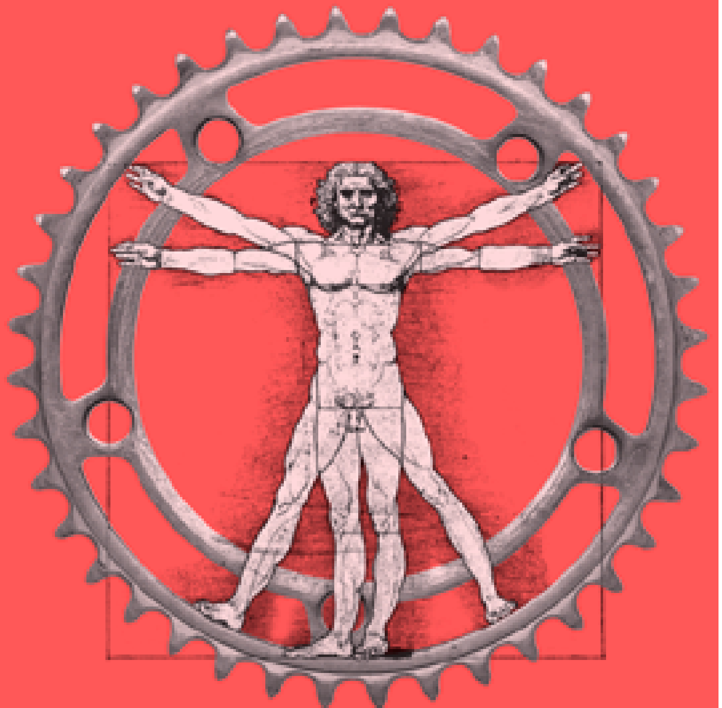

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In this issue

The mechanical efficiency of bicycle derailleur and hub-gear transmissions

Chet Kyle and Frank Berto have given us a long-awaited and very valuable report on a precise study of the efficiencies of a wide range of bicycle transmissions. It is both quantitative and well discussed. One intriguing conclusion is that, in general, hub gears have efficiencies about a couple of points lower than do derailleurs. However, hub gears that were "run in" and lubricated with light oil rather than grease showed efficiencies almost up to the derailleur level. As the authors state, one arrives at more (interesting) questions.

Technical notes

There is a better way than rolling

Detlev Tschentscher is following the pioneering work of John Dick, who made "Springwalker", in studying and building human-powered "exoskeletons" that promise to make walking faster and possibly capable of surmounting higher obstacles and rougher ground.

Further experiments on run-flat stability after front-tire deflation

Dave Wilson reports further experiments that seem to confirm (though on the basis of only two tests) that a good tight fit of tire to rim is also vital to provide run-flat stability and control.

Tire-rim compatability

John Stegmann relates, humorously at times, his adventures in manufacturing rims and in coping with the tendency of tires to creep and to allow the tubes to pop out and explode. He found that a good tight fit is important.

Control of hydrofoils using dynamic water pressure

Most (all?) HP hydrofoils have had their angle of attack controlled through a surface skimmer attached to a linkage. Al Taig has developed a lower-drag and cleaner alternative: using the impact (pitot) pressure picked up on the leading edge of the strut supporting the foil from the hull and controlling the attack angle from, e.g., a bellows.

Project review

CHicK-2000 project team "Active Gals"

Mark Drela reviews the report and videotape of a remarkable Japanese team that has achieved record performances with a talented woman pilot and an innovative plane. The wing uses a stressed-skin construction, allowing the main spar to be an I-beam and producing a ". . . wing-tip deflection [that is] amazingly small considering its low empty weight of 31 kg and its immense wing aspect ratio of 44."

Book review

Richard's 21st century bicycle book(s), by Richard Ballantine.

Your editor reviews two versions of the same book by Richard Ballantine: one in British English for the UK-European market, and one in American English for the North Americans. He gives two thumbs up.

Letters

Comments by Matt Weaver and John Stegmann on a paper in *Human Power* 51 on crank-arm length on recumbents, and responses by author Danny Too.

Editorials

Marek Utkin writes a guest editorial from Poland on aspects of the HPV scene there.

Your editor reviews some discussions on the future of HUMAN POWER.

Contributions to *Human Power*

The editor and associate editors (you may choose with whom to correspond) welcome contributions to *Human Power*. They should be of long-term technical interest. News and similar items should go to *HPV News* or to your local equivalent. Contributions should be understandable by any English-speaker in any part of the world: units should be in S.I. (with local units optional), and the use of local expressions such as "two-by-fours" should either be avoided or explained. Ask the editor for the contributor's guide (available in paper, e-mail and PDF formats). Many contributions are sent out for review by specialists. Alas! We cannot pay for contributions. Contributions include papers, articles, technical notes, reviews and letters. We welcome all types of contributions from IHPVA members and from nonmembers.

The mechanical efficiency of bicycle derailleur and hub-gear transmissions

Chester R. Kyle, Ph.D.
Frank Berto

Introduction

Since human power provides the propulsion for a bicycle, losses in mechanical energy are far more important than if purely mechanical or electrical power is used.

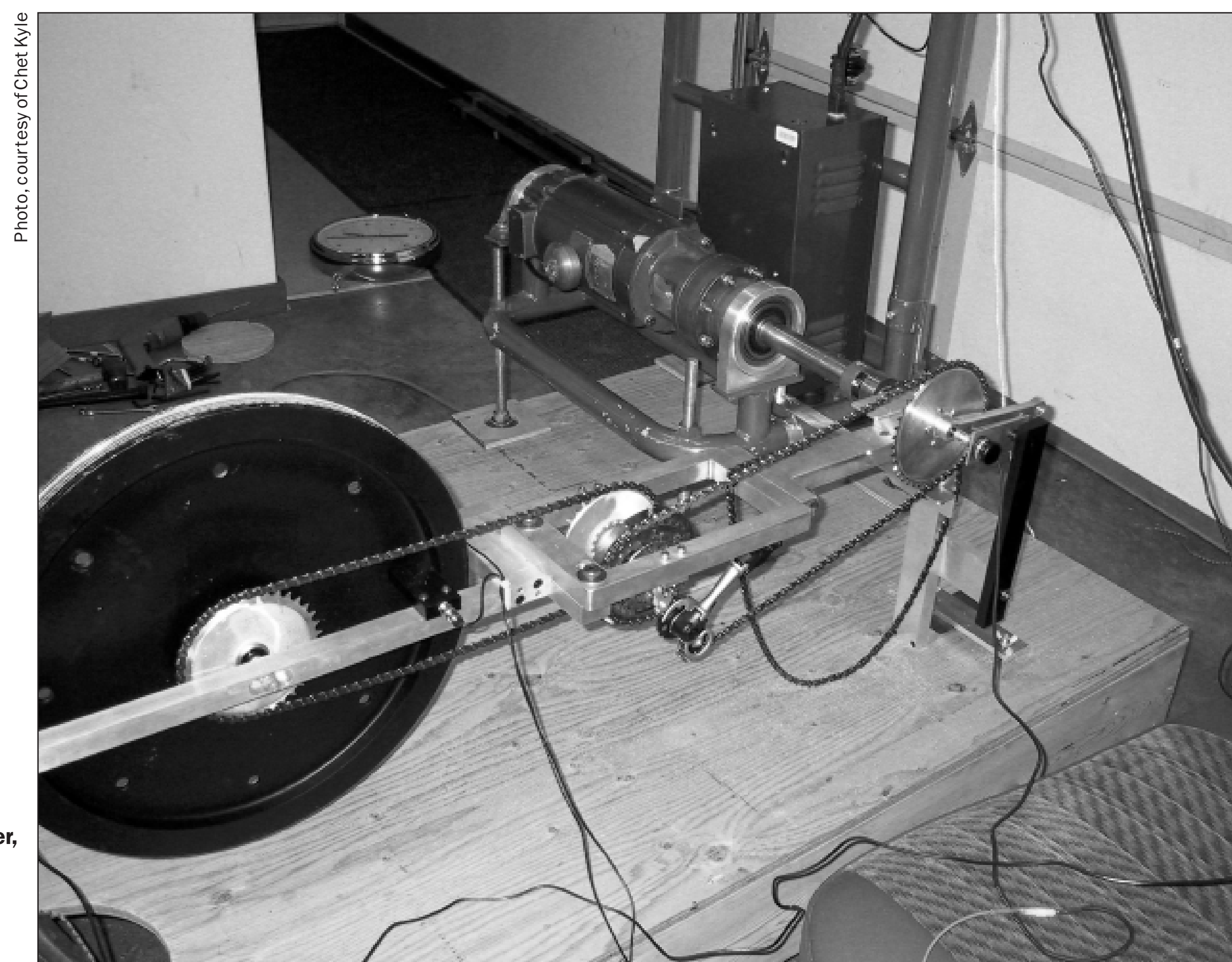
The mechanical efficiency of a drive system is defined as the ratio of the power output to the power input in percent. Typically, automotive drive systems are from 80% to 99% efficient [1], meaning that from 1% to 20% of the energy input is lost in friction. A well-oiled straight chain-and-sprocket bicycle drive can be as high as 99% efficient [2]. With other types of bicycle transmissions, however, the range in efficiency can be similar to an automobile, that is from 80% to 99% [5–11]. In a bicycle, small losses can mean large performance differences—especially in competition [3, 4].

For example, suppose Christopher Boardman, the present holder of the bicycle world hour record (56.375 km; Manchester, England, 1996), were to use a bicycle with a drive that lost 2% more energy than his record machine. Boardman would travel almost 0.5 km less in one hour [3]. The hour record has been broken several times in the past 30 years by less than 0.5 km. If an Olympic 4000-meter pursuit team were to use bicycles that were 2% less efficient, they would be about 2 seconds slower in the 4000-meter team-pursuit race, which would have moved them from first place to fourth place in the 1996 Atlanta Olympics (4 min 8 sec *vs.* 4 min 6 sec) [4]. By using the wrong fixed gearing, differences of 2% are easily possible.

Previous published reports

There have been many published reports on the mechanical efficiency of bicycle transmissions during the past century; however, only a few have measured the efficiency using accurate mechanical means [1, 2, 5, 6, 7, 8, 9, 10, 11]. These studies found that bicycle drive efficiency depends upon many conditions such as load, chain tension, rpm, gear sizes, and the transmission type. As mentioned, the efficiencies varied from about 80% to 99%. The factors causing energy loss will be discussed in more detail later.

Mechanical methods of testing normally employ dynamometers that measure torque and rotational speed at the input and output of the drive system (with mechanical or electronic transducers). The combined energy losses in



Photo, courtesy of Chet Kyle

Bicycle crank dynamometer, furnished by the USOC Sports Sciences Division

all drive-train components such as the bearings, chains, sprockets, gears, and derailleurs are usually included in the efficiencies. However, some studies report the efficiency only of isolated components [6, 7, 9]. Thom [6] measured the efficiency of three-speed hub gears and bearings without including sprocket losses. Dell'Oro [7] isolated derailleur losses from the rest of the drive system. Cameron [9] measured the required static force to lift a known weight with a bicycle chain draped over a single sprocket. He assumed losses were constant with rpm, and estimated fixed-gear efficiencies under various loads. The remaining studies measured the overall efficiency of the bicycle drive system [1, 2, 5, 8, 10, 11].

Indirect methods such as repetitive field time trials, field or laboratory oxygen-consumption studies, crank-power-meter field trials, or crank-power-meter studies on stationary trainers, lack the necessary precision to give reliable results. Usually such methods have an error band of several percent.

New tests

During 9–13 October 2000, in the Laboratory of the Browning Research Facility on Bainbridge Island, Washington, the authors and Peter Kauffman, technical consultant to Browning Research, used a specially-devised dynamometer system to measure the mechanical efficiency of eleven bicycle transmissions. The transmissions were two Browning automatic bicycle transmissions (a 4-speed, and a 12-speed), a Shimano 27-speed mountain-bike derailleur transmission and eight internal hub-gear transmissions (Shimano 3-, 4- and 7-speed, Sachs 3- and 7-speed, Sturmey Archer 3- and 7-speed, and a Rohloff 14-speed.

Most of the previous bicycle-transmission tests were done on derailleur-type chain drives [1, 7, 8, 9] and these efficiency tests were limited to only a few gears. As far as the authors know, the wide-ranging 27-speed transmissions available today have not been tested, or at least the tests have not been published. No doubt manufacturers have tested their transmissions for efficiency, but if so, the results of their tests are unpublished.

Prior to the 1970s, before derailleur-equipped bikes became really popular, there were some efficiency tests performed on planetary hub gears [5, 6]. Hub gears are still quite popular today in Europe where they are used mainly on city commuter bikes. Hub-gear transmissions have the advantage of being nearly weatherproof, with low maintenance—and they permit a chain guard that completely shields the chain, and allow bicycle commuting without worrying about soiling good clothes on a greasy chain. However, they have never been popular with serious recreational cyclists or racers since the range of gears has been limited. Also, they are heavier than a derailleur-type transmission and they have had the reputation of being mechanically inefficient. Recently, however, there has been a revival of interest in the hub gear for several reasons. They are now available with an increasing number of gears (as many as 14), they lend themselves to fully automatic operation, and they can easily be adapted to bikes with an electric-motor boost. Regarding the hub gear's reputation for mechanically inefficiency, this paper will present information that shows this is not necessarily so.

Purpose of current tests

The purpose of the current tests was to compare the mechanical efficiency of the most common types of bicycle drives under identical conditions. Since limited time was available, the test apparatus had to handle all of the most common types of bicycle transmissions and to rapidly measure efficiency. Since power input to a bicycle crank is typically between 50 and 400 watts [4], and since losses can be as low as one to two percent, the transmission test system had be sensitive enough to determine power differences of just a few watts (less than 5).

Test equipment

The test system consisted of four main elements (see photo on page 3).

1. Bicycle crank dynamometer

To measure input power, a dynamometer fed power to a bicycle crank by means of a 2-horsepower variable-speed DC motor, mounted on gimbals so that the motor case could rotate freely. The motor case was restrained by a torque arm attached to an electronic load cell that measured the torque force. Oscillations in the load were smoothed by connecting the torque arm to the load cell through a thin nylon cord that transmitted force through a flywheel-type inertial damper. The rpm of the motor shaft was measured by timing each revolution electronically. The output shaft of the motor was connected to a bicycle crank through a flexible coupling. Knowing the torque and the rpm, the input power to the crank could be calculated. The dynamometer was furnished by the U.S. Olympic Committee (USOC) Sports Sciences Division.

The power input to the bicycle crank was given by:

$P_i = k\tau\omega$ where P_i is the power, k is a proportionality constant, τ is the torque and ω is the angular velocity of the crank.

2. Bicycle-drive-train fixture

A special test fixture was built to mount a bicycle bottom bracket, crank and chainrings, plus a rear hub without spokes or wheel. On the non-drive side of the hub, a sprocket was attached to the hub which drove a Monarch bicycle ergometer wheel. The adjustable fixture was built by Jim Merz for Browning Research, and it allowed rapid changing of front sprockets, chains and rear hubs.

3. Monarch ergometer wheel

To measure power output, a Monarch aluminum ergometer wheel was driven by a chain from the drive-train fixture through two 36-tooth sprockets, one on the ergometer wheel, and one on the non-drive side of the bicycle hub. A nylon cord, approximately 3 mm in diameter, was wrapped twice around the ergometer wheel with one end attached to a transducer and the other hanging downward with a suspended weight. The direction of rotation of the wheel was away from the hanging weight so the tension in the load-cell cord (slack side) was a small fraction of the applied hanging weight (load side). The ergometer load and thus the power output could be adjusted by hanging various weights on the nylon cord. Knowing the difference in tension between the two cords and the rpm, the output power from the bicycle hub could be calculated. The rpm of the ergometer wheel was measured electronically.

The power output of the system was: $P_o = k\omega_o(T1-T2)$, where P_o is the output power, k is a proportionality constant, ω_o is the ergometer wheel angular velocity, T1 is the weight, and T2 is the slack-side tension.

A disadvantage of this method was that the friction losses in the ergometer wheel drive were unknown. In order to find the corrected transmission efficiency, the ergometer drive losses would have to be determined, and this was done only at 75 rpm. However, for determining the rank order between transmissions, since they were all tested under identical conditions, no correction is necessary. The efficiencies reported in this article include ergometer-wheel drive losses, so the actual transmission efficiencies would be higher by 2 to 2.5%.

4. Data-acquisition system

A portable computer was adapted by Peter Kauffman of Browning to receive signals from the load cells and revolution counters. The computer sampled the transducers and averaged the readings over a selected time interval. The software automatically calculated ergometer power along with the mechanical efficiency of the bicycle drive including the ergometer drive. All of the data and calculations were displayed in tabular form on the computer screen, and the data were stored for later analysis.

Test procedure

- The load cells were calibrated using weights. The load cells agreed with the weights within ±0.2%. The accuracy of the angular-velocity transducers of both the crank and the ergometer wheels were checked by two methods. The crank rpm was verified with a stop watch. The rpms of both the crank and the ergometer wheel, as indicated by the transducers, were then used to compute the gear ratio which was compared with the known ratio. The calculated gear ratio agreed with the known ratio normally within three significant figures (one part in 1000).
- The first test series was with the crank dynamometer directly connected to the ergometer wheel through two 36-tooth gears. The purpose was to estimate the power losses of the ergometer wheel drive. Since chain tension is probably the most important factor in gear friction [8] the ergometer wheel weights were the same as those used in normal testing—from 1.8 kilos to 16 kilos. The speed of the crank and wheel were constant at 75 rpm. This test did not directly measure ergometer-wheel drive losses since the wheel rpm did not vary (as when testing transmissions). Also, the bottom-bracket bearings were in the loop, making an extra set of bearings. The friction losses were small (from 1 to 6 watts; see fig. 13*), but as previously mentioned, accounting for the losses would raise the reported efficiencies by 2 to 2.5%.
- The test fixture was then used to test the efficiency of eleven transmissions. Weights were chosen to produce 80 watts, 150 watts and 200 watts output power at 75 crank rpm. All chains were well oiled with light machine oil. Hub gears were usually left with their original grease lubricant, but this was replaced in two hubs with light oil.

The transmissions that were tested had the following gears.

Derailleur-type transmissions

4-speed automatic: Browning

This transmission has a gear layout similar to a standard derailleur system except electronically actuated hinged gear segments in the rear cluster shift the chain up or down either automatically or manually. The Browning chain guide and tensioner, with its two jockey pulleys, has a similar appearance to a derailleur, and probably has nearly identical friction characteristics. It is however a passive follower. In this paper, the two Browning transmissions and the 27-speed derailleur transmission will often be referred to as "derailleur-type" transmissions. The Browning 4-speed was tested with a 42-tooth front chainring and a 12-, 17-, 23-, and 32-tooth rear cluster.

12-speed automatic: Browning

An automatic transmission similar to the Browning 4-speed, except with three front chainrings 48/38/30, and the same 4-speed rear cluster 12/17/23/32. The gears are (1) 30/32; (2) 38/32; (3) 30/23; (4) 48/32; (5) 38/23; (6) 30/17; (7) 48/23; (8) 38/17), (9) 30/12); (10) 48/17; (11) 38/12; and (12) 48/12.

27-speed: Shimano

A Shimano Ultegra 27-speed mountain-bike transmission with three front chainrings (44/32/22 teeth) and a 9-speed rear cluster (12, 14, 16, 18, 20, 23, 26, 30, and 34 teeth). Because of time constraints, only 15 of the 27 gears were tested: (1) 22/34; (3) 22/26; (4) 32/34; (7) 22/20; (9) 32/26; (10) 44/34; (11) 22/16; (15) 32/20; (16) 44/26; (18) 22/12; (20) 32/16; (21) 44/20; (24) 32/12; (25) 44/16; and (27) 44/12.

Planetary-geared rear hubs

3-speed: Sachs

An internal planetary-geared rear hub with a 40-tooth front chainring and a 19-tooth rear cog. The three hub gears are: (1) Ratio = 0.75); (2) 1.00; and (3) 1.33.

3-speed: Shimano

A rear hub with a 40-tooth front chainring and a 19-tooth rear cog. The three hub gears are: (1) 0.74; (2) 1.00; and (3) 1.36.

3-speed: Sturmey Archer

A rear hub with a 40-tooth front chainring and a 19-tooth rear cog. The three hub gears are: (1) 0.75; (2) 1.00; and (3) 1.33.

4-speed: Shimano Auto D

A rear hub with a 31-tooth front chainring and a 23-tooth rear cog. The four hub gears are: (1) 1.00); (2) 1.24; (3) 1.5; and (4) 1.84.

7-speed: Sachs

A rear hub with a 40-tooth front chainring and a 19-tooth rear cog. The transmission shifter was damaged and could be shifted to only two gears: (1) 0.59 and (4) 1.00.

7-speed: Shimano Nexus

A rear hub with a 40-tooth front chainring and a 19-tooth rear cog. The seven hub gears are: (1) 0.63); (2) 0.74; (3) 0.84; (4) 0.99; (5) 1.15; (6) 1.34; and (7) 1.55.

7-speed: Sturmey Archer

A rear hub with a 40-tooth front chainring and a 19-tooth rear cog. The seven hub gears are: (1) 0.60; (2) 0.69; (3) 0.80; (4) 1.00; (5) 1.24); (6) 1.45; and (7) 1.68.

14-speed: Rohloff

A rear hub with a 40-tooth front chainring and a 16-tooth rear cog. The fourteen hub gears are: (1) 0.279; (2) 0.316; (3) 0.360; (4) 0.409; (5) 0.464; (6) 0.528; (7) 0.600; (8) 0.682; (9) 0.774; (10) 0.881; (11) 1.000; (12) 1.135; (13) 1.292; and (14) 1.467.

Results and discussion

We tested each transmission at three loads: 80 watts, 150 watts, and 200 watts (power output at the ergometer wheel)—all at 75 rpm. The crank speed of 75 rpm was chosen as being typical of recreational cyclists. There was insufficient time available to test each transmission at both variable load and variable rpm. The power outputs of 80, 150 and 200 watts, represent the typical energy requirements of commuting or recreational cyclists in good physical condition, traveling at speeds from 24–35 kph (15–22 mph), on a level, smooth road with no wind [1, 3]. Bicycle racers can produce steady

*See pages 8–11 for figures and tables.

power outputs that are much higher than this for periods of more than one hour—from 300 to 450 watts [3]. Although the occasional recreational cyclist may produce over 200 watts, it is doubtful that cyclists using hub gears would frequently put out more than 150 watts unless being chased by rabid dogs. The results of the tests are shown in figures 1–14.

Plotting efficiency

In figures 1–12 the efficiency is plotted in three ways.

1. Efficiency vs. power output

Here all of the individual power and efficiency data points were plotted for each gear. These curves give the detailed performance of each transmission under varying load. As examples, see figures 1, 4 or 5. All transmissions were not plotted but they could be, using the data in tables 1 and 2.

2. Average efficiency vs. gear number

Here, efficiencies for all test loads were averaged for each gear and the averages were plotted against the gear number. This curve shows the effect of gear ratio on efficiency under varying load conditions. For examples see figures 2, 6, 8, 10, or 11.

3. Average efficiency vs. load

Here, transmission efficiencies for each load were averaged for all gears. This curve is a measure of the performance of each transmission under varying conditions. For example, see figures 3, 7, 9, or 12. These curves provide probably the simplest way to compare transmissions.

Conclusions

By viewing the curves, several general observations and conclusions can be made.

1. Efficiency generally increases with the load—for all transmissions.

Figures 1, 3, 4, 5, 7, 9, 12, or 14 all show this trend. Although friction increases with chain load, rpm, and other factors [8], obviously the residual friction in a gear train becomes less important as the input power increases, while the friction factors that increase with load go up less rapidly than the load.

The clearest example of this is shown in figure 14. This was the only case where we tested a transmission at over 200 watts and under 80 watts. More tests were planned, but a shear pin parted in the drive train and this experiment was aborted. The uncorrected efficiency increased from about 91% to over 97% as the output power increased from 50 watts to 370 watts at 75 rpm.

By assuming that ergometer-wheel rpm has no effect on the drive losses (fig. 13), a rough estimate of the absolute system efficiency can be made. Spicer shows that drive-train losses are a function of the crank rpm [8]; however, as previously explained, this effect was not measured. When corrected for ergometer-drive losses, the transmission efficiency increases from 1% to 3% (see fig. 14). Efficiency is over 98% at the highest load. The corrected efficiencies are in good agreement with Spicer [8] who found that efficiency was over 98% with 52/15-tooth sprockets at 200 watts.

2. Hub gears are generally about 2% lower in efficiency than derailleur-type gears. But there are exceptions.

This is illustrated by figures 3, 6, 7, and 12. Figure 12 shows that the efficiencies of the Shimano 4, Sachs 7, Shimano 7, Sturmey 7 and the Rohloff 14 all cluster about two percent lower than the Browning 4, Browning 12, or the Shimano 27.

However, two of the 3-speed hub gears did not follow this trend.

The grease in the Sachs 3 and the Sturmey Archer 3-speeds was replaced with light oil, and unlike the other hub gear transmissions, the efficiencies of the Sachs 3 and Sturmey 3, compare well with the best of the derailleur transmissions (figs. 7, 9, and 12). Also, these transmissions were worn in, whereas many of the others were new. Manufacturers would do well to replace heavy grease in their hub gears with light oil. Although oil wouldn't last as long as grease, the energy savings would be significant. Unfortunately commuters have a tendency to ignore maintenance until something breaks, so light oil probably wouldn't be a popular choice.

Also, with the Shimano 4, the first gear (a 1.0 ratio) had a higher efficiency than the derailleur transmissions, even though gears 2, 3, and 4 had a lower efficiency (see fig. 6). In a planetary transmission (also called epicyclic), even when the hub ratio is 1.0, the planet gears are still in motion [12]; however, all of the planetary transmissions we tested had high efficiency at 1.0 gear ratios.

3. As the gear ratio increases, the efficiency tends to decrease for all transmission types.

This is illustrated by the trend lines in figures 6, 8, 10, and 11. Even though the greatest efficiencies are sometimes near the highest gear ratios, the average efficiency decreases with higher ratios, (the high efficiencies were: Shimano 4 = gear 1, Rohloff = gear 9, Browning = gear 2, and Shimano 27 = gear 21).

4. With modern transmissions, where multiple gears are available, there is often a difference of 1% to 3% in efficiency between adjacent gears.

This applies to both hub gears and to derailleur gears. See figs. 2, 6, 8, 10, and 11 (especially figures 8, 10 and 11).

In figure 11, in the Shimano 27-speed, there is a 4% difference in efficiency between gears 21 and 24 and between gears 24 and 25. In figure 8, for the Rohloff 14, there is a 3% difference between gears 7 and 8.

An average 2% difference in efficiency is thus easily possible if the wrong gears are chosen.

If racers, or even commuting or touring cyclists, could choose optimum gears they would be hundreds of meters ahead at the end of 60 km (37 mi). For example, if Lance Armstrong, in the Tour de France 58.5-km time trial (36.4 mi) were to choose the wrong gear, a drop of 2% in efficiency would cause him to be 410 meters behind (27 seconds) at the end of the time trial, easily enough to lose the stage [3]. Incidentally, Armstrong averaged about 54 kph (33.6 mph) for the time trial (58.5 km long = 36.4 mi).

With commuting riders who travel 24 kph (15 mph), instead of 54 kph (33.6 mph), it only gets worse. A 2% drop in efficiency would lead to an 800-meter gap (about 2 minutes). The reason for the increasing gap is that the slower cyclist spends much more time on the course [3]. The point is, why waste energy when it is unnecessary.

5. The tests show that some gears are inefficient.

Hub gears

In hub gears, such as the Rohloff 14, the efficiency no doubt depends on how many elements of the gear train are in motion as each gear is selected (see fig. 15). In the Rohloff, gears 3, 5, 7, 12, and 14 have the lowest efficiency. This superb but complex transmission has roller bearings and uses light oil as a lubricant. Shifting is quite simple: successive gears are reached by pulling on the single shift cable in one direction or the other. No attempt will be made to explain this mechanism. It is obvious from the diagrammatic illustration (fig. 15) that it cannot easily be explained.

Derailleur gears

On the other hand, factors affecting the efficiency of derailleur gears become clear by examining the curves in figures 10 and 11. For example, a 12-tooth sprocket seems to cause inefficiency. In the Shimano 27-speed, gears 4, 9, 15, 18, and 24 have the lowest efficiency. The two gears with the lowest efficiency of the 15 tested, both use a 12-tooth sprocket. The gears with 12-tooth sprockets (18, 24 and 27) have an average efficiency of 91.2%, while those involving 16-tooth sprockets (11, 20 and 25) have an average efficiency of 93.5%.

Other gears

In the Browning, the 12-tooth sprockets averaged 92.1% efficiency, while the gears involving a 17-tooth sprocket averaged 92.9%. The two lowest efficiencies of the 12 gears tested had 12-tooth sprockets (gears 9 and 12). Apparently the sharp angle of chain link bend in the 12 causes increased friction compared to larger sprockets. So it appears that larger gears than 12 are necessary for efficient operation. When there is a choice of gear ratios that are close, cyclists should choose the gearing combination with larger diameters [8].

Cross-chain gears make little difference. In the Shimano 27, the cross chain between the two big gears on the Shimano has a higher-than-average efficiency (gear 10, 44/34), while the cross chain between the two small sprockets involves a 12-tooth sprocket (gear 18, 22/12; see fig. 11). In the Browning, the large cross-chain gears (gear 4, 48/32), have a higher-than-average efficiency, while the small-gear cross chain involves a 12-tooth sprocket (see fig. 10).

For some reason that is not apparent, the mid-chainrings on both the Browning 12 and the Shimano 27 did not have high efficiencies. On the Browning 12, gears using the 30-tooth chainring (1, 3, 6, and 9) had a lower-than-average efficiency. On the Shimano 27, gears using the 32-tooth chainring (4, 9, 15, 20 and 24), all had a lower-than-average efficiency. This does not appear to be a coincidence, but the reason is not clear.

Had more time been available, it would have been interesting to measure the effect of such things as rpm, all gears in the 27-speed, a wider range of power inputs, and various chain and hub-gear lubricants. As usual, there are more questions than answers.

Credits

The authors wish to thank Browning Research for making available the facilities of their laboratory for this project and for supporting this study.

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The authors

Chester Kyle, adjunct professor of mechanical engineering at California State University at Long Beach, is a consultant on the science of sports equipment and has worked with several teams and organizations: US Postal Service 2001 Tour de France team, design teams for USA 1984 and 1996 Olympics cycling teams' bicycles and clothing, and Nike, as well as others, for aerodynamic sports clothing.

Co-organizer of the first International Human Powered Speed Championships at Irwindale, California, in 1975, Kyle and eleven others founded the International Human Powered Vehicle Association (IHPVA) the following year. Kyle is past president and secretary of the IHPVA, as well as the *de facto* historian of the organization. Editor and publisher of *Cycling Science* (1989–1991) and science editor of *Bicycling Magazine* (1984–1989), Kyle is a frequent contributor to scientific and popular publications.

Chet Kyle and his wife, Joyce, live on ten acres of rural pasture and forest in a home they and their four, now-grown children built near Weed, California.

Frank Berto, author of more than 150 articles and several books on cycling technology, was engineering and West Coast editor of *Bicycling Magazine* (1986–1990). Berto is a consultant on oil field gauging and instrumentation, cycling equipment and technology (especially gearing), as well as a frequent expert witness on cycling litigation. He is also a historic aircraft and machinery enthusiast.

Frank and Connie Berto live in San Anselmo, California, on a large plot of land affectionately called "Sleepy Hollow".

Berto's latest book, *The Dancing Chain*, was reviewed in *Human Power* 51, Fall 2000.

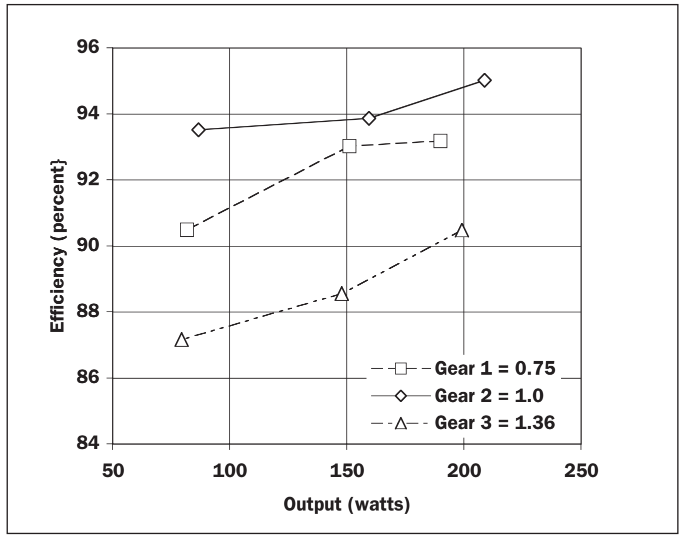

Figure 1. Shimano 3-speed (efficiency *vs.* load)

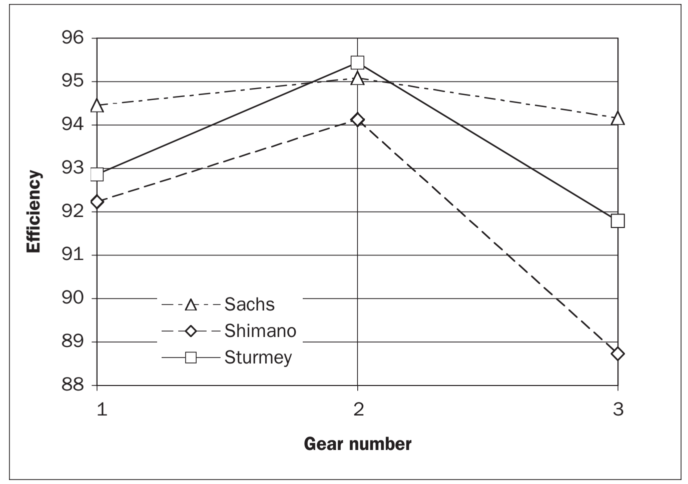

Figure 2. Sachs 3, Shimano 3, Sturmey 3 (average efficiency *vs.* gear)

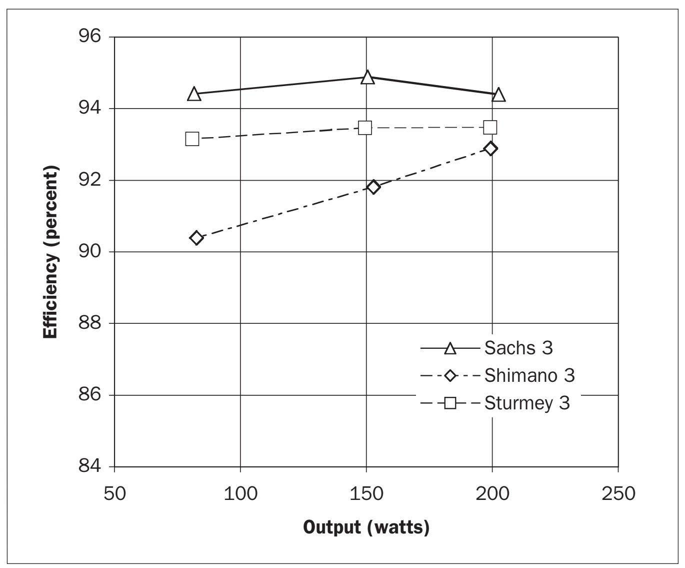

Figure 3. Sachs 3, Shimano 3, Sturmey 3 (average efficiency *vs.* load)

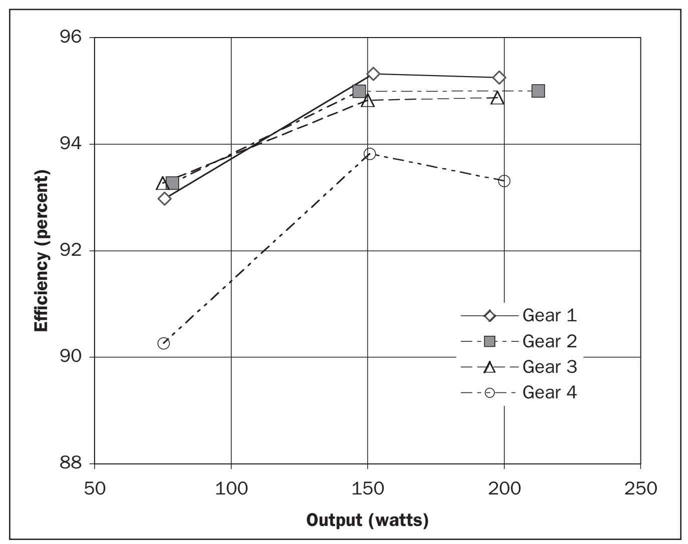

Figure 4. Browning 4-speed (efficiency *vs.* load)

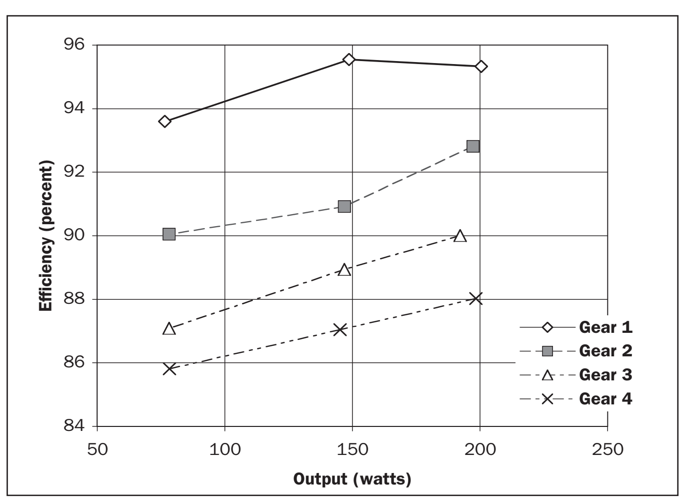

Figure 5. Shimano 4-speed (efficiency *vs.* load)

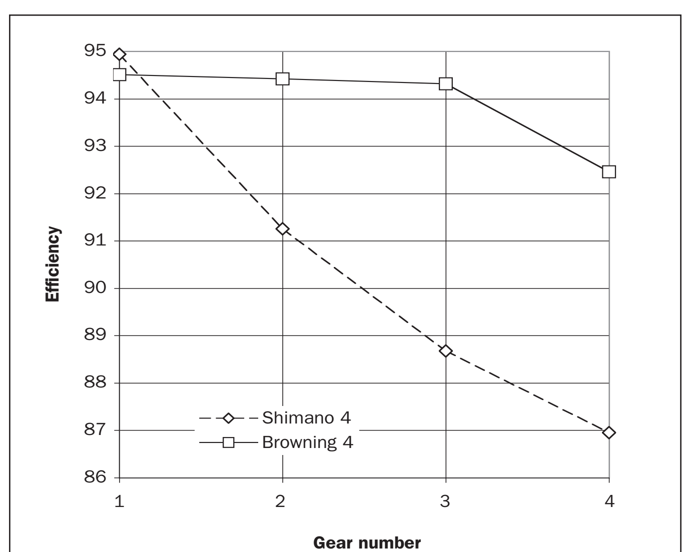

Figure 6. Browning 4, Shimano 4 (average efficiency vs. gear)

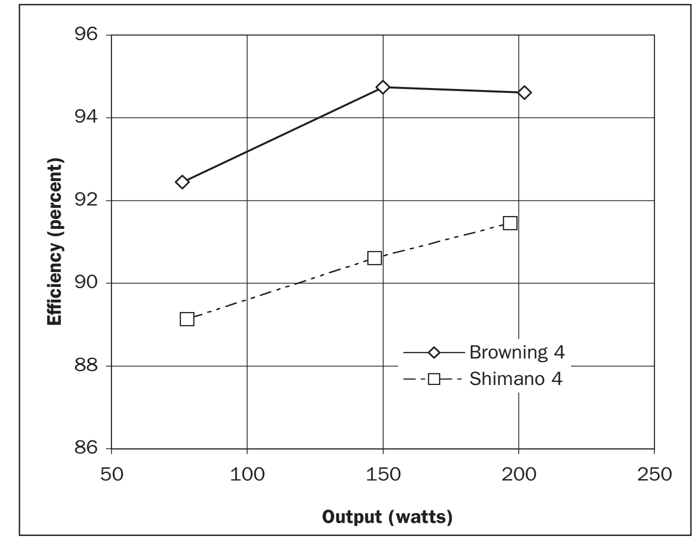

Figure 7. Browning 4-speed, Shimano 4-speed (average efficiency *vs.* load)

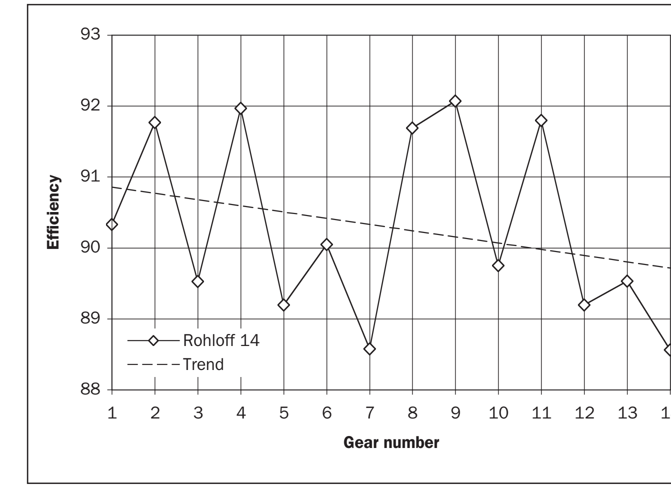

Figure 8. Rohloff 14 (average efficiency *vs.* gear)

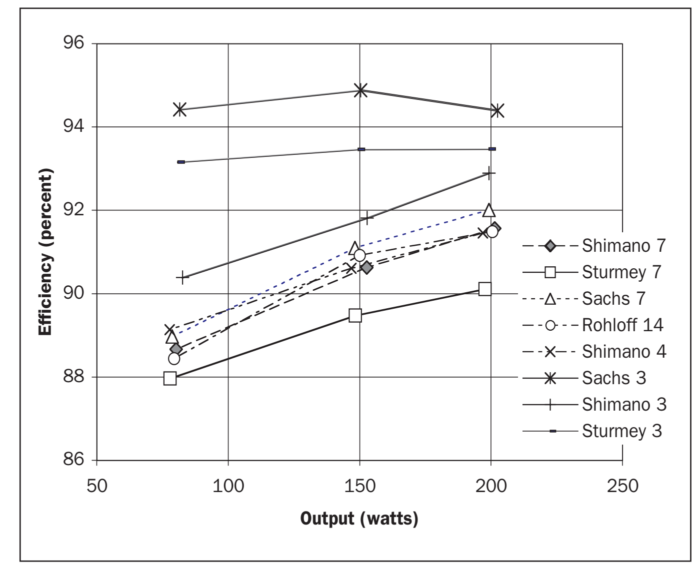

Figure 9. Hub gear bicycle transmissions (average efficiency *vs.* load)

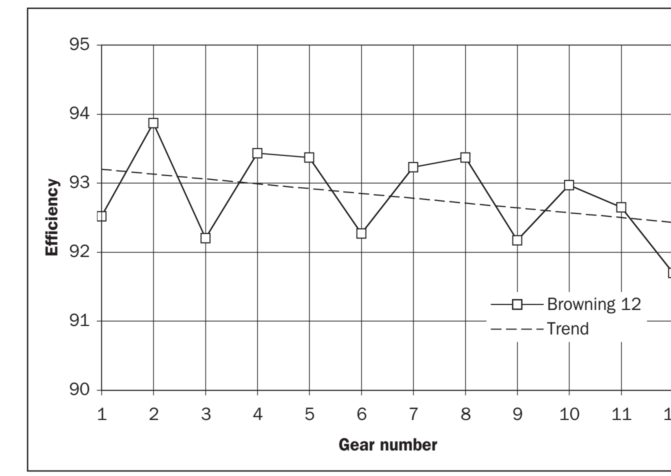

Figure 10. Browning 12 (average efficiency *vs.* gear)

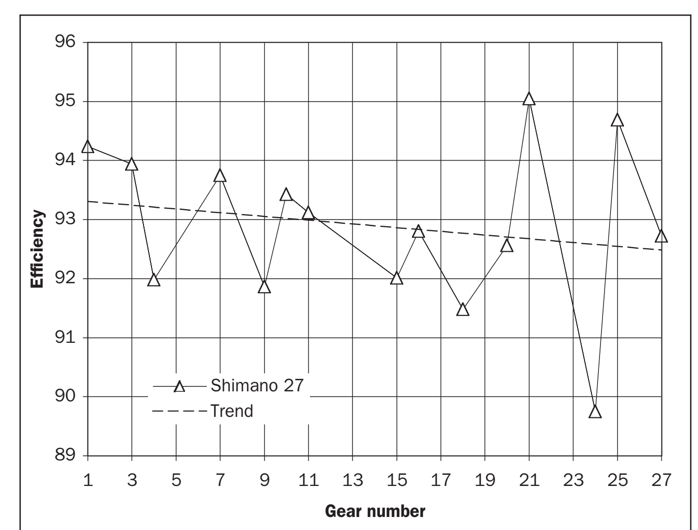

Figure 11. Shimano 27 (average efficiency *vs.* gear)

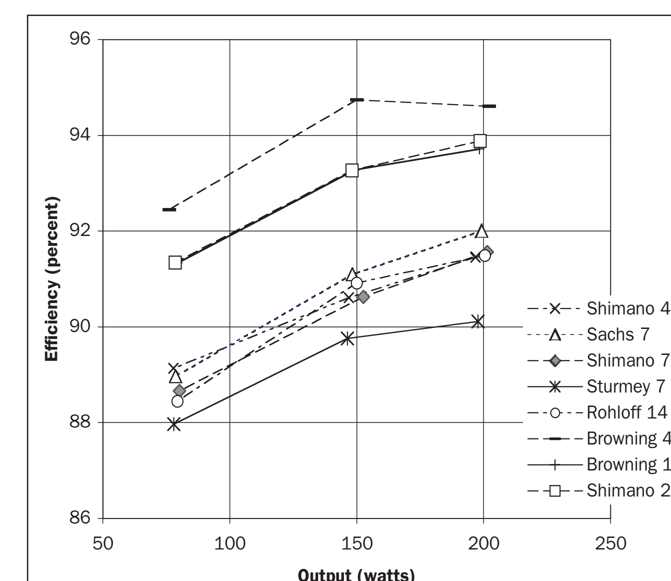

Figure 12. Derailleur-type transmissions compared with hub gears (average efficiency vs. load)

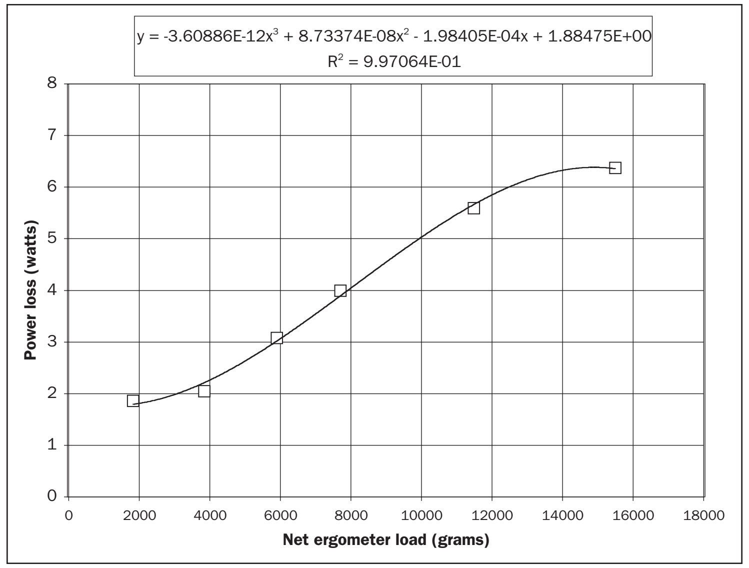

Figure 13. Power loss *vs.* net ergometer load

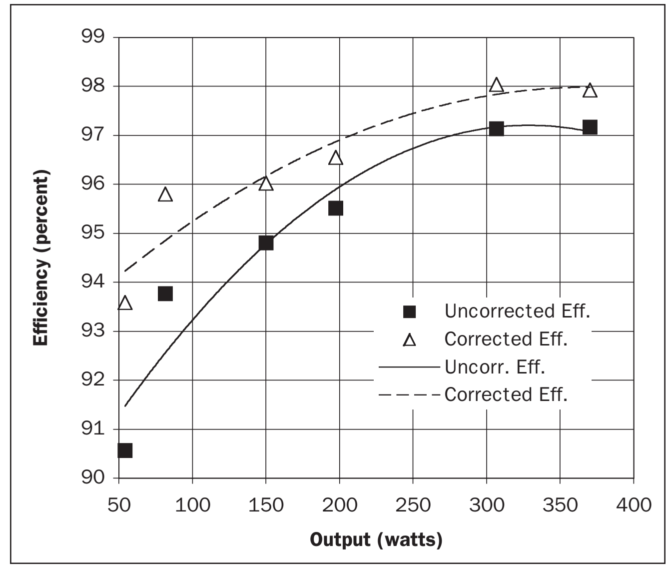

Figure 14. Shimano 27-speed, gear 25 (44/16) uncorrected and corrected (efficiency *vs.* load; 75 crank rpm; correction is estimated)

Table 1. Hub gear transmisions: mechanical efficiency *vs.* load

Maker/Speeds	Gear = Power	1	2	3	4	5	6	7	8	9	10	11	12	13	14
		Efficiency Percent**													
Sachs 3	80	95.0	92.9	93.6											
	150	94.2	95.6	94.8											
	200	94.1	94.9	94.1											
Shimano 3	80	90.5	93.5	87.2											
	150	93.0	93.9	88.6											
	200	93.2	95.0	87.2											
Sturmey 3	80	92.3	95.4	91.8											
	150	93.3	95.3	91.8											
	200	93.0	95.6	91.8											
Shimano 4 Automatic	80	93.6	90.1	87.1	85.8										
	150	95.6	90.9	88.9	87.0										
	200	95.3	92.8	90.0	88.0										
Sachs 7	80	88.7	—	—	89.2										
	150	89.9	—	—	92.3										
	200*	91.0*	—	—	93.0*										
Shimano 7	80	90.8	90.7	87.4	89.0	83.6	90.9	88.2							
	150	91.8	92.9	89.9	89.0	85.6	92.8	90.4							
	200	92.8	94.5	90.3	91.8	86.4	93.7	91.4							
Sturmey 7	80	87.3	88.7	88.4	93.0	89.3	86.0	83.0							
	150	89.1	89.0	91.1	93.3	90.4	88.5	85.4							
	200	89.7	90.3	91.3	94.7	91.0	88.6	85.3							
Rohloff 14	80	89.1	90.3	87.8	90.3	87.5	87.8	86.1	89.7	90.8	87.7	89.7	87.1	87.8	86.1
	150	90.6	92.5	89.9	92.2	89.6	91.0	89.9	92.6	92.7	90.4	92.3	90.4	89.7	89.1
	200	91.3	92.5	90.9	93.4	90.5	90.9	90.2	92.8	92.7	91.1	93.5	90.0	91.1	90.4

The shift mechanism was broken, and would shift to only two gears.

*** All efficiencies are uncorrected for the power consumed by the ergometer wheel drive. Although this is not large, it would increase the indicated efficiencies by 2 to 2.5% in most cases.*

Table 2. Derailleur-type transmisions: mechanical efficiency *vs.* load

Maker/Speeds	Gear = Power	1	2	3	4	5	6	7	8	9	10	11	12	15	16
		Efficiency Percent													
Browning 4 Automatic	80	93.0	93.3	93.3	90.3										
	150	95.3	95.0	94.8	93.8										
	200	95.3	95.0	94.9	93.3										
Browning 12 Automatic	80	91.1	92.5	91.3	91.6	92.5	91.2	91.9	90.7	90.9	91.1	89.8	89.8		
	150	93.8	93.9	92.5	94.5	93.3	92.9	93.8	93.5	92.2	93.7	93.4	91.8		
	200	92.7	95.2	92.8	94.2	94.3	92.7	94.0	94.4	93.4	94.1	93.2	93.5		
Shimano 27 Ultegra Mtn. Grupo	80	93.1	—	92.8	89.4	—	—	92.6	—	90.0	92.1	91.7	—	89.5	91.0
	150	94.6	—	94.6	92.9	—	—	94.5	—	92.5	93.9	93.8	—	93.0	93.6
	200	95.0	—	94.5	93.6	—	—	94.2	—	93.1	94.2	93.9	—	93.6	93.9
Shimano 27 (continued)		18	20	21	24	25	27								
	54	—	—	—	—	90.6									
	80	90.7	90.9	94.3	86.9	93.8	91.1								
	150	91.8	93.0	95.0	91.0	94.8	93.3								
	200	91.9	93.8	95.9	91.4	95.5	93.7								
	307	—	—	—	—	97.1									
	370	—	—	—	—	97.2									

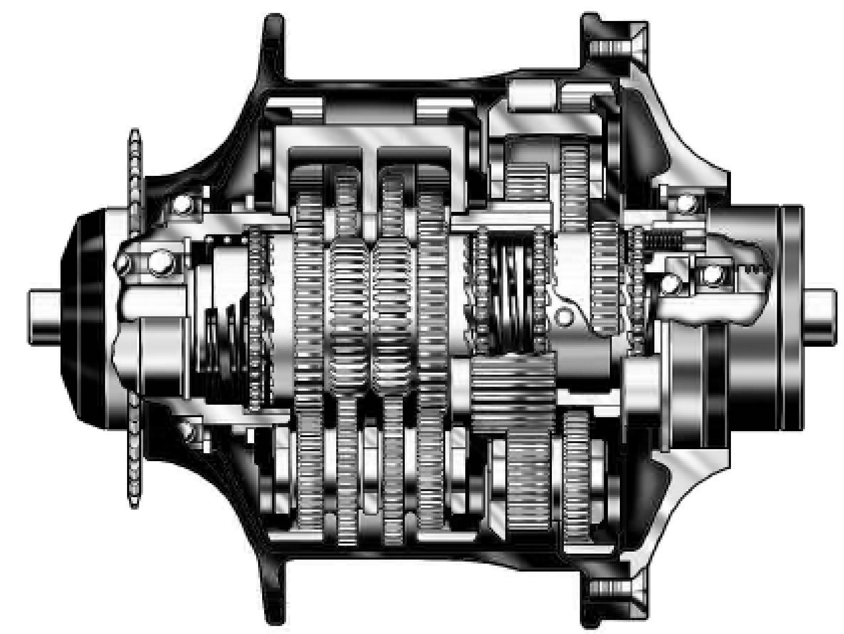

Figure 15. Diagrammatic view of the Rohloff hub

TECHNICAL NOTES

Bicycle stability after front-tire deflation

Dave Wilson (reporting partly for Soohyun Park)

We reported in *Human Power*, 51 (pp. 16–18) on experiments to provide steering stability after a front tire has deflated, there having been many reports of "flopping" instability that caused riders to be thrown off virtually instantaneously. We reported the tentative findings of Andy Oury, who increased the bead-seat diameter of so-called "drop-center" rims[1] and thereby greatly decreased the tendency of deflated tires to "flop" from side to side.

This past academic year another MIT undergraduate student, Soohyun Park, chose to do her BSME thesis[2] on a continuation of this study. She first researched an improved bicycle model, resulting in the use of a BMX bicycle with a weight mounted on it representing approximately a rider's weight and center of mass. She found that over a wide range of weight values and positions the tire behavior when this bicycle was pushed across the laboratory floor was better represented than in the previous program.

Park then built up the bead-seats of a wheel that had caused me serious trouble when the tire had deflated as I was in front of a very large truck. She used fiberglass tape and polyester resin (Oury had used layers of masking tape, which gave a soft seat of low strength). She found, as did Oury, a steady improvement in behavior as the bead-seat diameter was increased. Subsequently I continued the build-up (the fit between the tire and rim was exceedingly loose) until the diameter was too large for the tire, and then machined it down (using a profiled router) until a smoothly shaped round-ed-edge bead seat was produced that allowed the tire bead to snap into position only after the tube was inflated to about half final pressure.

When this final step was taken the difference in performance changed dramatically. Flopping disappeared entirely, and the tire could provide safe and stable bicycle direction during the deceleration after deflation.

These results therefore add to the previous somewhat tentative recommendation: that wheel and tire manufacturers and standards organizations should arrive at standards for the sizes and profiles of rims and of tire beads so that a fit tight enough to produce stable steering under deflated conditions is achieved. There seems little doubt that many deaths and injuries would thereby be prevented.

—Dave Wilson
<dgwilson@mediaone.net>

1. Cycle and motorcycle tires and wheel-rim standards. ETRTO, supplied by the Taiwan Bicycle Industry R&D Center, Taichung, Taiwan, 2001.
2. Park, Soohyun (2001). *Run-flat performance of bicycle tires and modified rims.* BSME thesis, Cambridge, MA: MIT.

There is a better way than rolling

by Detlev Tschentscher

Human-powered vehicles on land usually have wheels. But there are attractive alternatives.

Walking and running aids

Humans are just ordinary mammals except for two differences:

- we walk on two legs; and
- we consider ourselves to be intelligent.

This should mean that we have the ability to improve our lives. It is surprising therefore that we do not use this intelligence to improve our natural way of movement: walking on two legs. John Dick (one of the designers of the Springwalker, member of the DARPA* team) describes the situation as follows: "We have had 150 years of engineering now, and still there is no powered exoskeleton."

When we refer to human-powered vehicles on land, we usually mean wheeled vehicles. And, as we all know, there has been enormous progress as these types of bicycles and other HPVs have evolved. They seem to be given attention only when breaking a record. Nearly every college in the U.S. has a project group devoted to human-powered vehicles. The technology and parts to build an advanced bicycle are available for reasonable prices all over the world. But what makes us believe that rolling is the only way of moving such that it is worth so much attention? Of course, if we had to choose between walking or riding a bike for traveling a distance of a few kilometers on a gravel road we would definitely choose the bike. The reason for this is obvious: riding a bike takes less effort than walking. But what does this prove? Simply that a mechanically supported method of movement is easier than a non-supported movement.

To be able to compare walking to rolling, as in the situation described above, we should establish equal opportunities between the two methods. Both the cyclist and the runner should be mechanically supported. But what does a device for the support of human running look like? To give an answer we first need to analyze where and how the human way of running needs to be supported. One of the main weak points in human running is, that (because of our leg design) we use only little energy for the forward movement. If we would divide a normal step into separate actions, only the part where we jump up to move forward is useful in gaining ground. The rest of the movement is wasted for our fight against gravity.[1] Another approach for support is to focus on increasing the distance covered with just one step.

Research in bionics shows that kangaroos for example can run long distances at very high speed with very low energy consumption. They can jump up to a length of six meters and store the energy that would normally be wasted by a kind of spring-mechanism, using their tail as a kind of spring. Several approaches have been made to make this simple phenomenon available for humans.

In the early 1920s a number of patents came up which basically used the idea of a pogo-stick attached to the lower leg. But these patents did not result in much improvement to running. Until 1990 nothing really significant occurred. Then a group of people around the technician John Dick built a prototype of an improved exoskeleton which he called the "Springwalker". The device was a huge step towards a usable walking device. It combined the use of artificial legs with a spring pack on the back of the runner (see fig. 1). Although the Springwalker was reported in all news media it was never improved to become a functioning device for the market. Today inventors focus on servo-powered versions of the Springwalker for military use. But a few other attempts have gone into production. Several kangaroo-boots have appeared on the market recently.[2] Most of these are aimed at fitness enthusiasts and are based on several spring-systems attached to ski-boot-like boots. With these boots it is possible to jump up to four meters at two meters high. Two technicians, Atanow and Gordejew, of the Lufthansa-university of Ufa, even created a boot powered by a fuel engine. With this device it is possible to take a one-hour walk using only a matchbox full of petrol (gasoline). But these efforts cover only a small section of the latest research on walking machines.

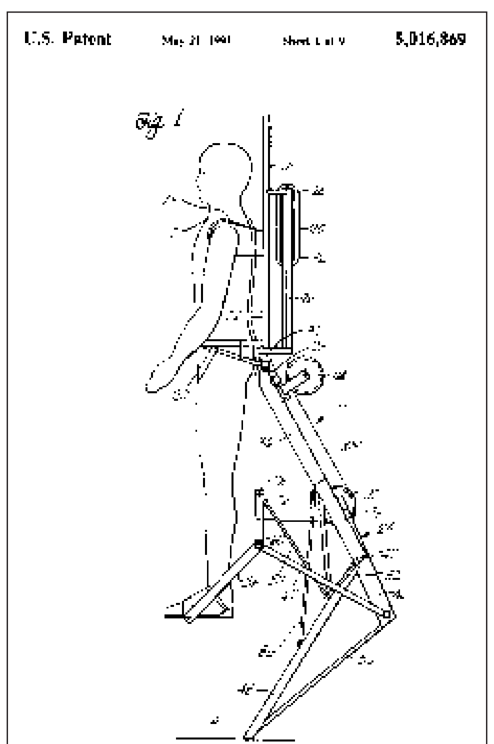

Figure 1. One of the illustrations from the Springwalker U.S. patent document

Most of the current research is in robotics. To create a servo-powered device that is able to walk requires sophisticated development combining biomechanics with information technology. This challenge has stimulated most advanced research institutes and companies to work on walking robots in some form or another. The number of projects is immense. Even big overviews such as the "walking machine catalog" of the German institute of the FZI Research Center for Information Technologies at the University of Karlsruhe[3] are not able to show the full range of historic and current projects. Most of the projects are focused on multi-legged vehicles or humanoid-legged robots. The key issue of these projects is to control the complex process of moving servo-powered legs without losing the balance. But all these devices have one thing in common: they rely on artificial power sources.

Only very few studies follow the former Springwalker in using human power as the only power source. One of them is the network-initiative Kenguru[4] that I started. We plan to build an empowered running device such as the Springwalker, except that a different kind of technology will be used: the power of the runner's arms. First contacts to industrial and other organizations have been made. All actual information about all current projects and the Kenguru initiative can be viewed on the internet on my homepage (see reference 4).

*US Defense Advanced Research Projects Agency, Department of Defense

References

1. Homepage of the DARPA: http://www.darpa.mil/dso/thrust/md/exoskeletons/index.html
2. A good example of "kangaroo" boots are made by Powerskip. See http://www.powerskip.com
3. Link to walking machine catalog: http://www.fzi.de/ids/WMC/walking_machines_katalog/walking_machines_katalog.html
4. Homepage: http://www.kenguru.de

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Tire-rim compatability

John Stegmann

Dave Wilson's thoughts on the subject of front-tire blowouts (*Human Power* 51, Fall 2000) reminded me of the difficulties we encountered making rims a decade ago. When I commented to Dave that we had not considered that tire manufacturers too might have difficulty in maintaining size standards, or that the wheel size might diminish during wheel building, he asked to reproduce the account which I wrote for *Cycling Science*, with adjustments if desired (*Cycling Science*, June 1990). I had been inspired in 1987, by articles that had appeared some years previously in Rodale's technical journal, *Bike Tech*, to make my own bicycle rims in order to overcome two difficulties. One was the high cost of imported aluminium-alloy rims, and the other was the difficulty in purchasing rims of unusual sizes and drillings that were needed for the recumbents we were making. Suppose I could make the rims I wanted and cover the cost by making and selling popular rims? I discussed the idea with friend and factory-owner Bill Rosenberg, and came to an arrangement whereby he would assist me to make the tools and would manufacture the rims in his factory, and I would provide the capital, the design, and marketing.

After reading Mario Emiliani's, "Heat treated rims: Are they worth the money?" (*Bike Tech* 2:5), I talked to a South African aluminium producer/extruder and decided that their 6063 aluminium alloy would be soft enough to roll easily, would be strong enough after heat treatment, and would then be suitable for anodizing if required.

Chris Juden's article, "The aluminium rim: Design and function," (*Bike Tech* 3:2), was the great inspiration. It provided a wealth of information on rims, tires and wheels. I chose to make a rim with an inside width of 16 mm which would suit tires from the then-popular 22-mm high-pressure tires to the more practical 38 mm. My new IZIZI profile should result in a mass of 280 gms/meter and suit the stock 4-mm aluminium rod that would be used to pin the joint. I based it on the successful Rigida 1622 which is similar to the Moulton. (IZIZI was the name I chose because it reads the same when viewed from either side of the wheel.)

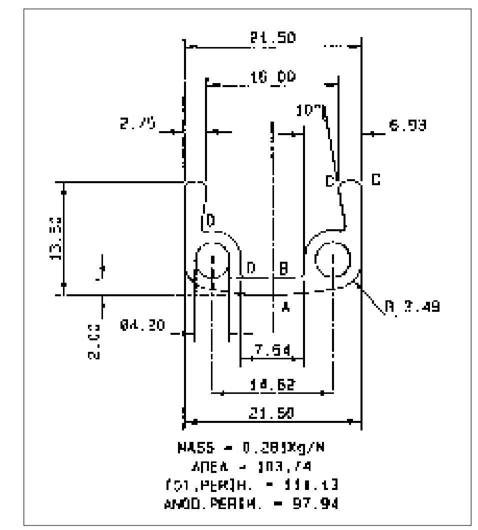

Illustration from *Cycling Science*, June 1990

By the end of January 1988 we had sample extrusions from the new die. We made two rollers—stage 1 and stage 2—and we marked out the hole centers by hand. It all looked good and we placed our first order for 50kg. The first rims we made were for my new experimental FWD recumbent with crank axle attached to the front fork, based on information and encouragement from Californian Tom Traylor. The bicycle was designed for 25-520 7.9-bar (115 psi) wired-on tires and was to be ridden in the Argus Tour on 5th March 1988. All it needed was wheels. There was no time for heat treatment as the bike was completed six days before the event and I still had to learn to ride it! To our surprise, the rims proved to be quite satisfactory and were never heat treated. Although heat treating after rolling is definitely the preferred procedure, a certain amount of hardening does take place during rolling, as well as during use and with age.

The next rim was made to suit a popular 25-622 fold-up high-pressure tire. We had it heat treated and anodized in dark bronze, and built a beautiful wheel with stainless-steel spokes and a red powder-coated hub. I had made an appointment in the morning with my bank manager to apply for a loan to start manufacture. I pumped the tire to 6.9 bar (100 psi) and set it aside. We were excited at the success and by the prospects. Little did we know that our problems were about to begin.

I was still busy in my workshop five hours later when the tube exploded! Why? Was the tire defective? Had the tire or tube been badly fitted? Was the rim under size? I checked as much as I could, yet found no obvious reason for the failure. I therefore fitted a new tube using talcum power to ensure that the tire moved onto the rim properly, and took the beautiful wheel with me to the bedroom to show my wife in the morning. Two hours later, at 3 A.M., we almost died from shock when we were woken by a very loud bang! Another tube had burst!

At that time Karl Wright, an electronic-engineering student, was boarding with us. He was an excellent student (graduated top of a class of over 800) but was equally puzzled. Together we measured and calculated, and destroyed several tubes under reasonably controlled conditions. We saw how, slowly, the tire would bulge and the tube creep out from below and then rapidly balloon. If we were quick enough we could deflate the tube before it burst. We made a series of rims, ever larger, until we could no longer mount the tire. It took more than a month to develop a theory to explain the phenomenon, and another six weeks to prove it. This is what we established.

1. We had assumed that because the flanges are almost 6 mm high, a variation of, say, 2 mm in bead-seat circumference would have an insignificant effect since this would make a difference of only 0.63 mm in the diameter of 622 mm. Wrong! The smallest difference in circumference can be disastrous. Tie a string around a beer can and you will easily slip a matchstick between them. Similarly, if the bead-seat circumference was 1952 mm instead of 1954 mm, there would be sufficient slack for the tire to blow off the rim. This also is the reason why rims work fine with quite a shallow well.

2. Given that the air pressure is equal, the stress in the casing of a narrow tire will be less than that of a fat tire since the force is a function of the cross-sectional diameter. Therefore, if there is a little slack in the rim/tire fit, at some point around the wheel the tire will lift a little. That lifting increases the cross-sectional diameter and consequently also raises the stress in the tire fabric slightly more than elsewhere. This increased tension slowly draws to that region whatever other slack there might be. This may take time, but can be speeded up by dusting the tire/rim interface with talcum.

3. The 25-520 tires for the first wheels had wire beads. The 25-622 fold-up tires that blew off used synthetic beads. We had only one other 622 tire. It had steel beads and worked fine. We found that the tires with synthetic beads could be mounted on a larger rim than the wire-beaded tire. However, we reasoned that that was not where our problem lay. Our problem had to do with the fact that synthetic beads squash. Our rim had a bead-seat ledge that was too narrow for the flattened synthetic bead, so the tire tended to slide off into the well. This action reduces the cross-sectional diameter which allows the tire to be pulled around, usually to the opposite side of the wheel, where a bulge would form and the failure occur.

Remedy

The remedy was twofold. First, we had the extrusion die altered very slightly, to broaden the bead-seat ledges to 2.5 mm. Five weeks passed before the new material arrived. Second, we had to ensure that the circumference of the bead seat was between being exact and no more than 0.5 mm smaller, (1953.5–1954 mm).

Post script

We had other difficulties, so the time delay between needing to beat the cost of imported rims and being ready to manufacture was almost two years. During that time the price of imported rims dropped significantly, aero rims and mountain bikes arrived. IZIZI rims were fitted to the recumbents used by Lloyd Wright for two of his winning rides in the 105 km Argus Cycle Tour, and to Wimpie van der Merwe's recumbent when he set the course record (which still stands) and three IHPVA world records, one of which still stands. Despite these achievements and the fact that we exported rims (Kingcycle), local dealers avoided us saying that buyers wanted a big-name rim. The expected (hoped-for!) swing to recumbents never happened. None of these factors was good for business. We did not make enough money to afford to re-tool to make aero or mountain-bike rims. In retrospect, that is probably what we ought to have done to save the business.

—*John Stegmann*
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Control of hydrofoils using dynamic water pressure

By Alastair ("Al") Taig

Hydrofoils are used on watercraft to provide lift, and/or stability. Generally, foils may be fixed or capable of varying their angle of incidence. Fixed foils may be angled to be part submerged, and part above the water surface, so that as they rise, the submerged area of foil decreases, and an equilibrium will be achieved. But foils which break the surface cause wave drag and suffer from "ventilation" (pulling air down to the upper surface of the foil due to decreased pressure). Thus, fully submerged foils, with some means to prevent them reaching the surface, are potentially more efficient.

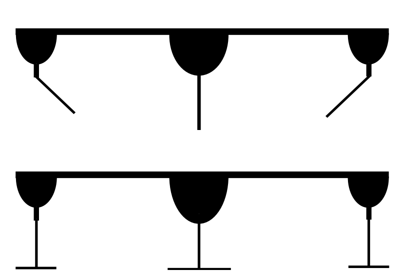


Figure 1. Angled and horizontal foils

Foil actuation

A foil boat generally requires some means to keep the boat from pitching and heeling excessively when lifting. If the foils, mounted on the tip floats, have a variable angle of incidence, they may be adjusted to provide variable lift, independently.

This could be by manual control, requiring a skilled "pilot", or by an automatic system which maintains each foil at a constant depth below the water surface.

Existing, state-of-the-art foil boats (such as the sailboats, *Rave* and Hobie *Trifoiler*) use devices that follow the surface (a kind of water ski on the *Trifoiler*) connected by a mechanical linkage to the adjacent foil. These surface followers provide increased water drag, and are vulnerable to damage.

The following diagram illustrates the proposed pressure-controlled system, in which dynamic water pressure is utilized to adjust the angles of the lifting foils.

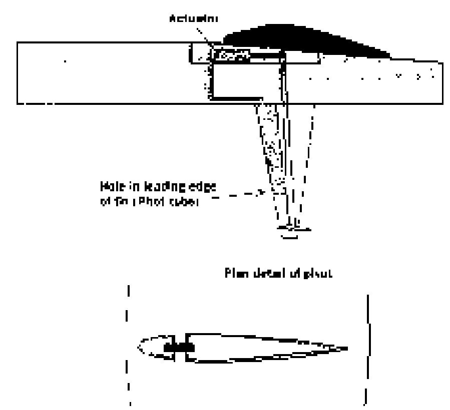


Figure 2. General arrangement of foil strut and pitot-tube location

In this design, a short tube, called a "pitot tube", in the leading edge of the "fin", about 200 mm above the foil, is pressurized by a combination of depth below the surface, and the dynamic pressure due to speed through the water. This pressure compresses the air trapped inside the fin, and is picked up by a bellows (or other sealed type) actuator. This has a piston that pushes on a lever fixed to the hinged foil, as shown. Positive pressure produces a positive angle on the foil, increasing its lift. When the hole reaches the surface of the water, pressure will be lost and the foil angle will decrease. As there will be a time delay as some water enters or leaves the tube, the pressure in the fin and actuator will tend to settle to just maintain the pitot at a "mean" water level. The diameter of the hole in the pitot tube controls the rate of

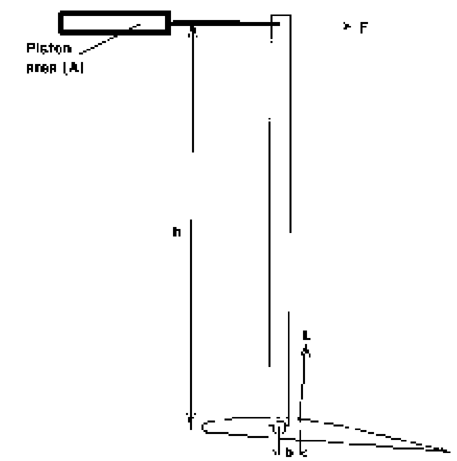


Figure 3. Diagram showing dimensions used in the equations

change of the foil angle, hence the sensitivity to waves. Static water pressure, at the pitot, also acts on the actuator, but the effect is relatively small as the craft rises on the foils.

The position of the foil pivot in relation to the center of lift of the foil determines the force required on the control lever, to increase the foil angle. It is advantageous to pivot the foil a small distance ahead of the lift center, so that the lift will act to reduce the foil angle. This is a stable condition and avoids any tendency for the angle (and hence lift) to increase uncontrollably. If the pivot is close to the lift center, the force required at the actuator will be relatively low and the size of the actuator can be minimized. This is important in order to reduce the volume of the air space in the fin (as shown in the diagram of the system).

The reason a space is sealed in the fin (rather than connecting the pitot tube to the actuator with tubing), is to provide a reservoir of trapped air in the fin to prevent water from entering the actuator. If the system filled with water, the reaction rate of the mechanism would be slow, and the static water pressure at the pitot tube would not assist in pressurizing the actuator.

Balance of foil forces

Foil lift (L) acts at a distance 'b' behind the pivot center

Actuator force (F) acts on a lever of length 'h'.

The moments of these forces must balance for equilibrium.

$L.b = F.h$

But, $L = 0.5\rho V^2 SC_l$

That is $L = 500V^2 SC_l$ (1)

where L is lift, Newtons ρ = water density, = $1000\,kg/m^3$

S is area of foil (sq. m)

V is speed (m/sec)

C_l is lift coefficient of the foil

And $p = 500V^2$ (2)

p is dynamic pressure, N/sq. m

Also $F = pA$ (3)

A is actuator piston area, sq. m.;

But $L.b = pSC_l\, b$ and $F.h = p.Ah$

Hence $SClb = Ah$ (p cancels on both sides of the equation)

Or $Cl = Ah/Sb$ = constant depending on the dimensions.

This implies that the foil lift coefficient will remain constant until the pitot tube reaches the surface (when p decreases).

The system performance can be modified by a return spring, which holds the foil at its minimum angle until the speed is sufficient to pressurize the actuator to overcome the spring. This arrangement is preferred, as the craft has less drag with the foils at minimum angle, and will reach "foil speed" more easily.

In this case:

$L.b = (F - P).h$ (4)

where P is the spring force acting at the actuator.

The factor, $500V^2$ does not cancel in this case

$P.h = F.h - L.b$

$P.h = pA.h - pSC_l.b$ (5)

And $p = 500V^2$ eq. (2)

Equations (1) (2) (3) and (5) can be used to find the proportions of the specific foilcraft.

Step 1.

Knowing the design weight (W) of the craft, assume that L is 0.5W

A practical maximum value for Cl is assumed as 0.8.

The design lift-off speed can be used to determine the foil area, S, from eq. (1)

Pressure, p, for this speed is found from eq. (2).

Step 2.

Assume a speed at which the foils should begin to provide lift. This must be within the fully immersed speed capability of the craft. At this speed, F = P, and $C_l = 0$. Calculate the pressure, p_0, at this speed, from eq. (2).

Then $P = p_0 A$. (The piston area, A, is not known at this stage.)

Step 3.

Using eq. (5) at lift-off speed, the area A can be found by substituting p_0A for P.

Sample calculation

Step 1.

Craft weight, W = 118 kg (260 lbf)

Foil lift, L = 59 kg = 579N (130 lbf)

Lift-off speed, V_l = 4.9m/sec (16 ft/sec)

$L = 500V^2SC_l$ ($0.97V^2SCl$) eq. (1)

S = 0.0603 sq m (0.654 sq ft) at Cl = 0.8

p = 12005 N/sq m (248 lbf/sq ft)

Step 2.

Assume speed when foils start lifting, V_0 = 3 m/sec (10 ft/sec)

p_0 = 4500 N/sq m (97 lbf/sq ft)

P = 4500A N (97A lbf)

Step 3.

Using dimensions

h = 0.61 m (24 in); b = 0.0127 m (0.5 in)

$P.h = pA.h - pSC_l.b$

$4500A \times 0.61 = 12005A \times 0.61 - 12005 \times 0.0603 \times 0.8 \times 0.0127$

Giving, A = 0.001607 sq m (2.49 sq in)

Return spring preload,

P = 4500 × 0.001607 = 7.23 N (1.63 lbf)

(The above numbers apply to a sailboat being developed by the author.)

This example shows that a practical design can be achieved, using dynamic pressure to operate the hydrofoils.

For the design of a human-powered boat, the lift-off speed will probably be less than the 4.9 m/sec used in the above (sailboat) example. A speed of around 3 m/sec would be more reasonable, and when applied to the above analysis would result in larger-area foils. But the utilization of dynamic pressure for actuation is still feasible.

Author notes

I built my first sailboat about nine years ago, and the angled foils were intended for stability rather than full-lift capability. They enable a sailboat to carry more sail-power than otherwise. The three later craft have lifting capabilities, but rarely get fully foilborne on my home lake, due to fickle winds and weeds. However, the latest, pressure-controlled foil system has been proven to work on the current (#4) boat. Even in the no-lift configurations the boats are fast and meet my goal of being the fastest sailboat on the lake. The name "Alf" comes from that crazy TV extra-terrestial who liked to EAT CATS.

By the way, I am a retired engineer, with a career devoted mainly to automotive steering and brake development.

—*Alistair Taig*
<alistair.taig@gte.net>

Project review

CHicK-2000 Project Team "Active Gals"

Remarkable achievement of ActiveGals HPA team in Japan

(from a communication from Toshiaki Yoshikawa)

This note gives some details of the human-powered aircraft "HYPER-Chick KoToNo Limited" built by the team "ActiveGals" in Japan, and sent by the team's leader Toshiaki Yoshikawa (letter, 26 March 2001). Mark Drela's review of the remarkable achievements of the team follow this note.

The technical data are shown in the drawing. The photographs show the plane itself and some of the team members, including the pilot Kotono Hori, who successfully made the first FAI I-C class human-powered flight in Japan in 1992.

On 4 and 5 November 2000, the team made the first flight of an HPA with stressed-skin construction.

Both the I-beam spars and the styrene paper mentioned in Mark Drela's review were reinforced with carbon fiber. The result was an aircraft that could fly (at a height of 2 meters) needing only 160 watts of power input to the pedals, a world minimum for an HPA.

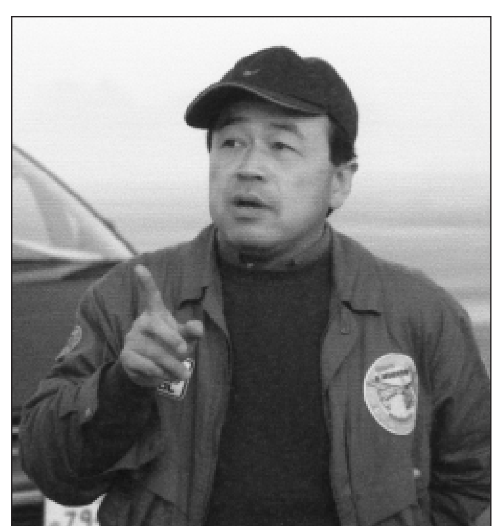
Yoshikawa wrote, "It has a composite structure, CFRP on spar and GFRPed styrene paper on skin." He wrote also that the team is "working to realize a new circling method," described thusly: "The new circling method is by twisting the flexible wings during banking by applied aeroelasticity.

"The twist of the right wing is applied in the opposite direction of that of the left wing. This has been found to reduce power loss during the HPA's turn."

Circling flight is difficult because of the greatly increased power losses and the control difficulty in the turns. (The "inside" wing goes much slower than the outer wing and tends to lose lift.) Stressed-skin construction allows the use of wing-warping (in opposite directions) during the turn. It also greatly reduces wing deflection and permits the use of a very high aspect ratio, 43.7, further reducing the aerodynamic losses.

The aircraft is on display at the Kakamigahara Aerospace Museum.

—Dave Wilson



In flight, above, with pilot Kotono Hori (left).
Right: Project leader Toshiaki Yoshikawa
Opposite: Working on one of the wings of the craft—and the technical chart.

Photos and chart, CHicK-2000 team

Review by Mark Drela

The CHicK-2000 human-powered aircraft by the ActiveGals group has a number of notable features.

The wing structure employs a stressed skin which provides the necessary torsional stiffness in addition to its usual duties of forming the airfoil contour. The most common approach has been to rely on a tubular spar to provide all the bending and torsional stiffness, with secondary foam sheeting and a thin Mylar wing skin providing the airfoil shape.

Using the stressed skin for torsion instead allows the use of a full-depth I-beam spar to provide the bending stiffness. The I-beam spar is a far more efficient bending member than the tube spar, and hence provides a stiffer and stronger wing for a given weight.

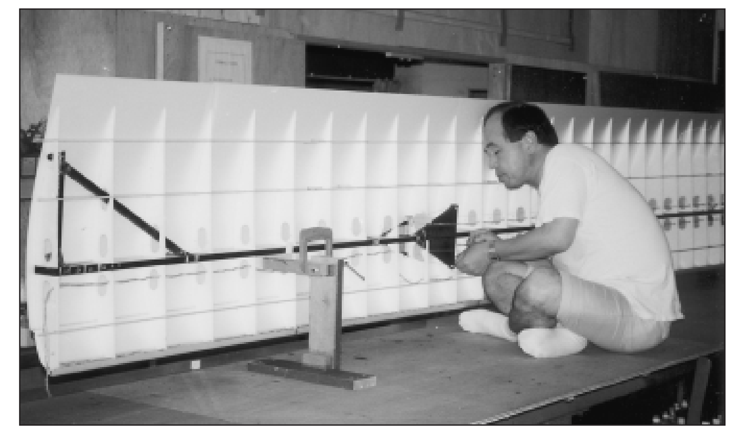
Not surprisingly, the wing-tip deflection of the CHicK-2000 under load is amazingly small considering its low empty weight of 31 kg and its immense wing aspect ratio of 44. The high aspect ratio obviously contributes to the modest specific flight power of 3.6 W/kg pilot mass, despite a fairly high wing loading of 46 Pa which gives a rather fast cruising speed of about 8 m/s. Low power coupled with high speed gives the potential for large range, and also gives the ability to handle windier conditions than more lightly-loaded HPAs.

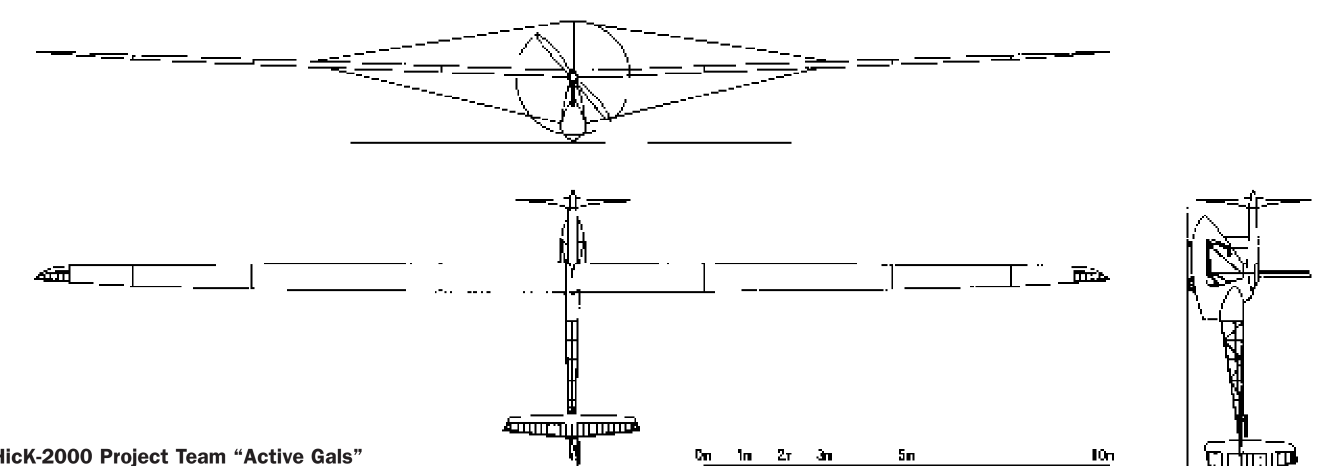
One practical disadvantage of a stressed-skin HPA structure is that common construction materials such as polystyrene foam do not have a sufficient shear modulus for the task. The ActiveGals group appears to have solved this problem with their fiberglass-reinforced styrene paper.

The stressed skin is also very demanding of design details and construction quality to preclude local buckling or failure. Again, these problems appear to have been surmounted as the aircraft is clearly structurally sound. Construction photos reveal meticulous craftsmanship.

Other reported innovations include the use of aeroelastic effects to twist the wings for roll control. Judging from the type of control yoke, the pilot appears to have full three-axis control of the aircraft, although it is not clear how the wings are twisted in practice.

—Mark Drela, MIT professor of aeronautics and astronautics, Massachusetts Institute of Technology (principal designer and constructor of several MIT HPAs).

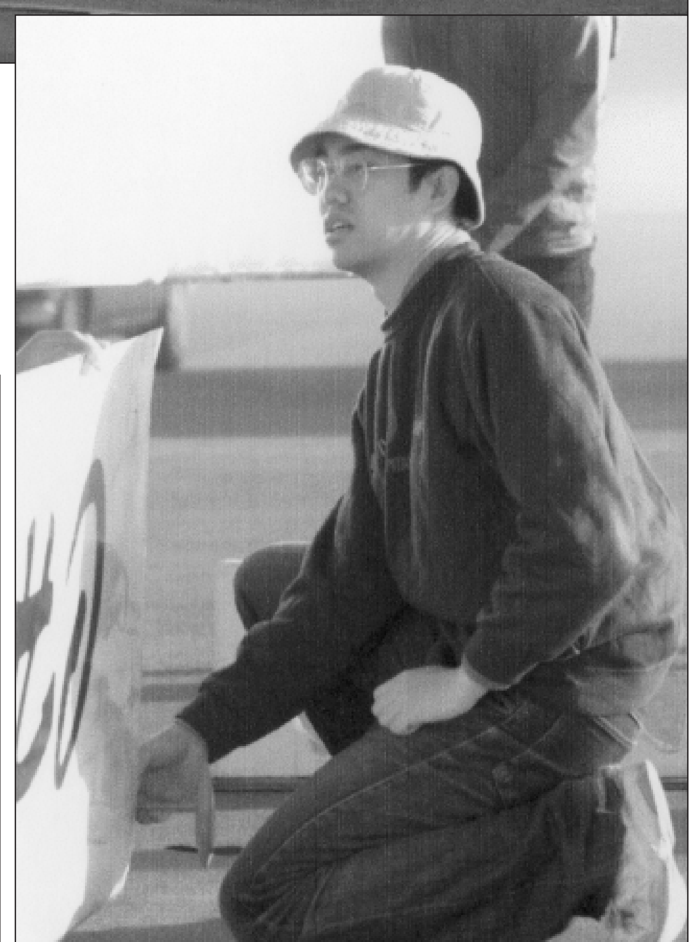




CHicK-2000 Project Team "Active Gals"
6-36-11 Suzuhara-cho
Itami city, Hyogo 664-0882, Japan
{no e-mail address provided]

CHicK-2000 TECHNICAL DATA. Pilot : KoToNo HoRi

Span	26.6 m	Length	7.12 m	Height	3.04 m
Wing area	16.2 sq.m.	Aspect ratio	43.7	Propeller	2.85 m dia.
Empty weight	31.0 kg	Flying weight	75.0 kg	Wing loding	46.3 N/sq.m.
Min.flying speed	7.2 m/s	Min.power at speed	160W @ 8.0m/s	Max. glide ratio	1:48
Airfoil	Wortman FX76 MP-160 ~ DAE-21 ~ DAE-31 ~ DAE-51				



Above: A closer view of the cockpit and propeller of the CHicK-2000 aircraft. Right, Takashi Hattori, right-wing runner; below, Kouta Sata, left-wing runner.

Book review

RICHARD'S 21st CENTURY BICYCLE BOOK(S) by Richard Ballantine.

reviewed by Dave Wilson

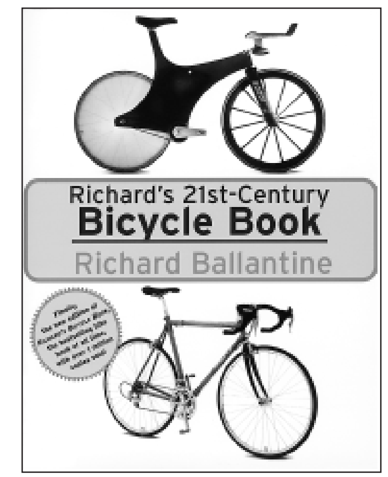
This book is two books, or one book in two versions. One is for non-North-American readers; and was published by Pan Books (Macmillan) in Britain at the end of 2000. Earlier editions came out in 1972, 1975 and 1989. It is a very successful book: one of the messages on the cover states "…the best-selling bike book of all time, with over one-million copies sold!" As I wrote this I was about to leave for Norwalk, CT, for the June first launching of the North-American version with the author himself.

Before I wax too enthusiastic about the book(s) I should confess my biases. I first met Richard Ballantine in 1980 in Bremen, Germany, at a bicycling conference called "Velo-City". I had brought along one of the first Avatar 2000s, which received a great deal of favorable publicity. (We hadn't patented it in Europe, and several rather faithful copies were subsequently manufactured by some new enterprises that did well with them.) Richard had already bought an Avatar, and put it (with a rather wild British female model) on the cover of "*BICYCLE*", the British magazine he published and edited, as "the bicycle of the future." He felt that it could also be fast, formed the Nosey Ferrett Racing Team, recruited Derek Henden, who made several full fairings for it, and named it the "Bluebell". This went on to win the IHPVA World Speed Championship in the US and many races in Europe over several years. Therefore I start by being biased in favor of Richard Ballantine. He did a great deal for the Fomac-made Avatars, and for recumbents and HPVs in Britain and Europe.

We are an "odd couple". Richard is an American who has lived in Britain for decades; I am an ex-Brit who has lived in the US for decades. We both do what we can for bicycling and HPVs. Richard has been more dedicated and more successful.

The books differ mainly in the use, respectively, of American English and British English, including some translations of slang. Some examples slipped through. For instance, how many American readers would know what to expect if an HPV were classed as "dodgy"? (Roughly it means that it wouldn't be a good bet.)

The British version of the book has, as the sole representative of cycling on the front cover, Richard's daughter in a carbon Windcheetah tricycle HPV with a lot of advanced components. We must give Richard some of the credit for the publisher's belief that a bike book with an HPV on the cover was not going to put people off buying it. The U.S. publisher, The Overlook Press (Woodstock and New York), apparently felt that doing this in North America would be too risky, and I think that many of us would agree, with some sadness.



U.S. version of Richard Ballantine's book.

Inside the books have many similarities: there are 22 chapters having the same titles, starting with "Get a bike!" to "Done!" All the chapters are written with a breezy enthusiasm coupled with a deep knowledge of the field and an instinct for telling people, from raw beginners to seasoned enthusiasts, what they want to know.

Of particular interest to HPVA members is chapter 5: "Zzzwwaaaammo!", 27 pages devoted entirely to extolling HPVs, and, on a quick scan, having more illustrations than any other chapter. That alone sets Richard's book well apart, (i.e., well ahead) of all competitors.

You will enjoy this book. The British version has the ISBN number 0 330 37717 5; it costs UK£16.99. The North American version has ISBN 1 58567 112 6; I bought my copy from the Overlook Press, Lewis Hollow Road, Woodstock, NY 12498, for US$28.50 including P&P; the bookstore price should be about US$18.00.

—David Gordon (Dave) Wilson
<dgwilson@mit.edu>

Letters

Danny Too responds to two letters about his article (with Chris Williams), "Determination of the crank-arm length to maximize power production in recumbent cycle ergometry" (*Human Power* 51, Fall 2000).

Battle Mountain crank arms

Matt Weaver

In light of crank-arm length, I have a few observations on the recent article on bicycle cranks [by Danny Too and Chris Williams]. It took me a moment to deduce what was actually tested, but if cranks are of any interest to you, I find it important to note and relay the following.

1. The test was a variable-rpm, "fixed-torque" test: relatively light pedal force, proportionately lighter for longer cranks; riders "gave their all" (maximum exertion) for 30 seconds. Cadences reached high rates (>170 rpm) for the shortest cranks, and modest rates (135 rpm) for the longest cranks; cadences dropped to the low 80 rpm range for short cranks, and low 90 rpm for longest cranks in the final five seconds. Calculated power output was proportional to cadence: the faster you can spin, the more power you get (fixed torque, and flywheel inertia ignored).

2. The torque decided upon was referred to as the "appropriate load" or "85 g/kg of subject body mass" (apparently total mass, not lean or leg-muscle mass).

3. I'm not sure what "appropriate load" is, but it can be deduced. The apparatus was as follows: a 52/14 single-chain drive to a flywheel with a 1.615-meter circumference, with a friction belt of known net tension wrapped about it. That's roughly a 0.5-meter diameter (20-inch) flywheel, (mass/inertia not given)—comparable to moving the belt tension force a distance of 6.0 meters for every revolution of the cranks: effectively a fixed-mean crank torque = 0.8 N.m per kg total rider body mass. (0.27 ft.lbf per pound total rider weight) (ignoring flywheel inertia).

For example, for me ("85g/kg" belt tension mass, and 80kg rider body mass, 175 mm cranks): belt tension = 6.8 Kg×9.81m.s–2 = 67 N (15 lbf); pedal tangential force = 364 N (82 lbf); or roughly 1/2 to 1/3 my mean pedal force normally developed during a sprint (30, and 5 sec, respectively).

4. Test starting rpm (at t=0) is not given, except that it said the ergometer flywheel was accelerated until "inertial resistance had been overcome."

5. Inertia still exists, and cadence varied between start rpm, 174 rpm and 82 rpm during 30-second test. Power sink/source from flywheel during rpm changes may or may not have been significant depending on flywheel inertia (not given, except for the ergometer model number).

Some observations about the experiment.

1. The "appropriate" fixed torque chosen was relatively low for a "healthy male" 30-second cycling sprint bout

2. Pedal cadence given maximum effort shot into the uncharacteristically high 170 rpm range (for short cranks) and swung through a nearly 2:1 cadence range.

3. Longer cranks, in spite of lower pedal force, yielded substantially lower peak pedal cadences (135 rpm) and consequently lower peak power.

4. Interestingly, longer cranks were turning faster at the end of the 30-second bout.

5. Unlike the experiment, real cycling consists of relatively steady cadence with mean pedal force varying with fatigue state of muscles.

6. It seems to me the following occurred: light pedal force reduces to a "spinning" contest, the fixed crank torque specified in the experiment was "light" for a sprint; peak power goes to the fastest spin in this experiment (with relatively light load, it is typically easier to spin rapidly unusually short cranks (110 mm) than unusually long cranks (265 mm)).

This is can be considered from experience, from neuromuscular adaptation/control given untrained subjects, and muscle group force/velocity curves. The final (slowest) cadence was likely lowest for the short cranks due to greater utilization/exhaustion of rider muscles using short cranks, and less-exhausted state of rider legs given long cranks and inability to generate as much work during the 30-second bout (i.e., short cranks wind out and then fatigue; long cranks lumber along awkwardly with less cadence variation for the brief 30 seconds).

7. Conclusions from the experiment as it relates to cycling:

I would not conclude in any way that short cranks are preferred for peak power; I would not conclude in any way long cranks are better for longer events (especially based on a brief 30-second maximal and large cadence variation anaerobic bout). I would suspect that if a sufficiently high "appropriate-load" torque were chosen, the experiment observations would reverse, with the longer cranks superior for peak power. My primary conclusion, honestly, is simply that a group of maximally-exerting healthy guys can spin the relatively light, arbitrarily-specified mean crank torque (0.8 N.m per kg total rider body mass) to higher cadences with unusually short cranks (174 rpm, 110 mm) than they can with unusually long cranks (265 mm, 135 rpm), and they are likely more fatigued after 30 seconds with the shorter cranks).

8. "Optimal" crank length and cadence is indeed important for serious cyclists, and critical for racers, as noted in the study.

9. To come to some useful conclusions about power/crank lengths. I would perform tests specific to the nature/duration of interest (e.g., for a kilo sprint, a simulated-inertia ergometer or real bike with different cranks/gearing and a stopwatch—perhaps already done); (e.g., for hour/long rides, various fixed cadences, and rider-chosen cadence, and measure total work and ideally O_2 uptake/CO_2, HR, etc.).

10. Given such fixed cadence data, a 2D cadence/power (aerobic and anaerobic bouts) "map" for different crank-arm lengths and cadence could be generated and would be greatly valued by many cyclists. Such maps of course are dependent also on muscle type, limb lengths, etc.

11. Some crank experiments:

In my garage in 1995 I prepared some tools to discover such basic information. I built a computer-controlled cadence ergometer to explore my 152-mm cranks. Cadence was regulated precisely by digital feedback control to an electromagnetic brake, and instantaneous torque/crank angle via a load-cell rig was logged as well. No matter how hard or easily I pedaled, the cadence remained fixed at its set point. I had great fun for several days stomping on the pedals and listening to the controller magnetics hum and surge with each leg stroke.

I had validated the velocity uniformity. But, during my enjoyable pedaling, a loose wire shorted and destroyed the power-electronics circuit. Other than logging a few torque profiles and discovering that my left (dominant, but knee-operated) leg is hopelessly weaker than my right, I ended up resuming crank-power testing by utilizing hill-climb tests, because of limited time.

More recently, I found something interesting about cadence. I was invited to test my output on an ergometer at the home of John Howard. I had time for only a 30-minute ride, but discovered quite dramatically how, beyond a certain increase in cadence, my "perceived exertion" rose significantly while my power output dropped simultaneously. I averaged 420 watts for the test, for which I was delighted, yet I would drop below 350 watts with a mere 10-rpm cadence increase over what appeared most productive. Such observations make the use of recently available tools like the "Tune" hub/downloader or the SRM meter (though crank-length changes may be a problem there) critical for racing cyclists.

If I had my wish, I'd have access to a Tune hub and get to know myself better! I hope enthusiastic and endeavoring researchers like Danny Too continue their quest in providing all the detailed studies necessary to fully and truly map out the relation of crank length and rider position to real cycling-performance characteristics. Good cranking!

—Matt Weaver <weaver@e2000.net>

Response to Matt Weaver

I wish to thank Matt Weaver for his comments and observations regarding my article in the last issue of *Human Power* 51 (Fall 2000) on crank-arm length. Matt Weaver's comments (#1–5) and observations (#1–11) above regarding my article on crank-arm length are correct and well summarized. However, I would like to comment and expand upon his observations.

This particular study was used to determine how power output/production (as measured/determined by pedal cadence or flywheel revolutions) changes (or the trend of power production) with changes in crank-arm length for a fixed load. The load selected (85 g/kg of each subject's body mass), although relatively low for "healthy male" 30-second cycling sprint bout (as stated by Mr. Weaver), was based on the load limitation of the ergometer (that would still ensure accuracy) with the heaviest subject tested. Based on the literature available with upright ergometers, there is an interaction between pedaling cadence, load, and power output. With the addition of another variable (crank-arm length), one would expect that there will be an interaction between crank-arm length, pedaling rate, load, and power output. Therefore, Mr. Weaver is correct in stating "that if a sufficiently high appropriate load torque were chosen, the experimental observations would reverse, with the longer cranks superior for peak power." In fact, this is what has been observed, based on the trends from data I had collected on females, examining the interaction between crank-arm length, load and power output in the same recumbent position. Females were selected because they are not as heavy (nor as powerful) as males, and the highest load used and tested (165 gm/kg BM), were within the maximum load capacity of the ergometer. The data and results from this study will be submitted for publication in some future issue of *Human Power*. Based on this one study and the delimitations and limitations of it, Mr. Weaver is correct in stating that "Unlike the experiment, real cycling consists of relatively steady cadence with mean pedal force varying with fatigue state of muscles" and that he "would not conclude in any way long cranks are better for longer events (especially based on a brief 30-second maximal and large cadence variation anaerobic bout)." To address these issues, I have data collected on power output and time to exhaustion, when different crank-arm lengths are used with different pedaling cadences, and incrementing workload until exhaustion (or when the selected cadence can no longer be maintained). I have not yet analyzed the data or the trends associated with it. That data will also eventually be submitted for publication in a future issue of *Human Power*.

In conclusion, there are clear limitations as to the information that can be obtained from a single study, as well as how the data are interpreted or can be interpreted. It is obvious that the data obtained in the laboratory are not always "specific to the nature/durations of interest" as noted by Mr. Weaver. Unfortunately, that happens to be the nature of the beast (research). Research to be undertaken properly and correctly is slow and tedious, needs to be done under an environment where all conditions and variables are very carefully controlled and accounted for (while the experimental variable is manipulated and tested), and is not always going to be "specific to the nature/durations of interest". However, research does produce empirical data that provide information and direction regarding how performance may be enhanced in the real world.

—Danny Too <too@brockport.edu>

Crank-arm length and leg length/proportions?

John Stegmann

Over the years Danny Too and co-researchers have taken the trouble to measure and explain things that most of us interested in recumbency would like to know but tend to rely on intuitive guesses. Their research into the effects of crank-arm length on power production has left me wishing to know why there was no discussion on the relationship between crank-arm length and leg length/proportions?

Cyclists have (always?) imagined that people with short legs (short femurs?) will be happier with short cranks, and long-legged cyclists with long cranks. The cyclists used for the test varied in height from 1.72 m to 1.88 m, and there must surely be the possibility of greater variation in their leg configurations? I now wonder if there is evidence to support this old notion. A century ago Archibald Sharp (*Bicycles and tricycles*, p. 266) considered the speed and motion of the cyclist's knee-joint and wrote: "The shorter the crank, in comparison with the rider's leg, the more closely does the motion of the knee approximate to simple harmonic motion; with simple harmonic motion the polar curve is two circles." From this it would appear that the near-circular motion produced by shorter cranks would favour higher pedaling speeds. Plotting the Too and Williams maxped and minped figures from Table 1 on the graph in figure 2 (p. 4), using a scale of 1000 W=145 rpm, the power and rpm curves are an almost perfect match. The crank-arm lengths in the study varied exceptionally; far more so than the cyclist's legs. Yet the results show that shorter cranks allowed higher rpm, and higher rpm produced greater power.

—John Stegmann <recumbent@cybertrade.co.za>

Response to John Stegmann

Stegmann: "Their research into the effects of crank-arm length on power production has left me wishing to know why there was no discussion on the relationship between crank-arm length and leg-length/proportions?"

First, we do have the data on leg lengths and leg-length/proportions (as well as other anthropometric data) on all subjects tested. Second, we did not discuss it because: (1) we did not examine the leg-length/proportion data in the study (since that was not the focus of the study), and even if we did, that would be a different topic and study altogether; (2) discussion of leg length and proportions would have detracted the reader from the "meat" and trends found in the study; (3) leg-length/proportions would have been expected to randomly vary (as would height and weight) for the subjects tested, and without having selected (or matched) equal number of subjects for different leg-length/proportion, the discussion based on the results could be biased and provide inaccurate or misleading information; and (4) discussion of leg length and proportions may result in more equivocation and confusion than clarity. Regarding this last point, we do not believe it is the leg length or leg-length proportions that is important, but rather, it is the hip, knee and ankle angles that results from an interaction between crank-arm length and leg length (or leg-length proportions) to maximize power production and minimize fatigue that is important. The reason for this statement? There is no theory or theoretical basis to explain or justify why differences in leg lengths or leg-length proportions should result in greater or lesser power production with different crank-arm lengths. However, there is a theoretical basis for why some joint angles (hip, knee, ankle) will result in greater force/power output, and how these joint angles may be produced with different combinations of leg length and leg-length proportions interacting with different crank-arm lengths. Based on the tension-length and force-velocity-power relationships of contracting muscles, there is/are some hip, knee and ankle angle(s) (or joint range of motion) that will maximize power production and performance, and that there is an interaction between crank-arm length, pedaling rate, and load. What is/are this/these optimal joint angles to maximize cycling performance? This has not been determined yet because of the difficulty in manipulating, reproducing and then testing various combinations of joint angles with subjects of different leg lengths and leg-lengths proportions. Even if the optimum joint angles were determined, to obtain these optimum joint angles may result in crank-arm lengths that are similar (or different) for subjects of the same leg length with different leg-length proportions, or of different leg lengths with the same leg-length proportions. This would imply (and possibly conclude) that the optimal crank-arm length is very individualized, and dependent on, both the leg length and leg length-proportions of the cyclist (in addition to other factors such as pedaling rate and load). This would not be a very satisfying answer to those looking for a quick and simple solution to a very complex problem. In some of my previous investigations, I was able to manipulate hip angles (and observe the changes in cycling performance), while maintaining the same knee angles, by adjusting the seat-tube angle in conjunction with the seat-to-pedal distance. This provided information regarding the optimum hip angle to produce power when interacting with a certain knee angle (and helped explain why power output in some recumbent positions is greater than in upright cycling positions). We have also examined the effects of changes in seat-to-pedal distance on joint angles and on power production (which we will be submitting to *Human Power* for publication). However, this interaction between hip and knee angle to maximize power production is much more complex when crank-arm length is systematically manipulated because a change in crank-arm length affects both the hip and knee angle simultaneously. This makes it very difficult to determine whether the changes in cycling performance when manipulating crank-arm length is primarily attributed to changes in hip angle, knee angle, or both (the net effect of this interaction) especially since this involves multi-joint muscles acting on multiple joint segments simultaneously.

Stegmann: "Cyclists have (always?) imagined that people with short legs (short femurs?) will be happier with short cranks, and long-legged cyclists with long cranks. The cyclists used for the test varied in height from 1,72 m to 1,88 m, and there must surely be the possibility of greater variation in their leg configurations?"

It is very possible that there is greater variation in their leg configurations. But again, it is probably not the actual leg configurations that is as important as the leg configurations that will result in the hip, knee, and ankle angle that will maximize power production when interacting with some given crank-arm length. And there are probably as many different leg configurations with crank-arm-length combinations that would result in the optimal joint angles to maximize power production as there are cyclists. Hypothetically, if a study was conducted with very tall people (long-legged cyclists) versus very short people (short-legged cyclists), I would suspect that similar curvilinear trends for both groups would be found for power output with incrementing crank-arm length. However, the crank-arm length that would maximize power output would probably be different for the two groups (depending on the load used, pedaling rate, and fatigue level) because the same crank-arm length would result in different joint angles (that may or may not be optimum) for the two groups.

Stegmann: I now wonder if there is evidence to support this old notion. A century ago Archibald Sharp (*Bicycles & Tricycles*, p. 266) considered the speed and motion of the cyclist's knee-joint and wrote: "The shorter the crank, in comparison with the rider's leg, the more closely does the motion of the knee approximate to simple harmonic motion; with simple harmonic motion the polar curve is two circles." From this it would appear that the near-circular motion produced by shorter cranks would favour higher pedaling speeds.

I don't know if there is evidence to support the notion mentioned above, but I would agree that shorter cranks would favour higher pedaling speeds (if minimal loads are used). However, as the load increases (and continues to increase), the pedaling speed (at some point) will start to decrease. At this point, longer cranks would be favoured to minimize fatigue and to maximize power output. If you are interested, there is a paper we had published (with data involving upright-cycle ergometry) that included a discussion of the interaction between crank-arm length and pedaling rate, and its effect on the kinematic and quasi-static moment contribution to the total joint moments to affect the joint-moment cost-function minimum. The reference for the paper is: Too, D. & Landwer, G.E. (2000). The effect of pedal crank-arm length on joint angle and power production in upright-cycle ergometry (*Journal of Sports Sciences* 18:153-161).

Stegmann: Plotting the Too & Williams MAXPED and MINPED figures from Table 1 on the graph in Figure 2 (p. 4), using a scale of 1000 W=145 rpm, the Power and rpm curves are an almost perfect match. There should be a very close match between MAXPED and MINPED from Table 1 with Peak and Minimum Power from the graph in Figure 2, since MAXPED and MINPED were determined from the flywheel revolutions for a 5-sec interval, as was Peak and Minimum Power (which was then used to generate a regression equation for Peak and Minimum Power).

Yes, the crank-arm lengths in the study varied exceptionally; far more so than the cyclist's legs. Yes. The crank-arm length was deliberately selected so it would vary exceptionally far more than the cyclist's legs. The reason? We wanted to examine the extreme ranges of crank-arm lengths that could possibly be used (and a few in between), in order to determine the trend in power production/output with incrementing crank-arm lengths.

Stegmann: Yet, the results show that shorter cranks allowed higher rpm, and higher rpm produced greater power.

Yes, that statement is correct, but only for the load used in that study (85 g/kg of each subject's body mass), since there is an interaction between pedaling rate, load, and crank-arm length. If the load is increased and continually increased, at some point, pedaling rate will decrease resulting in a decrement in power. At this point, a longer crank-arm length will be more effective in producing power. We have data to support this, and will be submitting a paper for publication in a future issue of *Human Power*.

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EDITORIALS

A bit of history viewed from Eastern Europe

by Marek Utkin M.A., Poland

In what must now be considered historical, a bicycle-design contest was organized by Dave Wilson in the magazine *Engineering* over 33 years ago. Two successful Polish entries became known through publicity in the technical HPV and cycle literature (e.g., in *Bicycling Science*).

One of the entrants, Kazimierz Borkowski, creator of the sliding-seat rowing-action bicycle, claimed that "it's just for sports and recreation". He was an engineer in the biggest Polish bicycle plant, Romet, and never developed his concept further. Nowadays Rowing-bikes, very sophisticated (and practical) machines, are made by Derk Thijs in The Netherlands.

Another entrant, Stanislaw Garbien, developed a modern SWB recumbent with front and rear suspension, drum brakes, optional fairing and variable-gear constant-ratio swinging cranks. Now, after the collapse of the Iron Curtain, I can reveal that the designer of this bike was working as an aircraft armament engineer, a really top-secret occupation. I got in touch with him when he retired in 1987, and he was still trying to develop a new kind of bike drive-train. The prototype of Stanislaw Garbien's bike was in use for a long time, but in about 1980 it was converted into an upright bicycle with conventional drive and with the front beam serving as a luggage carrier. This conversion from recumbent to upright is in some way significant. Eastern Europe (including Poland) is very fashion-sensitive. It's not easy to be a trendsetter there: if the item is not of Western origin, it is usually ignored or laughed at.

A young engineer, Jacek Ziolkowski, who made his three-wheeled HPV from a MiG-17 auxiliary fuel tank (*HPV News*, Sept. 1987), died after being hit by a car while riding his conventional bike. It's a great irony: if he had ridden a recumbent, he would have probably survived, maybe with a broken leg. His vehicles are now in Warsaw's Museum of Technology.

At the end of the 1980s and the beginning of the 1990s in Poland it seemed that HPVs would find their niche, but after the "opening to the West," cars became much more affordable, and the mountain bike appears to be the only bike on the market. The popularity of practical HPVs is inversely related to the availability of cars. This can be confirmed by looking at France and Britain in the 1920s, Sweden during and shortly after WW II, or Poland and Russia in the 1980s, when all sorts of pedal cars and HPVs were made by amateurs and also by small manufacturers.

It is difficult to fight the car, but there is another medium that helps HPV enthusiasts to communicate and not to feel alone: the world wide web. May human power be with you!

—Marek Utkin
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The future of HUMAN POWER

This heading is deliberately ambiguous. It refers to the movement as well as to the journal. There are discussions underway on a new basis for the international association, and therefore for the journal. Here is a concise summary of the present position just written by Richard Ballantine, who is taking an active part in trying to ensure a healthy continuation of both.

"The IHPVA was founded as a US-based organization in 1975. HPV clubs from other countries joined the IHPVA as chapters, of equal status with chapters based on US states or regions. Over time, the non-US clubs grew in size, and eventually came to want independent status and an international organization that was truly democratic rather than US-dominated.

"Following the Lelystad Declaration in 1995/96, the IHPVA was reorganized in January, 1998. The original IHPVA became the North American HPVA. HPV clubs from other countries became autonomous, and a new IHPVA was formed, comprised of representatives of clubs and groups from various countries. Yet the work of the IHPVA, which is the responsibility of the member clubs, continued and continues to be performed almost exclusively by North American HPVA members, who do the record-keeping, maintain the web site, and produce *Human Power*, all on a volunteer basis. At the Brighton meeting, HPVA representatives politely said that this cannot go on – other IHPVA member countries need to do their fair share of the work.

"The US IHPVA representative, Paul Gracey, raised this matter in a recent post to the IHPVA Board:

'The one aspect of the IHPVA that is ongoing day-to-day and seems to me to be under-appreciated is/are the Web and Internet services. Our visibility to the world is embodied in those services located in cyberspace and the publication *Human Power*. Like everything else that is related to the Internet, the bloom of newness may be fading and this may at some time need to be put on a compensated basis. This, and the caring for records reports and other archival materials is the major reason it may yet be desirable to try to establish a physical location and some sort of endowment to see to its care.'"

Voluntary submissions to *Human Power* have fallen, and most of my many letters to people in many countries asking for contributions are unproductive, so that our publication frequency has dropped to semi-annual from quarterly. Another long-term editor of HPV-related magazines told me recently that he realized soon after he started that in this field the editor has to write most of the material. (I've tried to avoid that, except for the reviews.) Another view recently expressed is that, with the internet giving instant information on anything, we don't need paper publications any more. I hope that that is not a general belief. With each new revolution in electronic storage I find that my records of another part of my life are lost forever (I've recently thrown away reels of unreadable computer tapes, and large-format diskettes, likewise inaccessible) and I value paper for archival material more and more.

The message I want to pass along here is simply "prepare for more change, and take part if you wish to influence the direction!"

—Dave Wilson
Editor, Human Power

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