to the front wheel do more than just affect your comfort—they also affect your stability. At 50 mph (22 m/s), I want the front wheel to be able to hit a bump and keep on going in the same direction as before. With a rigid fork, a good jolt can knock the wheel sideways.

Observation #2 – Rear suspension isn't a big deal. I used to think rear suspension (in addition to front suspension) was a "must have" feature on the "ultimate" recumbent (I'm still waiting for someone to build the ultimate recumbent, but that's a whole other thread). What got me thinking this way was my switch from a no-suspension to a full-suspension MTB. For off-road riding (and riding on really poorly-maintained paved roads), I would still consider full suspension a "must have" feature, but for recumbent riding on "typical" paved roads, I would now downgrade rear suspension to a "nice if it's done right" category. The reason for the downgrade is that rear suspension typically adds weight, complexity, maintenance requirements and cost, and when it's poorly designed it robs power, creates annoying bio-pacing, and generally ceases to be a net benefit. On a recumbent, a decent amount of rear "suspension" can be achieved with a well-designed seat and fat tires. For most conditions and most riders, this is probably sufficient, particularly given the overall high level of comfort on a recumbent. Don't get me wrong, though—I'd love to have rear suspension on my 'bent, but only if it worked well, didn't add much weight, and wasn't a maintenance headache. Which leads me to…

Observation #3 – Simple is good. One example of where this would apply is rear suspension. Rather than construct a MTB-style rear suspension unit with one or more pivots and a shock, which would be heavy, costly to manufacture, might create unwanted bio-pacing and other effects, and would require regular maintenance, an alternative would be passive rear suspension. This could be achieved with cantilevered chainstays, for example, or a compliant material such as titanium. This type of simple solution is well-suited to recumbents (as opposed to off-road MTBs), as the need for suspension is less, given that the surfaces over which a recumbent typically travels are reasonably smooth, and other features of the recumbent already provide some form of suspension. Why build MTB-style rear suspension for a non-MTB, when something much simpler will do?

Observation #4 – More suspension travel isn't necessarily better. Sure, there are recumbent applications where a lot of travel is desirable, but in many cases more travel doesn't equate with better suspension. Travel is marketing hype in the MTB world, especially for downhillers. For road riding, all you need is enough travel to soak up vibrations and small bumps. Unless you're riding through big potholes on a regular basis, you won't be using the extra travel 99.9% of the time. Why carry around the extra weight? The Ballistic fork on my SWB recumbent has only 25 mm of travel, and I have never felt the need for more, even when I've hit large potholes at speed. Sure, I feel the bump (!), but the suspension I have is enough to take the edge off and allow me to maintain control. As proof of this observation, consider upright bikes (just for a moment). The Rock Shox Ruby fork for road bikes has only 30 mm of travel, whereas most cross-country MTB forks have about 90 to 100 mm of travel. As an aside, most people for whom I've helped adjust their MTB suspension weren't using anywhere near all of their travel.

Question #1 – Why is anti-dive a desirable feature? I've never felt that dive was a problem on my MTB or my recumbent, both of which use telescoping forks. Given that simple is good, what justifies the additional complexity of an anti-dive fork as compared with a simple telescoping fork (telescoping legs or telescoping steerer/head tube)?

Question #2 – What are the power losses with rear suspension? For those who have rear suspension on their recumbents, I'd be interested to hear observations on this issue. I know I lose about 2% of my power in the drivetrain (assuming that derailleur gears are 98% efficient), and I'm sure some power gets lost in frame flex and seatback compression. How do power losses to rear suspension compare with these losses?

Question #3 – Would cantilevered titanium stays with a 20" (406-mm) wheel provide decent suspension and minimize power losses? What would be the potential for sideways flex (which is bad, as compared with vertical flex which is good) and torsional flex? My mental picture of the "ultimate" recumbent includes dual 20" wheels, a titanium frame and an Action-Tec/Headshock-style suspension fork.

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Note: More letters appear on pages 7 and 9.

EDITORIALS

HPVS, HEALTH AND SPINNING

by Dave Wilson

The hpv internet mail list has been diverted recently by self-congratulation because medical researchers are reporting that exercise conveys benefits on more and more conditions, mental and physical. Sage voices have interjected words of caution into the celebration, warning that exercise is not a magic bullet, even when taken in a comfortable recumbent position. Let me nail my colors to the mast: I am old and very fit and convinced that my daily commute of 26 km, including some steep grades, is to be credited, at least with keeping the heart, lungs and associated pipework in good shape.

I have also been self-satisfied that the recumbent position enabled one to impose a load of greater than my weight on my legs. The reason I was smug about this was a report in one of the medical newsletters that seem to multiply like rabbits to the effect that bicycling (of the "regular" sort) does not benefit sufferers from osteoporosis, or fragile bones. For this condition to be ameliorated, we were informed, weight-bearing exercise must be practised, and bicycling did not qualify. Well, I thought haughtily, I survived a considerable drop onto asphalt when a ladder I was using to fix the gutter over our second-floor windows slipped. I bent the ladder considerably and knocked myself out, but the fact that no bones broke showed that recumbent biking is better than regular biking because we can easily push with more than our body weight. To add to this paean of self-approval, I considered that there is something virtuous in not being a spinner, because people who pedal at high RPM, 90–170, produce high power with low pedal forces, whereas I am more in the cart-horse category and use low RPM and high pedal forces.

I was rash enough to mention this to Jim Papadopoulos, who was visiting because we are collaborating on the third edition of *Bicycling Science* (which will change from being by Whitt and Wilson to Wilson and Papadopoulos). In my opinion he is the foremost scientist-engineer working in, in fact totally dedicated to, the bicycle-human-power field today. He immediately put my hypothesis to the test, getting me to ride up the 20-percent grade of the hill we live on using a regular bike in a gear that just required me to stand up on the pedals. When I put my recumbent in the same gear I couldn't get up the hill: I was not in fact putting a force equal to my weight on the pedals.

So are recumbents no better than regular bikes in combatting osteoporosis? Is strong pedalling at slow RPM no better than spinning? Would experts please comment!

—Dave Wilson

DIFFERENT STROKES?

by Dave Larrington

Dear reader:

When the Mad Professor (a.k.a. David Gordon Wilson) asked me to contribute a guest editorial for this publication, not only did I have a horrible feeling that he had become so mad that he now had other madmen living in his beard, but I also worried that I would be unable to contribute anything worthwhile. He wants me to write something about non-circular drives? What do I know? Anyway, after enough head-scratching to make those sitting nearby worry that I had become infested with some sort of especially repellent parasite, I have produced the following. While it's not very technical, I hope it will be of some interest. Errors of fact are all mine.

Throughout the history of cycling, there have been many alternatives to the regular circular pedal motion. In spite of claiming many advantages, notably "increasing the length of the power stroke", few have been successful. John Kingsbury, my predecessor as editor of the British Human Power Club Newsletter, opines that the human engine, like an internal-combustion engine, requires an "exhaust stroke", and moreover that the duration of this exhaust stroke is of the order of six times that of the preceding "power stroke". Increasing the length of the power stroke will therefore not produce a gain in power for any significant period. (Misguided people are always trying to do this.)

Some studies, and as I'm neither an academic nor particularly technically adept, I can't cite references, have found the circular pedal motion to produce more power than non-circular ones, but I believe I'm right in saying that few of them have allowed the victim, or rather subject, sufficient time to acclimatise to a different pedal motion.

However, one case where a non-circular pedal movement may be of some net gain is that of the streamlined HPV. Constraining the rider's feet to move more horizontally than vertically means that the height of the front end of the vehicle can be reduced, to the considerable benefit of aerodynamic efficiency. One such system was re-invented for about the fifth time by Miles Kingsbury a few years ago, and was dubbed the "K-drive". This utilises an

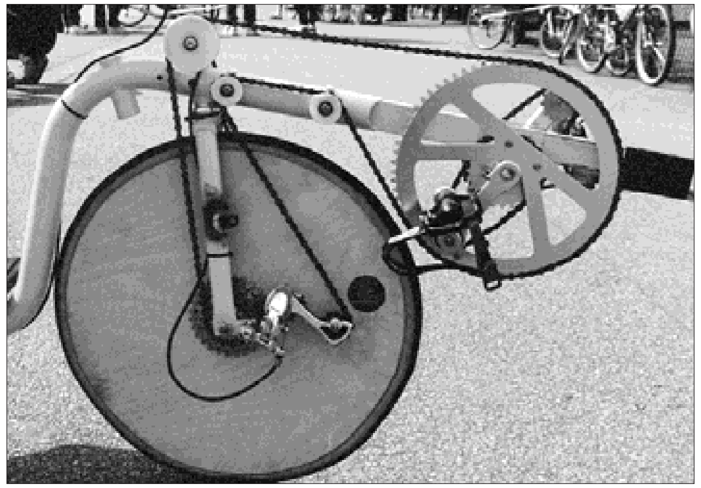

Figure. 1. K-drive-equipped Tchaikowski. Photographer unknown.

Figure 2. The K-Drive being worked on by Steve Slade and Miles Kingsbury.

arrangement of secondary cranks, sprockets and chains, with the net result that the feet move in an elliptical path, with the long axis horizontal (see fig. 1). The Kingsburys, father and son, had long been experimenting with non-circular drives, so their first move was to acquire a test subject, set him up on the trainer, measure his power output, and repeat over a prolonged period until he felt at home with the system. They found that once the rider had become acclimatised, although there *was* a small reduction in power output, it should be more than offset by the gains in aerodynamic efficiency the transmission would make possible. Great was the rejoicing in High Wycombe that day.

If I remember rightly, the K-drive transmission first appeared on the experimental Tchaikowski front-drive recumbent bicycles. Lessons learned from this application, and the transmission system itself, were rapidly transferred to the semi-production Wasp low racers (fig. 2). Initially there were a few bugs in the system, most notably the inability of various custom-machined parts to withstand the stresses imposed on them by Steve Donaldson (K-Drive tester-in-chief), but reliability was eventually achieved and the machines have dominated the faired class of the British Human Power Club's annual Championship race series ever since, culminating in 1998's clean sweep, with Steve Slade taking the title for the fifth successive year, from Roy MacDonald's and Steve Donaldson's similar machines. This in spite of the steel-framed bikes weighing in somewhere well above 20 kg.

As raced by the above-named three, the machines are FWD recumbent bicycles, with a seat height around 0.30 m (12"). They carry fibreglass nose and tail fairings, with a nylon fabric fairing enclosing the remainder, but leaving a fairly substantial gap at the bottom, around the front wheel. Thus the aerodynamics could fairly be described as sub-optimal, and on a *really* fast course, where manoeuvrability is less important that straight-line performance, they have sometimes suffered at the hands of fully-enclosed machines, such as the latest incarnation of Jonathan

Figure 3. Jonathan Woolrich in "Oscar" leads gNick Green in "Morse's Law".
Photo: Ian Chattington

Woolrich's "Oscar" (fig. 3), Nigel Leaper's tiny "Low" and Nigel Sleigh's "Plastic Maggot", ridden by Ian Chattington. Not to mention the time when Andy Wilkinson appeared with his multiple-road-record-breaking Windcheetah trike for a hilly 50-km road time trial and blew everyone into the weeds. Much the same applies in European competition, though here there is an additional problem with which to contend, namely gigantic Dutchmen.

The Plastic Maggot is perhaps worthy of a little further elaboration, as underneath, it too is a K-drive-equipped Wasp. The brainchild of Liverpudlian architect Nigel Sleigh, the machine's fairing is an angular but effective construction of the corrugated plastic known in the UK as "Correx" or "Corriboard" ("Coroplast" in the USA, "Corflute" in Australia), with a nose cone derived from part of a pilotless military target aircraft. Like most fully-enclosed bikes, it can be a bit of a handful in windy conditions, but has been unofficially clocked at speeds of more than 80 km/h (50 mph).

Of course, uppermost in the designers' minds when the K-drive was conceived was to use it in a *serious* streamliner - something lower, smaller and faster than the Yellow Bean and Bean II machines in which Pat Kinch set numerous records a few years back. The first fruit of this train of thought was the Beano (fig. 4), raced by Steve Slade at the 1995 World Championships in Lelystad, The Netherlands. It was quick alright, but…it turned out to be too small for intended pilot Pat Kinch, while its still-born little brother the Bambeano was found to be too small for *anybody*. The Beano ultimately found a new home in Switzerland, where Rosmarie Bühler has used it to set a women's one-hour record (as yet unratified) of 55.5 km. But for the past two-and-a-half years, observers have been eagerly awaiting the first appearance of the K2….

The chassis made its public debut in April 1997, another FWD low bike but this time running on ISO 406 wheels, shod with fat slick tyres. And full suspension!

Figure 4 (above). The Kingcycle Beano (left) and the Kingcycle Wasp, with (left to right) Meindert Valenteyn (Netherlands recumbent builder), Steve Slade, Steve Donaldson and Miles Kingsbury. Photo: Dave Cormie

Figure 5 (left). Morse's Law with gNick Green: A much-modified K-drive Tchaikowski combined with a cast-off Kingsbury fairing. Photo: Tina Larrington

Variations on the theme appeared intermittently during 1997 and '98, but one has yet, to the best of the writer's knowledge, to be united with its fairing. The shells, exquisite carbon-Kevlar mouldings, *have* been spotted, but not yet in public. And further delays may yet come from the collaboration between John Kingsbury and gNick Green (himself racing a fully-faired K-drive-equipped machine based around assorted ex-Kingsbury bits and pieces) to construct some lighter carbon-fibre frames. So we're still waiting.

In the meantime, we were at least hopeful that *someone* might come up with a better all-round vehicle/rider combination than Steve Slade and the Wasp.

gNick Green's "Morse's Law" (fig. 5) and its successor, the "Wooden Fish On Wheels" are usefully rapid machines, but lack the engine power to be winners, while Paul Davies' prosaically-named "Faired Bike" has a better motor, but probably more drag. Nigel Leaper's machine is rather more at home on tracks with easier corners than the venues used so far in the '99 season. And Oscar is taking a year off while Jonathan Woolrich concentrates on the arm-powered machine being campaigned successfully by Kevin Doran.

A full hard shell was on the cards for Roy MacDonald's machine, but I've heard that Roy has decided to stop racing and that both the "95% complete" plug and the bike itself are up for sale. Double unfaired champion Dave Richards has promised a fully-faired machine, but so far only the chassis has appeared. Yorkshireman Mike Weaver, whose Mikew 4 has been intermittently effective (due to Mike's inability to attend many of the race meetings), has a new machine on the drawing board for '99, but this too has yet to appear, while the Mikew 4, now owned by Ian Chattington, suffered a variety of problems on its debut which prevented it from even completing the race. In short, people are discovering just what a lengthy process building an HPV—whether "just" a fairing or a complete machine—can be.

Meanwhile, the Wasps roll on, with three wins to Steve Slade and three second places to Steve Donaldson. And the real high-performance non-circular-drive HPV is still confined to bed in High Wycombe….

Dave Larrington (fig. 6) is editor of the British Human Power Club Newsletter, and has been an HPV enthusiast since being part of the Imperial College team which set a Round-Britain HPV record in 1983.

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Figure 6. Dave Larrington, author, here racing "The Pink Fairy", a faired, standard Kingcycle. Dave is editor of the BHPC Newsletter. Photographer unknown.

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