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IN THIS ISSUE

CONTRIBUTIONS TO *HUMAN POWER*

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

The phantom trailer

By Andreas Könekamp

The Institute of Electromechanical Engineering (EMK) at the University of Technology Darmstadt is developing the concept and a prototype of the phantom trailer in cooperation with AKASOL Darmstadt (a student project that produced "Pinky", a vehicle that won the Tour de Sol three times). The pushing and pulling forces of this trailer will be completely compensated by a self-controlled electric drive ("inverse over-running brake").

INTRODUCTION

Situation #1: A family is touring a hilly landscape by bicycle. As long as the path gently follows the river, dad has no problem hauling the bike trailer and two children inside, weighing about 40 kg together. He is quite fit because he bikes regularly. But when the first big hill appears, the enjoyment disappears: dad and the children have to walk, pushing bike and trailer. After the second or third hill his mind is made up: there will be no more bike trips until the children can ride on their own.

Situation #2: Mom takes the children to kindergarten before work at ten o'clock. On the way back she has to buy beverages and groceries. As often happens, the time is a little short and she pedals a bit harder. When she reaches home dripping with sweat at a quarter to ten, she has two alternatives: to go to the office soaked with sweat or show up late after taking a shower. Thus she will prefer to take the car next time.

The recent success of bicycle trailers, especially trailers for children, shows the willingness of many people to look for alternatives to cars in order to accomplish certain transport tasks without automobiles, like the transport of essential consumer goods and small children who cannot ride on their own. The share these kind of trips have on the whole automobile usage is enormous: On weekdays up to 50 % of all car trips are shorter than 5 km. (see fig. 1).[1]

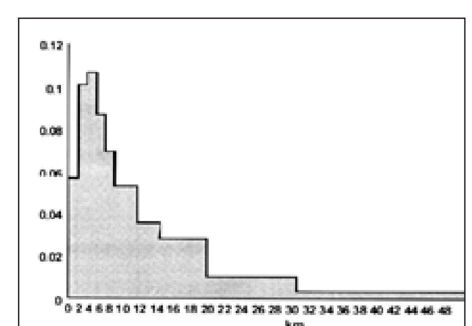

Figure 1. Histogram of the length of car trips (after Hautzinger et al., p. 75)

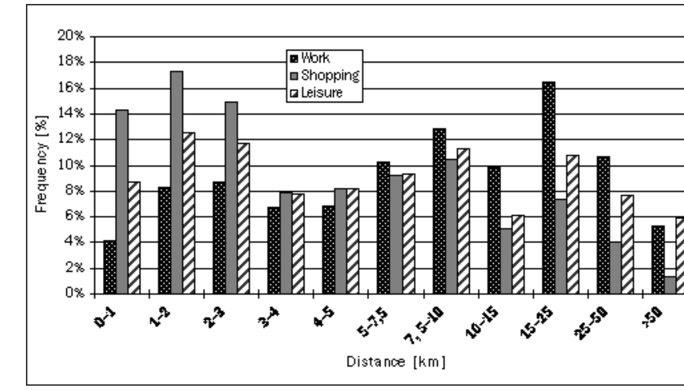

Figure 2. Frequency of automobile trip distances depending on purpose (adapted from Emnid)

Interestingly there is nearly no difference in the percentage of short-distance trips between rural regions (50 %) and populated areas (45 %) (Ueberschaer *et al.*, p. 51). A significant difference, however, can be seen in the distribution of the purposes of the short-distance trips (see fig. 2). While half of all automobile trips to the place of work are longer than 7.5 km, the median driving distance for shopping is only 3 km.

The modular combination of bike and trailer has many advantages: The bike remains fast and manoevrable, especially on short distances, and takes up little space. Many people are willing to put up with the necessary—more or less moderate—exertion to benefit from these advantages and to achieve fitness.

The trailer, which is carried along only if necessary, expands the transport capacity considerably. Nearly all short-distance transport tasks are possible this way (see Burrwitz *et al.*, p. 100). However in particularly hilly terrain, with many stops and starts (traffic lights, crossings etc.), or with considerable head wind, the work load increases significantly. A typical example: a woman (60 kg) riding a bike (15 kg) with a coupled trailer (15 kg) including two children (35 kg) has to expend two thirds more energy to pull that trailer. Many people fear such experiences and are reluctant to use a bicycle trailer.

AN ELECTRICALLY-DRIVEN BICYCLE TRAILER

A bicycle trailer with auxiliary drive offers the chance to avoid the extra work load, so a larger percentage of short-distance transportation could be carried out with bikes (see Neupert, p. 36).

For reasons of environmental protection[2] and of ease of operation, only electric drives are suitable. Unlike many electric bikes having the image of a vehicle for elderly people, this solution preserves the sportive image and the above-mentioned advantages of the bicycle as the towing vehicle.

The primary goal of the trailer concept presented here is to replace frequent short-distance trips by car with the environmental-friendlier ride of a bike and electric trailer. And possibly, it avoids the purchase of a second car. Furthermore the trailer should be suitable for bike tours.

THE TECHNICAL CONCEPT

As described above, the electric drive in the trailer should not replace the biker's muscles, but only compensate for the additional load of the transported weight. In addition, problems with the driving dynamics (behavior in curves!) may be expected. Here, the drive only performs assistance similar to power-assist bikes. This is done by measuring the tractive force in the drawbar with a sensor and compensating for it.

The trailer is also equipped with an automatically controlled brake so it will follow the bike almost without being noticed—like a phantom!

REQUIREMENTS FOR THE PHANTOM TRAILER

Driving performance

The requirements for speed and maximum gradient should match the performance of a typical biker without trailer: Without headwind, a speed of about 20 km/h should be reached on the flat, or about 10 km/h up a 5 % gradient.

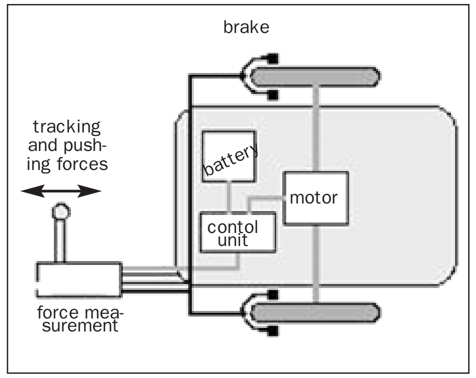

Figure 3. Function structure of the phantom trailer

Two hours' driving duration in a gently rolling landscape should be the minimum and twice this is desirable. The range may be increased by extending the battery capacity or by recuperation, i.e., feeding back some of the braking energy into the battery.

The above performance should be possible with a load up to 50 kg; with higher loads less performance could be accepted.

Safety

The trailer must be safe, especially when transporting children. Besides suitable seats and storage places, adequate safety belts and roll protection are necessary. Furthermore all technical parts (brakes, electric equipment, moving parts and wheels) have to be child-proof. The parts as well as the child passengers and transported load must also be protected from rain, loose chippings, or too much sun. Finally the system has to show fail-safe-properties[3] in any case of a defect (failure of the brakes, electrical fault, etc.). All valid standards and rules (e. g., national traffic laws) should be complied with.

Handling

During the ride itself the Phantom Trailer obviously does not need any manual control, but in all other situations the handling also has to be safe, simple and comfortable. Relating to this, simple coupling to different bicycles, easy battery charging from the grid within one or two hours, and easy loading and unloading, are requirements as is the possibility of safe and space-saving storage.

PRESENT PROJECTS

Two studies concerning the main problems of the phantom trailer are presently under way: measuring the tractive force within the drawbar, including the steering of the braking system and the design of the drive system and control.

Tractive force measurement and brake system

In this project the system for measuring the tractive force is to be developed and integrated into the drawbar. After evaluating different measurement systems, e. g. strain gages, hydraulic-pressure sensors, etc., the most suitable principle will be chosen and a prototype for the Phantom Trailer will be developed.

The braking system will also be developed. For an overrunning brake, a force measurement within the drawbar is also necessary, so it is obviously advantageous to combine both systems, e.g. hydraulically.

DRIVE AND CONTROL

The goal of the second project is to develop the drive and its control. Particularly important is the choice of a suitable motor with high power per weight and high efficiency across a large speed range. While the DC-motors conventionally used for electric bicycles do not fit the second criterion too well, modern drive concepts like asynchronous or switched-reluctance motors with electronic rotating field generation promise significantly better results.

The aspects of safe functioning and minimized energy consumption are crucial for this project. Therefore it is important to select the best concept of control and to take the possibility of recuperation from braking into account.

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FOOTNOTES

1. Considering the frequency of different distances (not the altogether driven distances) seems to be adequate because of environmental and psychological reasons. The emissions after each starting from cold are distinctly high; the decision between different means of transportation is made at the beginning of each trip.
2. That applies especially because of the high emission load for the inner city area by the typical alternative for the electric drive: a two-stroke combustion engine.
3. Fail-safe-properties: appearing defects do not lead to a dangerous situation but are intercepted safely.

AUTHOR

Andreas Könekamp has studied electrical engineering at the University of Technology Darmstadt (TUD) specialising in electro-mechanical engineering design. Since 1993 he has been working at the Center for Interdisciplinary Studies in Technology at the TUD as a consultant supporting engineers in developing environmentally-friendly products. He is also conducting several projects on bicycle trailers.

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CALL FOR PAPERS

FOURTH EUROPEAN VELOMOBILE SEMINAR

The fourth European velomobile seminar will be held in conjunction with the world championship for human power, 18–22 August 1999. The seminar is on the first day, Wednesday August 18, and will be in Interlaken, Switzerland. (This will be the first time the HPV championship is to be held in Switzerland, and it is certain to be a very special occasion.) The host organization is Future Bike, and the symposium co-chairs are Andreas Fuchs <fuchs@isbe.ch> and IHPVA chairman Theo Schmidt <tschmidt@mus.ch>. Prospective authors should write to either Andreas or Theo. The emphasis is on power-assisted HPVs.

Measuring drive-train efficiency

by Angus Cameron

ABSTRACT

A simple but effective procedure for measuring the static mechanical efficiency of bicycles is presented which would be suitable for backyard builders, high-school science labs, science-fair contestants, and even small manufacturers. Test results from a 21-speed bike showed efficiencies ranging from 92.4 to 98.0% depending on chain tension and size and condition of cog, with uncertainties between 0.15 and 0.35%.

INTRODUCTION

The measurement of chain efficiency may have started with Prof. R. C. Carpenter at Cornell University in 1887, as reported in the "100 Years Ago" section of the journal *Nature*, 2 October 1997. He is quoted that "frictional loss has been found to be between 1/2 and 3/4 per cent of the total power transmitted."

Although some values of chain efficiency have been published for normal bicycles (see table 1, for example), little information exists about the effects of novel designs incorporating extra idlers, chain tubes, intermediate drives, toothed-belt drives and so on. The techniques described here will allow any curious person who has access to a spring balance and a set of calibrated weights to make his/her own measurements.

The term mechanical efficiency is commonly defined as the ratio of the power out to the power in to some mechanism. Measuring power requires knowledge of speed, which makes it a dynamic measurement. Power, however, is difficult to measure without extensive lab resources. Therefore I chose to measure the static efficiency; the ratio of the work out to the work in; where work is defined as the product of a force and the displacement it causes. Static in this sense doesn't mean that there is no motion involved, just that the speed is small and needn't be known.

Table 1. Comparisons of single-speed, multi-speed hub and derailleur gearing

	1-spd	3-spd hub gear			6-spd derailleur		
		Low	1:1	High	24T	19T	13T
50W	96.0	90.6	93.4	87.3	94.2	94.1	92.1
100W	97.3	92.8	95.7	90.9	96.2	96.4	94.9
200W	98.1	94.0	96.9	92.9	97.4	97.6	96.9
400W	99.0	95.0	97.9	93.9	98.1	98.4	97.8

My test vehicle was a 9-year-old mountain bike equipped with Deore XT components and a lubricated chain that had lengthened 1/8 inch per 12 inches.

My first experiment was to find the friction loss in a simple loop of chain running over the large 42-tooth sprocket. I supported the bike with a sturdy stand with the bottom bracket at table height. After removing the drive chain and setting it aside, I laid a second shorter loop of chain over the chain wheel. Using strong steel hooks (coat-hanger wire) I hung a 6-kg weight on each side of the chain so that they balanced (see fig. 1).

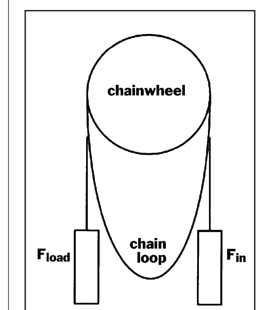

Figure 1

Originally I intended to add small brass weights to one side until the friction force was just overcome. At this point the chain wheel should have continued to rotate at a constant speed once given a small push.

However, the friction force was far from constant and also a little "lumpy" due to the engagement of the chain with the teeth, making it impossible to determine the value of the friction force with any consistency. So I substituted a sensitive spring balance for the brass weights. Using a steady hand to pull each of the 6-kg weights in turn towards the floor at a steady slow pace, it was easy to watch the pointer and "eyeball" the average friction force.

When the tension in the chain was 6000 grams the friction force (f) was found to average about 55 grams. Dividing the friction by the tension and multiplying by 100 gave a relative loss (or inefficiency) of 0.9%. Subtracting this from 100 resulted in an efficiency value of 99.1%.

The reader must excuse my use of mass units (g and kg) rather than the correct force unit, the Newton. There are two pragmatic reasons: first, the weights and measuring instruments were calibrated in grams, and second, since efficiency is defined as a ratio of two quantities, the units will always cancel regardless whether in grams or newtons.

Encouraged with this result I replaced the spring balance with a Pasco electronic force balance and a data logger and made additional readings with weights of 6, 11 and 16 kg. As can be seen in figure 2, the data points fell close to a straight line, showing that friction increases in proportion to the chain tension. More importantly it told me that the technique was producing consistent and reproducible data.

When the efficiencies were calculated they averaged 99.1% and showed a slight increase with an increase in tension (see fig. 3).

The next question was whether this same technique could be used to find the overall efficiency of the complete drive train. Somewhat to my surprise, the algebra said yes. Starting with the definition of "work", it can be shown (see appendix 2) that efficiency = $1 - f/F_{100}$ where f is the friction measured at the rear wheel and F_{100} would be the equilibrium force required if the system was free of friction. The friction force f can be found by averaging the two measured forces, f_{up} and f_{dn} , using equation 7, while F_{100} can be found using equation 6. Note that the magnitude of only the smaller

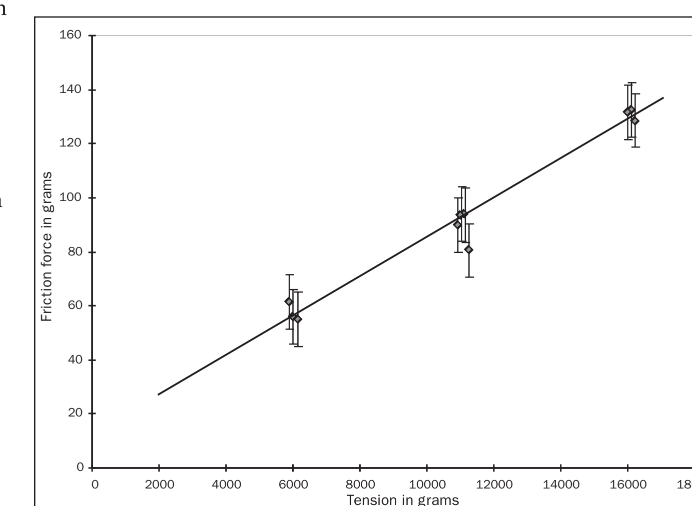

Figure 2. Friction on the 42-tooth sprocket

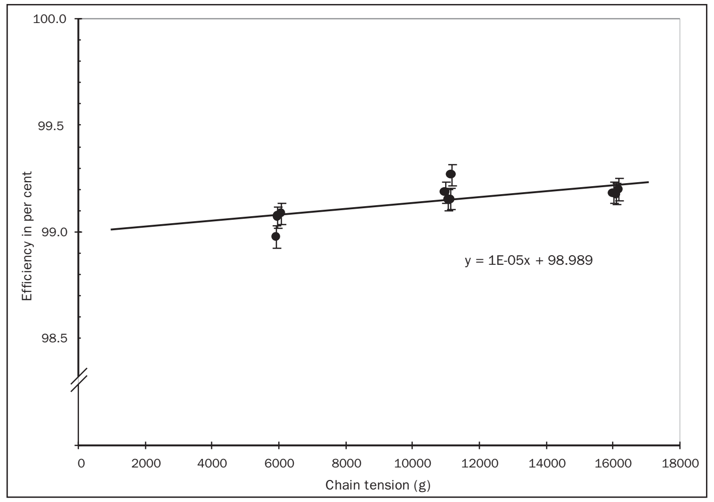

Figure 3. Efficiency values for a single 42-tooth sprocket

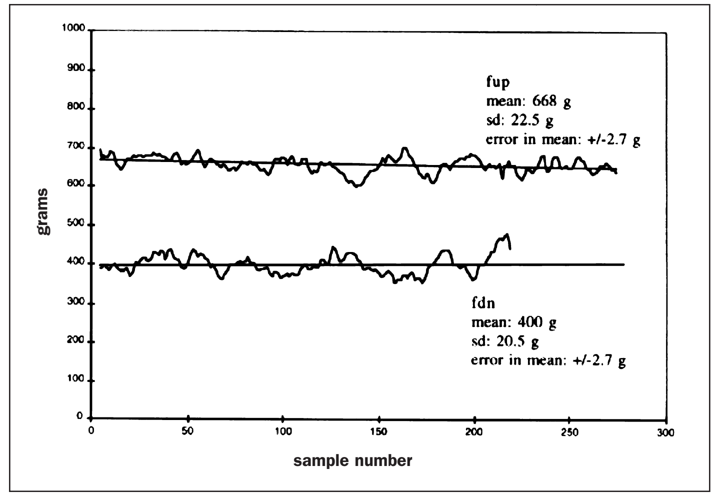

Figure 4. An example of the data stream from the force balance

weight F_{load} needs to be known with precision, permitting one to use scrap iron or sand bags for the larger weight F_{in}.

The chain was then reinstalled on the middle 34-tooth ring. A turn and a half of heavy fish line wrapped around the bare rim was used to support the output load (F_{load}). The large input weight was suspended from the 42-tooth ring using a separate loop of new chain (fig. 6). I used 6.5-, 13- and 26-kg weights because the resulting chain tensions corresponded approximately to that generated by a typical rider spinning at 75 rpm (see table 2).

Table 2. Power transmitted through a 42-tooth sprocket assuming it is rotating at 75 rpm

mass on 42-tooth ring	chain tension 34-tooth ring	equivalent power at 75 rpm
6.5 kg	79 N	43 W
13 kg	158 N	85 W
26 kg	315 N	170 W

My test procedure was straightforward. For each series of measurements I checked the zero and span of the force balance and selected an appropriate value for F_{load}. Then, with the force balance in my hand, I lifted F_{load} steadily upwards to measure fup. I always turned the data logger on after the initial jerk, and turned it off before reaching the top limit. This procedure was then repeated in the opposite direction to measure fdn . The mean, standard deviation and error in the mean were then determined. A typical result is shown in figure 4.

I used a value of 99.6% for the efficiency of the 42-tooth drive sprocket when correcting for its contribution to the overall efficiency. This value came from the assumption that the efficiency of 99.1% found previously was caused by the product of 99.6% efficiency as the chain engaged the sprocket on one side and another 99.6% as the chain disengaged from the opposite side.

Plotting the corrected efficiency against the number of teeth in the cogs, a set of curves showed that the highest efficiency values occurred at highest chain tensions (see fig. 5). The higher efficiencies tended to occur in the middle cogs, presumably where the chain line was straightest. The efficiencies of the smaller cogs were consistently lower than any of the others, presumably because their small radius causes each chain link to rotate through a larger angle, requiring commensurably more work to be done in overcoming friction even if the actual friction force didn't change. Two of the curves, labeled 26 kg (170 W) and 13 kg (85 W) also overlapped two broken curves representing the 200- and 100-W data from table 1, suggesting a heartening degree of agreement between the two completely different measurements.

The error bars on the graphs were estimated using a standard method (using derivatives) of error analysis with the following assumptions: the uncertainty in the weights used for F_{load} was 1%, the uncertainties in f_{up} and f_{dn} could be found using their standard deviations and number of samples that the movement of the system could be kept perfectly steady.

There are at least two ways to minimise the uncertainties in f_{up} and f_{dn}. One is to increase the number of samples. The second would be to devise a mechanism, perhaps using a motor, that would eliminate the human factor in keeping the weights moving at constant speed. The deviations from the means can never be totally eliminated because the engagement of the chain with the teeth will always produce a varying effective sprocket radius.

There are two bicycle components that can't be measured. The first is the oval or BioPace chainring, because the mechanical advantage doesn't remain constant. The second is the internal gear hub, which is unfortunate, because my original objective was to find the efficiency of a Nexus 7-speed hub. The problem occurs when the hub is rotated backwards and a set of pawls engage, causing the hub to shift into a lower gear. I have taken measurements using a severely restricted range of motion, but the number of usable readings is too small to obtain any reliable values.

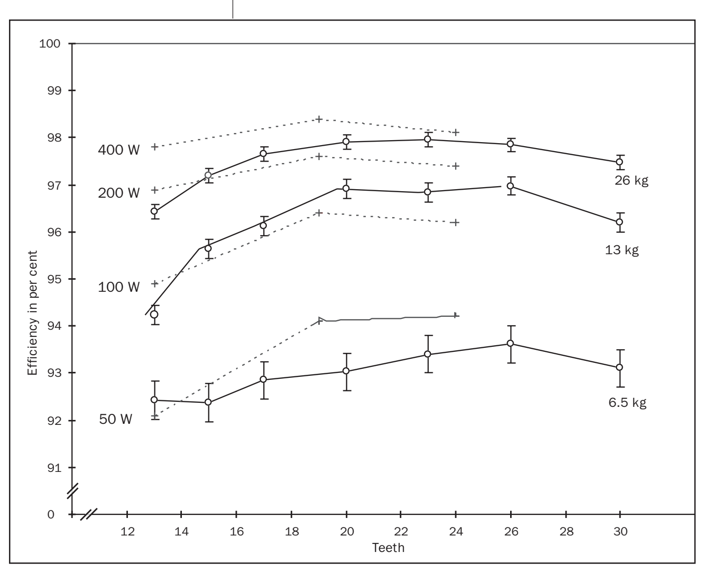

Figure 5. Solid lines represent the measured drive-train efficiency. The dashed lines represent data from table 1.

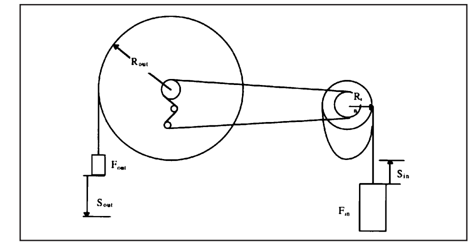

Figure 6. Arrangement of the weights

SOME PROJECTS THAT MIGHT BE OF INTEREST

1. How much does the efficiency depend on the diameter of the sprocket?
2. Does the efficiency depend on tension on the slack side of the chain?
3. Does a chain tube affect efficiency?
4. What is the effect of over-tightened bearings in the bottom bracket and the rear wheel?
5. How much does the efficiency depend on misalignment between two sprockets?
6. How much does the efficiency depend on the lubrication and wear of the chain?
7. Find the efficiency of a bike using intermediate gears.
8. Find the efficiency of a toothed-belt drive.
9. For bikes that use a drive-side idler to divert the chain under the seat, how is efficiency affected? How does it vary with the angle of deviation from a direct pull?
10. For someone with sophisticated lab facilities, do the results of this simple technique agree with traditional methods?

CONCLUSIONS

In conclusion, although this report hasn't produced any surprises about chain efficiency, it has proved that the concept works and that accurate static measurements are easy to make. Even if it is ever shown that results using the static method don't agree absolutely with dynamic measurements, the relative values still provide valuable information.

REFERENCES

The values in table 1 are courtesy of the archives of the Hardcore Bicycle mailing list. The study was commissioned by Fichtel & Sachs AG and was published in Radmarkt Nr.12/1983.

Pasco Scientific (http://www.pasco.com) sells electronic force balances and brass weight sets.

APPENDIX 1. DEFINITION OF TERMS

IMA: ideal mechanical advantage = Sin/Sout = (Rin/Rout)*(Tout/Tin)*(internal hub ratio)

R_{in}/R_{out}: ratio of crank radius to wheel radius

T_{out}/T_{in}: ratio of number of cog teeth to chainwheel teeth

TMA: true mechanical advantage = F_{out}/F_{in}

F_{in}: input force

F_{out}: output force

F_{100}: ideal output force assuming 100% efficiency = F_{in}*IMA *Eqn 1*

f: true friction force measured at the output side

f_{up} , f_{dn}: measured friction forces

F_{load}: a load which roughly balances the input force, ~F_{out}

S_{in}: S_{out}: linear distances

W_{out}; $Work_{out} = F_{out}*S_{out}$

W_{in}; $Work_{in} = F_{in}*S_{in}$

E; efficiency = W_{out}/W_{in}

APPENDIX 2. DERIVATION OF THE FORMULAS

E □ $W_{out}/W_{in} = F_{out}*S_{out}/(F_{in}*S_{in})$

$= F_{out}/(F_{in}*IMA)$

Let F_{out} □ $F_{100} - f$

Then $E = (F_{100}-f)/(F_{in}*IMA)$

$= (F_{in}*IMA-f)/(F_{in}*IMA)$

$= 1- f/(F_{in}*IMA)$ *Eqn 2*

which reduces to $E = 1-f/F_{100}$ *Eqn 3*

f and F_{100} can be found experimentally as follows:

f_{up} and f_{dn} are the friction forces measured with a spring balance. f_{up} is positive when the spring balance is pulling upwards; f_{dn} is positive when it is pulling downwards.

F_{load} is the magnitude of the weight suspended from the rim

$F_{load} + f_{dn} = F_{100} + f$ *Eqn 4*

$F_{load} - f_{up} = F_{100} - f$ *Eqn 5*

adding 4 and 5:

$F_{100} = Fload + (f_{dn} - f_{up})/2$ *Eqn 6*

subtracting 5 from 4:

$f = (f_{dn} + f_{up})/2$ *Eqn 7*

THE AUTHOR

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COMMENT ON ANGUS CAMERON'S ARTICLE

Angus has done a nice job, ingeniously using available equipment, to make sensitive measurements of an elusive quantity (chain-drive losses). His writeup raises some questions, but I am not certain how significant they are.

Angus doesn't explain how he corrects for the losses where the input weight hangs from the 42-T sprocket. Furthermore his 'calibration' of this loss involves an unusual chain loading—both ends taut—which differs somewhat from the test setup. This difference, plus the low precision of the 0.4% correction, suggest that some of his error bars should be greater. My recommendation would be to apply the input weight to a drum attached to the crank, using fishing line.

Chain-drive mechanical advantage varies several percent as a small-sprocket tooth passes. If a slow motor were used to pull the weight, the measured tension would vary a great deal (twice its nominal value). So Angus' hand is providing some valuable filtering!

I am a little concerned by the up/down technique. On the one hand it obviates the need to measure the output weight. On the other hand, chains do not work exactly the same in forward and reverse. It would be worth at least checking whether this one does, before relying on it.

Angus mentioned the impossibility of measuring elliptical chainrings. But chain wear (which is typically uneven) will produce a similar problem: where chain pitch is greater, the chain will sit out at a larger radius on the sprocket. So, as he suggests, longer test runs could be useful.

Congratulations on a useful article!

—Jim Papadopoulos

Predicting wheel dish from hubs

by Vernon Forbes

ABSTRACT

A number of freewheel widths, hub widths and axle lengths exist. Both the number of speeds and the center-to-flange measurements are shown to predict poorly the resulting rear-wheel dish. It is suggested that dish is best measured as a ratio.

INTRODUCTION

What factors influence rear-wheel strength? Certainly the amount of dish does. With increasing amounts of dish the tension of the spokes on the freewheel side of the hub, or drive side, increases and they approach their elastic limit.

Dish is the result of several variables: freewheel width, axle length and hub width. People seeking a stronger rear wheel will often choose a 7-speed because it is thought to be less severely dished. Axle length is another variable. Manufacturers have been increasing axle length to make room for wider freewheels. Putting the freewheel further outboard requires an increase in the bottom-bracket spindle length in order to keep the chainline correct. It is well known that wider bottom brackets are harder to pedal.

Ever since the introduction of multiple-cog freewheels, wheels have had to be dished. A freewheel moves the hub over to make room for it. Cyclists are long familiar with spokes on the drive side being tighter and more likely to break. What is needed is a way to predict how much a wheel will have to be dished from any hub to be used as a guide in hub selection.

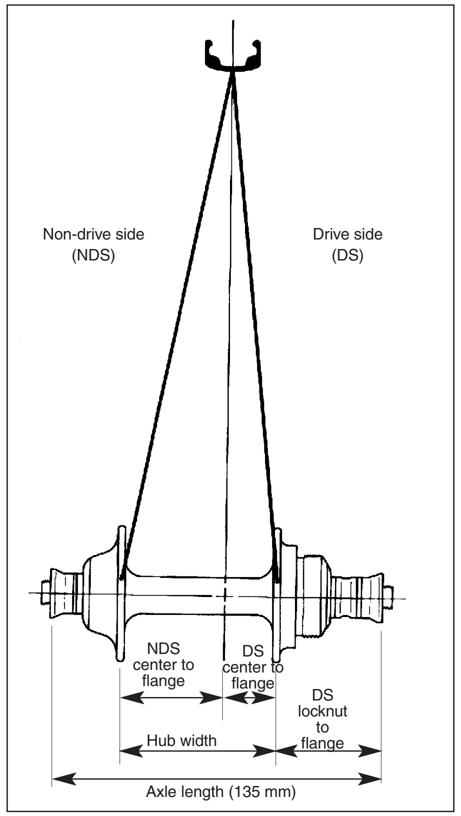

Figure 1. Wheel dish. This diagram appears in *The Bicycle Wheel* and is used with the author's permission.

FREEWHEEL WIDTH

We initially set out to establish how much narrower the freewheel spacing on 7-speed hubs is compared to 8-speed hubs of 135 mm spacing.

The hubs listed in Sutherland's[1] were used in this and all subsequent analyses. Only hubs having a 135 mm axle length were used. Although 130 hubs are listed as having a 135 mm axle length only 74 are labeled. Sutherland's lists 31 hubs as 8-speeds, 43 as 7-speeds, 9 as 7/8-speed and 1 as 6/7-speed; 45 are not categorized. Sutherland's gives the center-to-flange measurements for both the drive side and the non-drive side. Freewheel width was taken as the distance from the drive-side locknut to the flange center (see fig. 1). An examination of 74 labeled hubs in Sutherland's was conducted to find out if there were any consistent patterns that constituted 7- or 8-speed spacing. Considerable variability exists as to the width of both 7- and 8-speed freewheels. The following values (mm) were obtained from labeled hubs only. The table below shows the number of speeds (#Speeds), the number of hubs analyzed (N), the average (AVG), the lowest and highest values (Range) and the standard deviation (SD). The 68% confidence intervals (CI68) are discussed later.

Freewheel widths: labeled hubs

# speeds	N	AVG (mm)	Range (mm)	SD (mm)	CI68 (mm)	Effective CI68 (mm)
8	31	48.0	45–51	1.5	46.5–49.5	>47
7	43	44.4	39–49	2.1	42.3–46.5	<46

What the ranges clearly show is that, contrary to popular wisdom, in many cases a 7-speed hub is actually wider than an 8-speed hub. While there is an average difference between 7- and 8-speed hubs the ranges actually overlap, so that the narrowest 8-speed freewheel is actually 4 mm narrower than the widest 7-speed freewheel. It is easy to choose a 7-speed hub (out of the belief that it will build a stronger wheel) that actually requires more dish than a wheel built with an 8-speed hub.

Since the ranges overlapped we sought to determine what standards might exist by which to judge whether a hub was a 7- or 8-speed. To sort the 45 unlisted hubs as either 7- or 8-speeds a 68% confidence interval (CI68) was used where one standard deviation (SD) is subtracted from the mean (AVG) to get the lower limit and one standard deviation is added to the mean to get the upper limit. Hubs less than 46 mm were classified as 7-speed. Those greater than 47 mm were classified as 8-speed. There were no hubs with a freewheel spacing of 46.5 mm. Because the Effective CI68 for 7 and 8 speed hubs did not overlap, all 45 were sorted as either 7- or 8-speed, as illustrated by the table below. We offer the above confidence intervals and show (in the table below) how it can be used to determine whether a hub with a 135 mm axle length of uncertain spacing is either 7- or 8-speed. Since 7/8-speed hubs did not form a separate category but overlapped both 7- and 8-speeds they were sorted into either 7- or 8-speed hubs.

Freewheel widths:
labeled and sorted hubs together

No. of Speeds	N	AVG (mm)	Range (mm)	SD (mm)
8	56	47.8	45–51	1.3
7	73	44.6	39–49	1.7

HUB WIDTH

Hub width is the distance between the flanges, center to center (see fig. 1). In an attempt to reduce the effects of increased dish created by wider freewheels some hub manufacturers are using "dishless" hubs. These are nothing more than narrower hubs. The tendency to use narrower hubs as the freewheel gets wider is reflected in labeled 8-speed hub widths being narrower than 7-speed hubs (X8-speed = 54.8, X7-speed = 55.3). These hub widths have, however, a very large standard deviation (SD7-speed = 5.12, SD8-speed = 3.46) and overlapping ranges (7-speed = 46-65, 8-speed = 48-61). If there is variability among freewheel widths there is even greater variability among hub widths.

Clearly hub selection determines the dish of the resulting wheel into which it is built. What is needed is a way to compare hubs along both dimensions simultaneously that predicts the strength of the resulting wheel. Center-to-flange numbers are likely candidates for this measure because they account for freewheel width, axle length, and hub width.

THE DISH RATIO

Considering the importance of dish in determining rear-wheel strength how do we measure it? Center-to-flange measurements allow quick comparisons between hubs. The longer the drive-side center-to-flange distance, the stronger the resulting wheel is. Dish, however, is the difference in tension between both sides of the wheel and can be expressed as the difference between the two center-to-flange measurements for each side of the wheel. To measure dish truly we must somehow include the center-to-nondrive side. Using the labeled hubs only, figure 2 plots the length of the drive side (center-to-flange) against the non-drive side, illustrating the variability of these two measures. The straight lines that appear are not regression lines but are lines of constant dish explained below.

The amount of dish a hub puts on a wheel can be measured by comparing the center-to-flange measurements of each flange to each other. One such comparison is the dish ratio, achieved by dividing the smaller center-to-flange measurement on the drive side by the center-to-flange measurement on the longer non-drive side. The dish ratio compares the center-to-flange measurements on each side of the hub. For example, if a hub has a 20 mm center-to-flange measurement for the drive side and a 40 mm center-to-flange measurement for the non-drive side, dividing the drive side by the non-drive side yields 0.5, indicating that the center-to-drive-side length is 50% of the non-drive side. A front wheel with no dish yields a value of 1.0, indicating that one side is the same length as the other. The following values were obtained for labeled hubs only.

The lines of constant dish were added because for any given dish ratio, e.g. 0.5, there are a number of combinations that achieve the same dish ratio of 0.5. A hub with a 20 mm center-to-flange measurement for the drive side and a 40 mm center-to-flange measurement for the non-drive side yields a dish ratio of 0.5. A hub with a 25 mm center-to-flange measurement for the drive side and a 50 mm center-to-flange measurement for the non-drive side also yields the same dish ratio of 0.5. The lines of constant dish were added as guide to the dish ratio of each hub plotted.

Dish ratio = DS/NDS (1.0 = no dish)

# speeds	N	AVG	Range	SD
8	31	.56	.72–.46	.06
7	43	.75	1.0–.53	.14

What should be apparent from the above table is that 8-speed hubs reflect increased dish compared to 7-speed hubs. Several of the 7-speed hubs had no dish (dish ratio = 1.0) and were threaded to accept a disc brake.

Although the center-to-flange ratio measures the amount of dish, it does not discriminate between different widths of hubs. We conceptualized the hub's influence on the resulting wheel strength in the following way. We saw dish as a function of freewheel width impinging on the amount of strength available to divide between different sides of the wheel provided by hub width.

While some manufacturers are using narrow "dishless" hubs these hubs come in different widths. Figure 3 plots the amount of dish (as measured by the dish ratio) against the hub's width for both seven-speed and eight-speed hubs. The regression lines plot both seven-speed and eight-speed hubs, illustrating how increased dish is often coupled with narrower hubs. This is especially so for eight-speed hubs.

Examination of figure 3 also reveals that the amount of dish is independent of hub width. For example, two hubs have a dish ratio of approximately 0.69, yet one of them is 49 mm wide while the other one is 59 mm wide. A solution measuring dish that accounts for hub width is problematic.

The addition of hub width can enhance the discrimination of the dish ratio. The most straightforward way would be simply to specify hub width along with the dish ratio. For example, 0.69–49 mm or 0.69–59 mm allows us to discriminate between two different-width hubs having the same dish ratio.

DISCUSSION

What we are seeking is a way to predict wheel strength from a hub: a single number

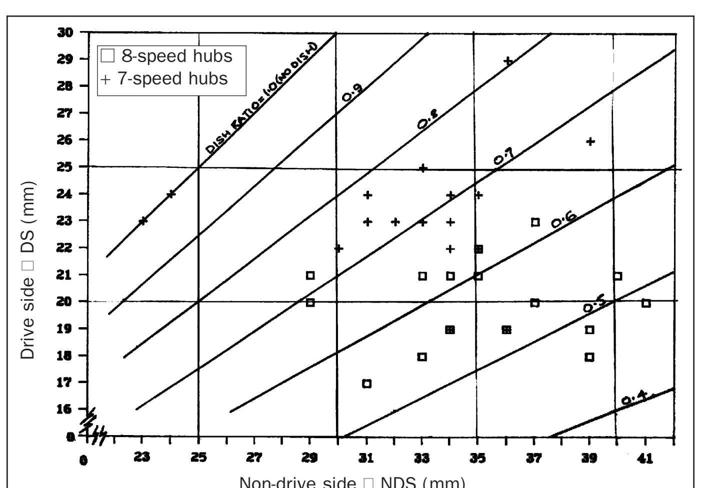

Figure 2. Dish ratio □ drive side/non-drive side

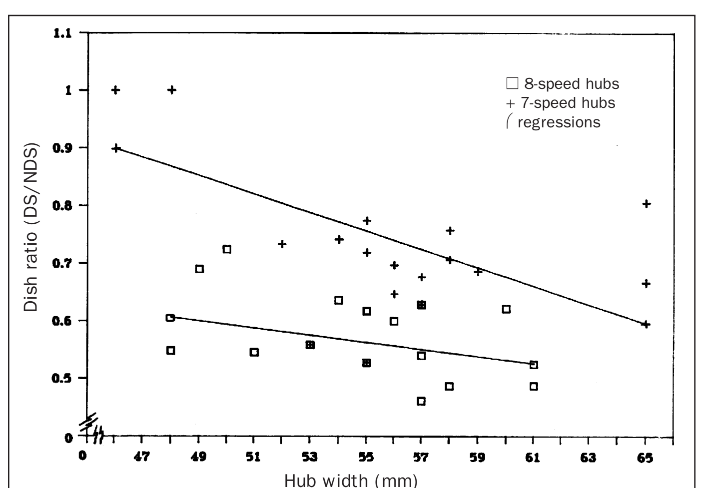

Figure 3. Hub width by dish ratio

that allows a quick comparison among hubs that is easy to derive from known measurements.

Both the number of speeds and drive-side center-to-flange width are too inconsistent to be a source of meaningful comparison. The dish ratio alone may be the truest measure of dish but does not account for hub width. We feel the additional specification of hub width allows meaningful comparison among different hubs.

Suppose we have two hubs with the same dish ratio but different widths, like the 0.69–49 mm and the 0.69–59 mm above. Which builds a stronger wheel? Consider different kinds of strengths: lateral strength and durability. A wider front hub is laterally stronger. This is also true for the rear wheel, but only up to a point. As this measurement increases, the spokes on the non-drive side lie at a slacker angle and are increasingly loose. As they fall further below their elastic limit they would have further to stretch in response to a non-drive-side load. A loosely spoked wheel is more likely to collapse, or permanently assume a potato-chip shape in response to a lateral load.[2]

Mountain bikes are frequently subjected to lateral loads. While hub width is important, increasing the hub's width beyond a certain point would, of necessity, decrease the dish ratio and increase the dish. So for each freewheel width there must be an optimum hub width. While it would be a simple matter to make all hubs "dishless", we are reminded of figure 2, which illustrates no such systematic relationship evident from any current manufacturing trends. The current trend towards wider freewheels with narrower hubs seems unlikely to result in wheels less likely to collapse. The ideal is a wider hub with less dish.

Increased tensions resulting from dish also affect spoke fatigue. Fatigue results from a spoke's movement, or stretching in response to stress. After many thousands of stress cycles the spoke begins to develop cracks from what is originally a small surface irregularity. Fatigue results from repeated movement. A tighter spoke has all the "stretch" taken out of it and is going to move less. It is for this reason that Rafael Raban[3] claims that spoke failures occur more frequently on the non-drive side. Brandt[4] also claims that spokes fail from fatigue but observes that fatigue is the result of an interaction between stress cycles and the baseline level of stress the spoke returns to. At higher baseline stress levels the spoke is more likely to fail from fatigue and it is for this reason that he claims spokes are more likely to fail on the drive side. It is possible that some dish ratios and widths are more likely to result in wheels that resist collapse at the expense of fatigue while other hubs give wheels more likely to collapse than to resist spoke fatigue.

While the author has observed the increased frequency of spokes to break on the drive side, he recently heard of a wheel built with Union titanium spokes that had repeated spoke failures on the non-drive side. This is especially curious given titanium's superior fatigue resistance. The reported hub was a Hugi 7-speed (cassette). If this is so it had a 22 mm DS length with a 35 mm NDS length for a dish ratio of 0.63, which is mid-way between a 7-speed and 8-speed dish ratios (see above). The hub width was 57 mm which is 1.7 mm wider than the average 7-speed hub (see above). While wheel collapse and spoke fatigue are beyond the scope of this article, we are attempting to offer only a basis for the measurement of dish and to suggest its utility.

It should be mentioned that the axle spacing is frequently changed by mechanics to keep the chainline correct. This changes the amount of resulting dish.

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Figure 1 is after *The Bicycle Wheel* by Jobst Brandt and appears with his permission.

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LETTER

WIND RESISTANCE PEDALING VS. COASTING

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

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

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A bicycle with auxiliary hand power

Duhane Lam, John Jones, John Cavacuiti and Andrea Varju

ABSTRACT

Modifications have been made to a conventional bicycle, allowing the rider to supply auxiliary power to the front wheel by pumping with the arms. The gear ratio of the auxiliary power train can be continuously varied, to match the arm power available. The design and construction of the auxiliary drive, and the experience of riding the modified bicycle, are described.

Keywords: Bicycle, front-wheel drive, auxiliary hand power, Handle-Drive system

INTRODUCTION

There are several reasons that a rider might want to use his or her arms to supply auxiliary power. Firstly, there is the goal of maximizing power output:

> "Pedaling as on conventional bicycles enables riders to approach their maximum power output. However, mechanisms that give noncircular foot motions or nonconstant velocities, or both, and mechanisms that allow hands and feet to be used together, seem to be required if the absolute maximum power output is to be obtained".[3]

Secondly, having two powered wheels can improve traction on steep climbs, or on loose or slippery surfaces.

Whether adding hand-power is the best way to improve total power output remains controversial. In *Bicycles and Tricycles* (first published in 1896), Archibald Sharp states, regarding the possibility of obtaining greater speed or maintaining the same speed with less effort by incorporating different muscle groups:

> "A number of cycles have been made from time to time with gearing operated by the hand, having the intention of supplementing the effort communicated by the pedals. The idea of the inventors is that the greater the number of muscles concerned in the propulsion, the greater will be the speed, or a given speed will be obtained with less fatigue; but though this (using different muscles) may be true for extraordinary efforts of short duration, it is probably quite erroneous for long-continued efforts."[2]

The argument is that the limiting factor on a bicycle is cardiovascular, not muscular. Sharp, Whitt, and Wilson all agree that for short, high-power efforts, hand power would be an advantage. For these types of exercise, the limiting factor is the anaerobic threshold of the muscles. It is not as clear what the limiting factor for long-duration activities is; the question frequently arises whether one can add hand cranking power to pedaling and obtain a total power output that is the sum of what one would produce using each mode independently.

Kyle and co-workers found that, for periods of up to a minute, 11–18 percent more power could be obtained with hand and foot cranking than with legs alone. Whether or not this gain can be projected beyond the period of anaerobic work is not known.[3] Conversations with Dr. Tom Richardson in the kinesiology department of Simon Fraser University and Steven Cheung (a graduate student in exercise physiology), in addition to personal experience riding marathons, raise several relevant points: cross-country skiers have some of the highest VO_2 maximums (the ability to utilize oxygen) of all athletes. This is attributed to their use of almost the entire body (many muscle groups) in propelling themselves. Therefore, there is an increased efficiency of the cardiovascular system to meet the demands of the muscles. Another interesting lesson to be learned from cross-country skiers is the co-ordination of the legs and the arms. We shall see the relevance of this shortly. Muscle fatigue over long distances does occur, despite the fact that the muscles are operating aerobically. Less fatigue would occur over a given time if more muscles shared the load.

Riding a bicycle, especially a mountain bike, may involve hills and other terrain variations, requiring a series of rests and short bursts of power rather than one long, continuous effort. So it seems possible that auxiliary hand power would be an advantage.

Despite this, there are no hand-and-foot-powered bicycles on the mass market. One reason may be the difficulty of designing an elegant mechanism to harness power from the legs and the arms while still allowing the rider to balance and control the bicycle. The hands are traditionally used for steering the bicycle, shifting gears, supporting a portion of the rider's weight and, last but not least, braking. Although mechanisms harnessing hand power have been used on recumbent bicycles which involved radical drive-trains and rider positions, these machines were such a departure from the normal standard bicycle that they never gained, or were intended to gain, widespread use.

Previous hand-powered designs have included Francis Green's "Rockabike",[1] on which the rider rocked back and forth with a rowing motion, power being transferred from the pivoted handlebars to the pedals by a two-way chain and ratchet system. One difficulty encountered with this design was steering while moving the handlebars. Another mechanism using hand power to drive the front wheel was patented by J. P. Souhart in 1940.[4] Souhart's device used a swiveling handlebar to power the front wheel with a fixed gear ratio.

PROBLEM DEFINITION

The problem we set out to solve was to develop a safe and efficient bicycle that harnesses power from both the hands and the feet with potential mass-market appeal. We set ourselves a number of criteria to meet guidelines of efficiency and safety in addition to commercial attractiveness.

- Circular motion of the legs is retained.
- Feet and hands are used to drive the bicycle. Hands and feet can work together at the same frequency and 180° out of phase, while torque can be transmitted through both hands and feet.
- Range of speeds and gearing is similar to that found on modern bicycles.
- Addition of weight to the bicycle should be minimized.
- The rider is placed in a standard, upright, and visible (to motorists) riding position (non-recumbent).
- Steering is uncompromised and intuitive (e.g., a wheel should not be turned clockwise to get a vehicle to turn left).
- Option exists of stopping hand motion while still pedaling with the feet. This facilitates steering.
- Rider has full control of braking and gear-control functions without moving hands from driving position.
- System can be mounted onto a standard frameset.

- Mechanism requires no structural modification of the frameset. Therefore, users are not required to purchase a whole new bike. Riders should be able to learn to ride the new system in a reasonable amount of time (e.g., one day's worth of riding).
- The mechanism is robust enough to handle normal cycling conditions.

The requirement of hands and feet moving 180° out of phase requires some further explanation. Hand-and-foot drives have been implemented with hand motions similar to the foot motion (cranking), rowing motions with the hand while the feet cranked, or up-and-down motions simultaneously with both hands out-of-phase with the legs. None of these drives is entirely satisfactory. Often the hand motions feel unnatural and the other functions required while riding a bicycle are difficult to perform with the hands. We approached the problem from a new direction and came up with a different method of harnessing hand power on a bicycle that works in conjunction with the legs. In fact, although we might not realize it, we already utilize our upper-body muscles when riding a bike. Arm muscles are often used to provide a reaction force against the force of the legs pushing on the pedals. When climbing or sprinting out of the saddle, cyclists often sway the bicycle back and forth. In effect, this is using both leg and arm muscles to exert a torque on the cranks by pushing down with the legs on the pedals and pulling up on the frame with the hands. The action described above is a natural motion for experienced cyclists and is potentially the most efficient way for the body to work. Whitt and Wilson state, in reference to the work by Kyle and co-workers:

> "The power was greater when the arms and legs were cranking out of phase than when each arm moved together with the leg on that side."[3]

The goal is to mimic the natural action of a cyclist climbing as closely as possible but providing for even greater use of the arms and upper body. The desired action is such that when the right foot pushes down on the pedal, the right hand pulls up and the left hand pushes down. Similarly, when the left foot pushes down on the pedal, the left hand pulls up and the right hand pushes down. The timing of the hands and legs is very similar to that of a cross-country skier doing a diagonal stride.

THE HANDLE-DRIVE BIKE

After much thought, we arrived at a general solution that meets most of the conditions stipulated. We built a prototype, the Handle-Drive bike. Two levers mounted on the stem are moved up and down by the hands as shown in figure 1.

The hands grip a handlebar (see fig. 2) which protrudes out from the side of each lever. These handlebars are mounted to give the approximate reach, width, and gripping angle of standard handlebars on existing mountain bikes. The cyclist also has the option of gripping along the main levers.

Brakes, rear and front derailleur controls are mounted as shown in figure 2 to allow access even while using the arms to power the front wheel. A chain is mounted on an assembly that slides up and down the main lever on each side. This chain powers freewheel sprockets on either side of the front hub (see fig. 3). The levers are connected together by a gear such that when one lever moves down, the other lever is forced upwards and vice-versa. Note that this system is inherently a two-wheel-drive system. Various attempts have been made in the recent past to invent a two-wheel-drive bicycle that provides for greater traction. These implementations usually involve a flexible cable drive connecting the rear wheel to the front wheel. Because the hands power the front wheel and the legs power the rear wheel, the Handle-Drive system is automatically two-wheel drive. In addition, it has a very sophisticated traction controller (the rider) that can vary the amount of power applied to each wheel as circumstances and terrain dictate.

The rider drives the front wheel by moving his or her hands up and down. There is no mechanical connection between the rear-wheel drivetrain and the front-wheel hand-drive drivetrain. Thus the rider can synchronize the two drivetrains in whatever way feels the most natural. Generally, this will be the action described above whereby the hand pulls up while the leg on the same side pushes down. However, the rider is free to stop powering with his or her hands at any given time (e.g., to maneuver around obstacles or to apply the brakes). In fact, the rider can power with only hands, only legs, both together, or none at all. The levers can still support the cyclist's weight if the cyclist pushes down evenly on both levers at once. A good feature would be to allow a lockout whereby both levers can be locked together in one position. In this case, the bike would essentially ride like a normal bike. The point of connection of the chain to the lever is varied along the length of the lever to give a range of speeds varying from low (close to the bicycle end of the lever) to high (at the far end of the lever).

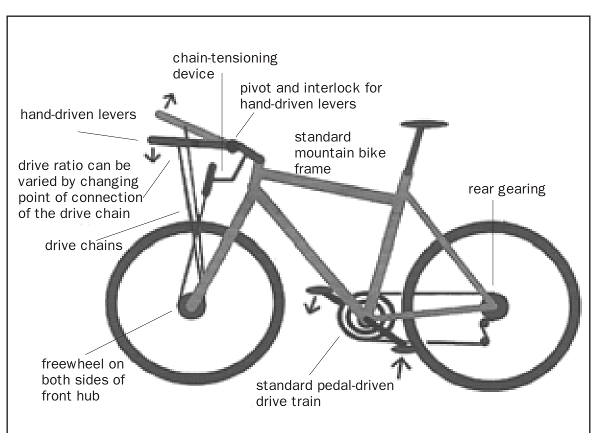

Figure 1. Side view schematic of the Handle-Drive system.

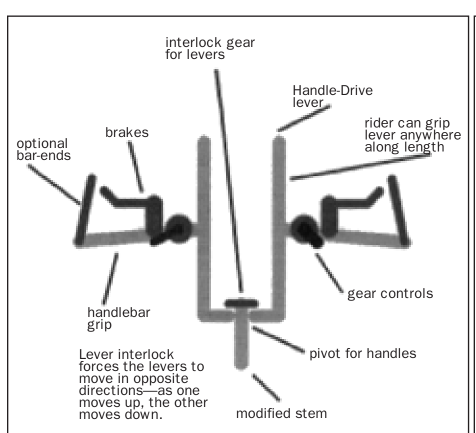

Figure 2. Top view schematic

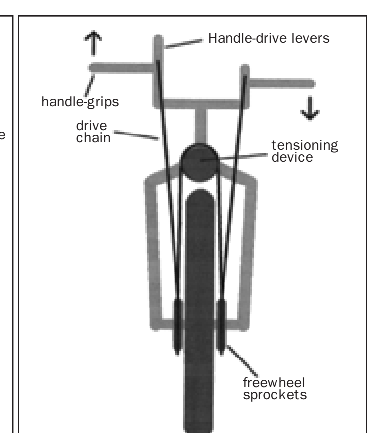

Figure 3. Front view

RIDING THE HANDLE-DRIVE BIKE

The prototype was built by replacing the stem, handlebars, and front wheel of a standard mountain bike. The stem and handlebars were replaced by the Handle-Drive levers linked together and a chain that ran down to a freewheel on either side of the front hub. A tensioning device maintains tension in the chain. We used a tandem hub with freewheel threads on both sides to accept a standard BMX freewheel on the right side and a BMX freewheel modified to freewheel backwards on the left side. Although the prototype was quite heavy and somewhat crudely made, learning to ride it and using the Handle-Drive system was surprisingly easy. Using the levers and timing the power stroke of the arms in relation to the legs was very intuitive. One could learn very quickly the action of the arms going up and down—it took us approximately ten minutes or so to get our actions coordinated and to begin to deliver power effectively to the front wheel.

To match the front drive ratio to the rear drive ratio, we designed and built a microprocessor system that sensed the speed of the bike and used this information to drive two stepper motors that adjusted the lever ratio of the front drive levers. The system worked well under no load but was not powerful enough to shift the gears reliably under power.

Luckily, a perfect match of front and rear drive ratios is not required to achieve synchronization of the arms and the legs. The reason is that the stroke of the front levers is not fixed. That is, the rider is not required to use all of the available travel and can control the length of stroke of the levers. Much in the same way that a rower can synchronize with another rower in the boat, the rider can synchronize the hand action with the feet by simply using a shorter or longer stroke on the front drive train (i.e., changing hand direction when the feet direction changes regardless of whether the entire stroke of the levers has been used up). If the synchronization is lost, the rider can simply stop stroking with the hands and wait until the legs are in an appropriate position to start up again. As long as the front drive ratio is within a range such that the travel limits of the lever are not encountered, and sufficient hand-stroke is allowed, wheel creep is not a problem.

Although the testing we did was limited due to the fragility of the first prototype, the results were very promising. The riding that we did do showed that the Handle-Drive system worked surprisingly well—better than we had hoped. Synchronization between the hands and feet was not a problem, even with the front-gear-changing system only partially functioning. On slippery uphills, the extra traction resulting from having both wheels driving was a definite advantage and perhaps the best feature of the system. Often, the ability of mountain bikers to climb up muddy or loose hills is limited by traction of the rear wheels. It remains to be seen whether such a system can indeed allow a rider to exert more power over short and long periods of time.

Some drawbacks of the prototype were the play between the two levers because of our crude connection system, and the rather ominous appearance of the long, protruding handlebars to pedestrians who happened to be standing in the way.

CONCLUSIONS AND RECOMMENDATIONS

In conclusion, we feel that the prototype we built has proven that the concept is viable and is a big step towards a design that might eventually be marketable. The synchronization of the hands and legs in the way that we have chosen seems to be quite effective, intuitive, and practical for use on a bicycle. In retrospect, a robust mechanical system for changing the front-drive ratio might be better suited to mountain-bike applications than an electronic system. As mentioned previously, the drive ratios do not have to be perfectly matched. Use of bevel gears to connect the left and right-hand levers, and lighter weight but durable components would improve the performance of the bike and the drive system. Finally, a lock-out that locks the front-drive handles into position so that the bike handles like a standard bike would be a nice feature that could be easily achieved with a gear interlock. In general, however, we are happy with the major design decisions we made and we are especially pleased with how well the system seems to fit with the natural movements of a cyclist.

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CORRECTION

Correction to: Reiser, R. F. & Peterson, M. L. (1998). Lower-extremity power output in recumbent cycling: a literature review. *Human Power* 45, pp. 6–13.

While reviewing Too (1994), the authors noticed an error. In this study cyclists were tested for anaerobic power output in three different recumbent positions, all with a body configuration of 105°. The torso angles, as determined by the backrest angle, were at 60, 90, and 120° with the hip orientations at −15, 15, and 45°, respectively (fig. 6). This was reported correctly in the review article. However, the power-output results were switched between the 60 and 120° torso-angle positions. The results then indicated that power output was similar for the two positions with the hips elevated above the bottom bracket and significantly greater than the power output in the position with the hips located below the bottom bracket (table 2). This led Too to conclude that the effects of gravity do play a small role in anaerobic power output with these effects increasing when the hips are below the pedals. This low hip position results in gravity pulling the legs away from the pedals during the power stroke portion of the pedal cycle. Gravity then assists the legs during the recovery phase, opposite of the effects of gravity when the hips are above the bottom bracket. This could place slightly different loads on the working musculature, causing the differences in power output between the positions tested.

Since the gravitational effects on a cyclic activity sum to zero (what is gained in one phase of the activity from gravity is then lost in a different phase) and the peak-power output is measured when the working musculature is in a non-fatigued state (minimizing the effects of slightly altering the loads on the musculature), there may be other factors involved that produce these significant differences in power output. One possible interaction that might cause differences in power output between a position with the hips above the bottom bracket and one below is in the foot-to-pedal interface. Toe clips, as were used in this study, have been shown to be a relatively sloppy interface (see foot-to-pedal interface articles referenced in the review article). The problems with the toe clips could be increased when the hips are below the bottom bracket, placing the foot effectively underneath the pedal during the entire pedal stroke. With the foot underneath the pedal, the toe clip may not provide adequate support to the foot-to-pedal interface which would result in reduced power output.

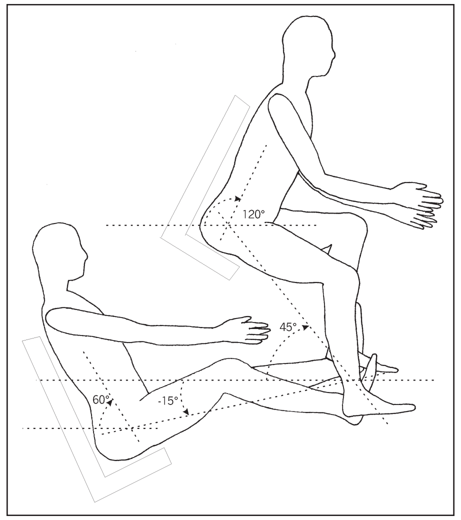

Figure 6. Range of test positions utilized by Too (1994): hip orientation of 15 to 45° in 30° increments. The torso angle was adjusted with hip orientation from 60 to 120° in order to maintain a 105° body configuration.

The numerous possibilities for why the cycling position with the hips below the bottom bracket are less powerful than the hips-above positions demonstrate how complex the system is that we are trying to understand. It also shows the need for more research in this area so that improvements may be made in the area of human-powered vehicles.

Additionally, hip orientation was referenced by Too based on seat-tube angle which is slightly different than the line between the hip joint and the bottom bracket. However, these two methods to determine hip-to-pedal orientation should be very similar (within a couple of degrees).

Also, since publication it has come to our attention that the speed record in the conventional riding position is above 50 mph. The current record of 51.29 mph was set by Jim Glover in a fully-faired Moulton AM7 at the 3rd IHPV Scientific Symposium in Vancouver [Expo 86 IHPSC] on 29 August 1986. Apologies to Jim and all the people who worked on that project.

Table 2 (corrected)

Hip orientation (degrees)	–15	15	45
Torso angle (degrees)	60	90	120
Body configuration (degrees)	105	105	105
Peak power (W/kg BM)	11.68	12.29	12.14
Average power (W/kg BM)	8.73	9.27	9.00
Fatigue index (%)	46.1	44.3	46.0

TECHNICAL NOTES

SUMMARIES OF PAPERS
by Danny Too

(Editor's note: These summaries were given by Danny Too on the hpv mailing list after Raoul Reiser and M. L. Peterson had discussed some of his papers in our last issue. He kindly agreed to these summaries being reproduced here. We are repeating figure 1 from the paper by Reiser and Peterson, p. 6, to illustrate the various angles that are referenced. —Dave Wilson)

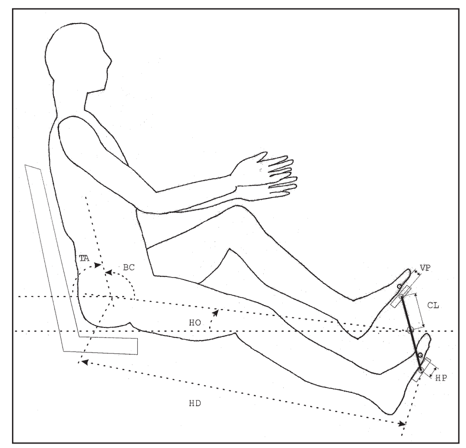

Figure 1.Geometrical variables which must be defined to completely describe the cycling position of the rider: hip orientation (HO), torso angle (TA), hip distance (HD), crank-arm length (CL), and horizontal (HP) and vertical (VP) foot position, as well as the foot-to-pedal interface (not shown). Body configuration (BC), which may be deduced from TA and HO is also included to help describe the cycling position.

Too, D. (1990). The effect of body configuration on cycling performance. In E. Kreighbaum & McNeill (eds.), *Biomechanics in Sports VI* (pp. 51–58). Montana State University, Bozeman, Montana.

This study examined the effects of changes in hip angle (while keeping the knee and ankle angles the same) on cycling duration and work output. Hip angles were manipulated by a systematic change in seat-tube angle (as determined from a vertical line passing through the crank spindle). Five seat-tube angles were examined: 0, 25, 50, 75, and 100 degrees. For each seat-tube angle tested, the trunk was always kept perpendicular to the ground, and the seat-to-pedal distance adjusted to maintain the same distance. Sixteen subjects were tested in each of the five seat-tube angles. The tests were on a Monark bicycle ergometer, with increasing load or cadence every three minutes until exhaustion. The results revealed a parabolic curve in cycling duration with changes in seat-tube angle from 0 to 100 degrees. The longest duration occurred with the 75-degree seat-tube angle and the trunk perpendicular to the ground. This same result was found regardless of whether a trained cyclist, triathlete, or untrained subject was tested. This corresponded to a minimum hip angle of 56.5 degrees and a maximum hip angle of 97 degrees during one pedal cycle. It may not be the seat-tube angle that is as important as the joint angles. Changes in joint angles affects muscle length and other variables that interact to produce force and power. Changing the seat-tube angle changed the minimum and maximum hip angle during a pedal cycle, but did not change the range of motion.

This changes where the fatigue is felt. In an upright position (e.g., seat-tube angle of 25 degrees), the stress occurs more on the quadriceps. In a very low sitting recumbent position (e.g., seat-tube angle of 100 degrees), the stress occurs more on the gluteal (buttocks) region. The 75-degree seat-tube angle apparently distributes the stresses more evenly over the quadriceps, hamstrings, and gluteal region, thereby reducing local fatigue in any particular muscle group (which may be one of the limiting factors to prolonged cycling performance). A change in seat-tube angle apparently changes the points at which the various muscle groups are active and inactive during a pedaling cycle (although there is no change in the pattern or duration of activation). This was based on another study I had published (titled: The effect of hip position/configuration on EMG patterns in cycling). This has major implications regarding efficiency and force and power generation.

Conclusion: the optimal mean hip angle that maximizes cycling duration and total work output with incrementing workload is 77 degrees, with a minimum of 57 degrees, a maximum of 97 degrees, and a hip range of motion of 41 degrees. This was found with a seat-tube angle of 75 degrees with the trunk perpendicular to the ground, and a seat-to-pedal distance of 100% of leg length (as measured from a standing position from the greater trochanter to the ground).

Too, D. (1991). The effect of hip position/configuration on anaerobic power and capacity in cycling. *International Journal of Sports Biomechanics*, 7(4), pp. 359–370.

This study was, in essence, the same as the previous one (summarized above). The difference was that testing was done anaerobically (with a 30-second all-out power test, using a resistance based on body mass) instead of aerobically. This information is more appropriate for those constructing HPVs to set new speed records, as opposed to distance/endurance records.

The purpose of this investigation was to determine the effect of systematic changes in hip position/configuration, while maintaining an upright trunk orientation, on cycling peak anaerobic power and anaerobic capacity. Fourteen male recreational cyclists (age 21–32) were each tested in four hip positions (25, 50, 75, and 100 degrees), as defined by the angle formed by the seat tube and a vertical line. Rotating the seat to maintain a backrest perpendicular to the ground induced a systematic decrease in hip angle from the 25- to the 100-degree position. The Wingate Anaerobic Cycling Test was used on a Monark Cycle ergometer with a resistance of 85 gm/kg of the subjects' body mass (5.0 joules/pedal rev/kg BM).

Repeated measures MANOVAs* and post-hoc tests revealed that (1) anaerobic power (AP) and anaerobic capacity (AC) in the 75-degree hip position was significantly greater than that in the 25- or 100-degree position ($p < .01$); and (2) a second-order function best describes the trend in AP and AC with changes in hip position ($p < .01$).

It was concluded that there is/are some hip position(s)/angle(s) that will maximize cycling performance as determined by AP and AC and that an intermediate position (50–75 degrees) produces the greatest power. To fully address the issues in this area require further research involving a series of investigations where selected body position, configuration, and orientation variables are systematically manipulated.

*MANOVA – Multiple Analysis of Variance (used when comparing 3 or more groups and there is more than one measured/dependent variable [e.g., peak power output and average power output])

In summary, the same parabolic trend was also found for anaerobic cycling performance. The 75-degree seat-tube angle resulted in the largest peak power (during any 5-second interval) and the largest average power over the 30-second test. This was true whether a trained cyclist was used or an untrained subject. The 0-degree seat-tube angle was not used because subjects were unable to complete the test with the load selected.

Too, D. (1994). The effect of body orientation on power production in cycling. *The Research Quarterly for Exercise and Sport*, 65, 308–315.

This study, based on the results obtained from the paper just summarized on anaerobic power and capacity, was a continuation to determine the most effective cycling position to maximize power production. Since a 75-degree seat-tube angle (with the trunk perpendicular to the ground – 90 degrees) apparently resulted in the largest peak and mean power, this seating position was selected. The purpose of this study was to manipulate the trunk orientation relative to the ground while maintaining the same 75-degree seat-tube angle, and maintaining the same hip, knee, and ankle angles. To accomplish this, the entire cycling apparatus was rotated forward 30 degrees to obtain a trunk angle 60 degrees to the ground, and rotated backwards 30 degrees to obtain a trunk angle 120 degrees to the ground. Differences in cycling performance between the 60, 90, and 120 degree trunk angle can be attributed only to differences in trunk angles and not to changes in hip, knee, or ankle angles. This was a major flaw in the following two studies:

"The influence of body position on maximal performance in cycling.", Welbergen E. and Clijsen L.P.

"The effect of posture on the responses to cycle ergometer exercise." Begemann-Meijer M.J. and Binkhorst, R.A.

These two studies did not control for joint-angle changes when seating position or trunk angles were changed. Therefore, it is unknown whether differences in cycling performance (if differences were found) were attributed to changes in the seating position, joint angles, trunk orientation, or an interaction of all of these variables.

ABSTRACT

The purpose of this investigation was to determine the effect of three different trunk angles (60, 90, and 120 degrees relative to the ground) on power production of 16 male recreational cyclists (age 20–36) when the hip, knee, and ankle angles were controlled. Wingate anaerobic tests were performed on a modified Monark cycle ergometer against a resistance of 85 g/kg of the subjects' body mass (5.0 J/crank rev/kg BM). The order of test conditions was randomly assigned, with a minimum of 24 hours between sessions. A DM MANOVA and post-hoc tests revealed that peak power at the 60- and 90-degree trunk angle was significantly greater than that at the 120-degree angle, and mean power in the 90-degree angle was significantly greater than that at the 120-degree angle. It was concluded that changes in cycling trunk angle may affect peak power and mean power.

The results of this study would suggest that, although a reclining position (120-degree trunk angle) may be more comfortable, it is not effective in power production. The reason? A reclining position where the feet are above the hips forces the cyclist to overcome not just the ergometer resistance, but also the weight of the legs. An analogy to this would be to cycle in an completely inverted position. In this position, it would be more effective to pull on the pedals, using gravity and the weight of one's legs (than to push against the pedals to overcome the leg weight and gravity). A neutral position (90-degree trunk angle to the ground) or one where the leg weight assists in pushing the pedals (60-degree trunk angle) would be more effective than a position where one has to overcome gravity. This clearly explains why recumbents (especially those where the pedals are above the hips) are not effective in climbing hills.

This study dealt with peak power production in a 30-second test because another study that I had conducted aerobically (cycling duration) with the same three trunk angles revealed no significant difference between all three angles. An EMG study, examining possible differences in muscle activity patterns with these three trunk angles revealed no differences in muscle timing, patterns, or duration among these three trunk angles. Unfortunately, quantitative data were not available, and may have supported the "overcoming leg weight" explanation of why the 120-degree trunk angle was less effective.

Too, D. (1994). The effect of body position/configuration and orientation on power output. In C. R. Kyle, J. A. Seay, & J. S. Kyle (eds.), *Fourth International Human Powered Vehicle Scientific Symposium Proceedings* (pp. 59–65). Cycling Research Association, Weed, CA.

This study is really a compilation and presentation of the data from the previous two studies on manipulation of seat-tube angle (presented as experiment 1) and manipulation of trunk angle (presented as experiment 2). See the preceding two summaries for the results and discussion.

Too, D. (1996). Comparison of joint angle and power production during upright and recumbent cycle ergometry. In J. A. Hoffer, A. Chapman, J. J. Eng, A. Hodgson, T. E. Milner, & D. Sanderson (eds.) *Proceedings of the Ninth Biennial Conference and Symposia of the Canadian Society for Biomechanics* (pp. 184–185). Simon Fraser University, Burnaby, British Columbia, Canada.

This study compared the 75-degree seat-tube-angle recumbent-cycling position with the standard upright-cycle ergometer position. Hip, knee, and ankle angles were compared; as was peak power and average power during the 30-second power test. All subjects were tested in both the recumbent and upright positions. The load selected was based on each subject's body mass. The recumbent position was found to result in significantly greater absolute and relative power (relative to body mass) in peak power and average power, when compared to the upright position. Only the minimum and maximum hip angles between the upright and recumbent positions were significantly different. There were no significant differences in the minimum, maximum, and range of motion of the knee and angle between the recumbent and upright position. This would suggest that differences in power production between the upright and recumbent positions were attributed to differences in hip angles.

Too, D. (1998). Comparisons between upright and recumbent cycle ergometry with changes in crank-arm length. *Medicine and Science in Sports and Exercise*, Vol 30, No 5, S81 (Abstract).

This study is a continuation of the preceding study, comparing the upright and recumbent position, but also manipulating crank-arm length. The crank-arm lengths examined were 110, 145, 180, 230, and 265 mm.

This investigation was: (1) to compare power production between an upright (UP) and recumbent (REC) cycling position with changes in crank-arm length (CL); and (2) to examine how joint angles (JA) change. Six male subjects (ages 24–35) were all randomly tested on a Monark bicycle ergometer (Model 814E) at 5 CL (110, 145, 180, 230, 265 mm) in an UP and REC position. For each CL in the UP and REC, the seat-to-pedal distance was standardized, the subjects' trunk kept perpendicular to the ground and pedal toe-clips worn. A 30-second Wingate anaerobic cycling test was used, with a resistance of 85 gm/kg of each subject's body mass (5.0 joules/pedal rev/kg BM) and at least 24 hours between tests. In each condition, JA for the hip, knee, and ankle for one pedal revolution were measured. Peak power (PP) and mean power (MP) were determined by a SMI Power Program for 5 and 30 sec, respectively. The mean JA, PP, and MP in the UP and REC position with changes in CL are as follows (see table on following page).

With increasing CL, there is: (1) a decrease in mean JA; with the JA for the REC less than for the UP; (2) a curvilinear trend for PP and MP in the UP; and (3) a decreasing and a curvilinear trend for PP and MP, respectively, in the REC. Paired t-tests between UP and REC with increasing CL revealed: (1) p = 0.04, 0.005, 0.001, 0.017, 0.099 for PP; and (2) p = 0.018, 0.026, 0.019, 0.019, 0.021 for MP. The data and results suggest that greater PP and MP in the REC position may be attributed to a more effective JA.

In summary, the recumbent position resulted in significantly higher mean power output with all five crank-arm lengths when compared to the upright position; and the recumbent position resulted in significantly higher peak power with all crank-arm lengths other than the 265 mm, when compared to the upright. Although this study revealed the highest peak power occurring with the shortest crank-arm length (110 mm), ergometer flywheel acceleration and deceleration was not accounted for (and if it was, slightly different results would be found).

The interaction between crank-arm lengths and cycling performance is much

Danny Too: Table showing differences depending on crankarm length (CL)

CL (mm)	110	145	180	230	265
	hip/knee/ank	hip/knee/ank	hip/knee/ank	hip/knee/ank	hip/knee/ank
UP (deg)	142/124/111	137/119/107	134/113/108	130/109/106	123/105/112
REC (deg)	80/115/100	80/109/96	77/105/94	75/95/93	73/94/91
POWER	PP / MP	PP / MP	PP / MP	PP / MP	PP / MP
UP (W)	880 / 546	913 / 690	949 / 741	859 / 697	843 / 683
REC (W)	1123 / 757	1103 / 786	1093 / 806	979 / 772	896 / 748

more complex, since changes in crank-arm length affect not only hip angles, but also knee angles. There are also other variables and factors to consider, including the interaction between muscle force-length, and force-velocity-power relationships; since there apparently is an interaction between crank-arm length, load, and cadence.

Currently I have two papers related to crank-arm length in review for publication:

1. Too, D., & Landwer, G. The effect of pedal crankarm length on joint angle and power production in upright-cycle ergometry. Submitted to *Journal of Sport Sciences*.
2. Too, D. The effect of pedal crankarm length on joint angle and power production in recumbent-cycle ergometry. Submitted to *Ergonomics*.

I am currently analyzing data for a paper, comparing the power production between an upright and recumbent position with changes in crank-arm length. The same subjects were used for all test conditions in the upright and recumbent.

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DANNY TOO RESPONDS TO QUESTIONS

(Danny Too responded to some questions on aspects of his papers, and was gracious enough to allow us to publish them. Questions are shortened in several cases. —Dave Wilson)

Question: John Riley (j.riley16@genie.com) wrote: "Out in the real world things get very complex and with unfaired bikes, people manipulate the position to get better aerodynamics. That said, the Tour Easy and Rans Stratus come close to matching your optimum position and they do have a reputation for good performance. The BikeE is also close, but does not have a good reputation for performance. The BikeE does apparently perform better when the rider hunches forward, and I think the rider also hunches forward in the fully faired Tour Easys that have won so many races. Your optimal position seems to have a riding angle (angle formed by a line from the BB to the seat base and a line up the seat back) of 115 degrees. Perhaps a slightly tighter riding angle, with the BB still below the seat, might be even better, especially for anaerobic work. The tighter riding angle can constrict the lungs and so might not be best for aerobic work."

Danny Too: There are many factors that affect cycling performance.

A cycling position that maximizes power production and cycling effectiveness, but also happens to maximize aerodynamic drag, may not necessarily maximize cycling performance (as defined by maximal velocity or minimal time to cover a pre-set distance). The optimal cycling position may very well result in a trade-off between the two. Rider conditioning and training in any given position will also be a factor.

But I would speculate that recumbents with similar cycling positions will not necessarily result in similar cycling joint angles and kinematics during a pedaling cycle. This would explain why different recumbents with similar cycling positions may not result in identical cycling performance. This would also explain why "hunching forward" in certain vehicles may improve performance. This "hunching forward", probably results in more effective hip and knee angles in the production of force. Recumbent cycling positions are as exclusive and diverse in trunk angles, joint angles, seat-tube angles, and crank-arm lengths as the vehicles themselves (and the people who design them). This, I believe, is what makes comparisons among recumbents very difficult. Each recumbent vehicle available on the market is unique in some fashion, and it is the interaction of a multiple of variables (trunk angle, joint angles, etc.) that ultimately results in performance. Therefore, to compare different recumbent vehicles is like comparing apples with oranges.

What I have attempted to do in my research is to eliminate all these interactions and confounding variables by systematically manipulating one variable while controlling for all the others. This, then, provides objective information regarding trends and patterns with extreme manipulations in crank-arm lengths, seat-tube angles, joint angles, trunk angles, etc.

Question: Cyril Rokui (croku@juno.com) wrote: "Thanks very much for the summary of your papers. I found it to be very interesting reading and may incorporate some of the findings in future bikes I intend to build. Have you done longer-duration (30 minutes or one hour) crank-arm-length studies that would simulate a bike ride rather than a very short test just for peak power? Also, I notice that mean power output is highest in the recumbent position for the 180-mm cranks and this was for 30 seconds vs. the 110-mm cranks at 5 seconds for the peak-power measurement. Does this mean that the 180-mm cranks are more efficient for long-term production of power?"

Danny Too: No, I have not examined longer-duration (30 minutes or 1 hour) studies with changes in crank-arm length. It may simulate a bike ride, but subject motivation would probably be a confounding variable affecting the results, and it would also be difficult to obtain subjects who would be willing to participate in such a study. However, I have collected data examining the effect of incrementing workload on cycling duration with changes in crank-arm length. I have not yet had the time to analyze the data.

First, a correction for flywheel acceleration and deceleration was not accounted for in that abstract. In the full manuscript (submitted to *Ergonomics*), this correction has been made and results in the 145-mm crank-arm length producing the highest 5-second power. Second, mean power, being highest for the 30-second test, would suggest that they are more efficient for long-term power. However, it is more complex than that. There appears to be an interaction between crank-arm length, pedaling rate and workload/resistance. When fatigue sets in (15 seconds into the 30-second test), pedaling rate starts to decrease. When pedaling rate is least during the last 5 seconds, the crank-arm length that results in the largest minimal power is the 230-mm crank-arm length. The 180-mm crank-arm length results in the largest mean power for the 30-second test, and the 145-mm crank-arm length will result in the largest peak power during the first 5 seconds of the test.

Question: Rolf Mantel wrote "Will you do more studies using cranks that represent a sampling of what's readily available in the marketplace e.g., 165, 170, 175-mm cranks? Even a small difference may be significant in racing or trying to set a speed record."

Danny Too: No, I will not be doing studies using cranks that represent a sampling of what's readily available in the marketplace (e.g. 165, 170, 175-mm cranks). I am using extreme (short and long) crank-arm lengths to observe the trend in performance that occurs, and to understand the mechanisms involved. It appears that it is not so much the length of the cranks that is important, as it is the joint angles of the lower extremities in producing power.

The difficulty with using the same individuals for repeated tests over a period of time is the training effect that would occur. The data with different crank-arm lengths would be confounded by the improvement in performance due to training. It would then be unknown whether performance differences with different crank-arm lengths are attributed to crank-arm lengths, a training effect, or both. To control for the training effect, the crank-arm-length test sequence needs to be randomized across subjects (i.e., a different crank-arm-length test sequence for each subject).

Question: Gary King wrote: "Though D. Too's experiments were probably very accurate, I don't believe they prove high-BB bikes (SWBs) are slower climbers than low-BB recumbents (many LWBs). Were his subjects using cleats? Did they know how to pedal high-BB bikes? The pull-back stroke is a very powerful stroke on these kinds of bikes. Also he showed that the upright position was not the most powerful position in the high-load situation (equivalent to climbing I assume). I read only the summary, but I guess the rig he used was fixed, rigid. In the real world the upright rider can sway the bike, use his arms to climb, centre his weight over each pedal etc. I suspect the results would have been very different if subjects were able to rock the test rig from side to side—only a slight amount would do it."

Danny Too: First, my experiments do not show or prove that "high-BB bikes (SWBs) are slower climbers than low-BB recumbents (many LWBs) or vice-versa. The experiments were never designed for that purpose.

They were designed to: (1) provide objective information regarding how cycling performance changes with systematic manipulations of different variables while controlling for all others; (2) provide objective and unbiased information that can be replicated and quantified by others; (3) provide information to designers in the development and construction of faster and more effective HPVs. How the data and results from my research are interpreted and used by others is not in my control.

My subjects did not use cleats, but used toe-clips. They were untrained recreational cyclists who did not know how to pedal high-BB bikes or who had any significant experience with recumbent bicycles—although some were engineering students involved in the development of HPVs. If trained cyclists (of uprights or recumbents) were used, the data would be biased and the results may very well have been different. This is due to specificity of training.

Subjects were not allowed to stand upright, sway the bike, shift weight, use the arms, etc., during testing in the upright positions (because the ergometer and seating apparatus are fixed structures, eliminating balance as a factor). If they were allowed, the results could very well be different, and then it would not be known whether differences in performances would have been attributed to the variable being manipulated, and/or to other uncontrolled variables that confounded the data.

Question: Akash Chopra writes: "Thanks for posting the summary of your papers. I do have one question regarding your claim in the paper 'The effect of body orientation on power production in cycling' where you state that: 'A neutral position (90-degree trunk angle to the ground) or one where the leg weight assists in pushing the pedals (60-degree trunk angle) would be more effective than a position where one has to overcome gravity. This clearly explains why recumbents (especially those where the pedals are above the hips) are not effective in climbing hills.'

"I would have thought that the majority of hill climbing would require aerobic effort (it certainly does where I live!) and that the performance would not be determined by peak power output. You mention that another study which was 'conducted aerobically…revealed no significant difference between all three angles.' This would suggest that the recumbent position is not responsible for any lack of hill-climbing performance (from the results of these papers, at least)."

Danny Too: Thank you for interest in my research and for the question you posed. The aerobic study to which you referred is

Too, D. (1989). The effect of body orientation on cycling performance. In W.E. Morrison (ed.). *Proceedings of the VIIth International Symposium of the Society of Biomechanics in Sports*, (pp. 53–60). Footscray Institute of Technology, Victoria, Australia.

In that study, there were no significant differences (statistically significant ones) between the 60-, 90-, and 120-degree body orientation. However the longest cycling duration was found with the 120-degree orientation, followed by the 90- and 60-degree orientation, respectively. Therefore, the trend in aerobic performance is similar to that found anaerobically. It is possible, with a larger sample size, statistical significance may be found. I am hoping someone will replicate my study to either support my results, or provide additional information.

Question: Sean Williams wrote: "Your abstracts did not state your position on the issue. I suspect that there is a decrease in power-output performance. I suspect the ergonomics of the recumbent position allows for greater endurance. Depending on leg mass, center of mass, how much of a change there is in vertical displacement and where in the cycle (in terms of the power stroke) it occurs there appears to be a loss of about 8% of available power.

"Assuming: almost all power is given to the pedals by the pushing leg; a 50-mm vertical rise in the centre of mass; all vertical rise in CoM is during power stroke; a 10 kilo leg; 100 rpm over one minute, then the total energy = 0.05 * 10 * 9.8 * 100 = 490 joules per minute = 8 watts. Given that both legs are doing this, 16 watts is removed from the power stroke just to lift the leg.

"The upright position gives this loss on the return stroke so different muscles are used than to provide push on the pedals.

On a recumbent going uphill the same muscles are used to lift the leg as to push the pedals. The ergonomic factors rather than power factors come into play on long bike rides. I would like to hear your opinion as you have done research on the subject and my opinion is merely supposition."

Danny Too: There are many fac tors that will affect cycling performance and there is a very complex interaction among these variables. Engineers often approach cycling performance from an aerodynamic and mechanical perspective whereas I am examining performance from a kinesiological perspective (and attempting to bridge the gap between man and machine by using an interdisciplinary approach in my research).

The change in cycling performance from manipulations in cycling position, orientation, crank-arm length, seat-to-pedal distance, etc., is not attributed just to mechanics and aerodynamics, but also to a complex interaction between muscle length (of single- and multiple-joint muscles), muscle moment-arm length, and the muscle tension-length, and muscle force-velocity-power relationships to produce force/torque/power. To truly maximize performance, all factors have to be considered and tradeoffs may have to be made.

My research is an attempt to understand how a systematic manipulation of each of these variables (while controlling for all others) will affect performance and the mechanisms involved in force, torque, and power production.

Based on what you have presented, your assumptions may very well be true. However, I suspect the change in joint angles may be a more important factor affecting performance.

Question: Raoul F. Reiser wrote: "I am hoping you could shed a little additional light on your subject populations from a couple of your previous studies? Specifically, in 'The effect of hip position/configuration on anaerobic power and capacity in cycling' (1991) and 'The effect of trunk angle on power production in cycling' (1994) you refer to the subjects as recreational cyclists. Do you recall what form of recreational cycling they used most often? Were they recreational road cyclists, off-road cyclists, track cyclists, or other?

"I ask because it seems that the position that a person uses for cycling might influence the optimal cycling position and the above three styles of cycling require slightly different body configurations from the rider."

Danny Too: The subjects, in general, were recreational road cyclists. There were a couple who also rode mountain bikes (but not competitively). In the 1991 study ("The effect of hip position/configuration on anaerobic power and capacity in cycling"), the type of cyclist tested would probably not have significantly affected the results. In that study, I had also tested one competitive road cyclist and one competitive triathlete. I did not include their data in the study, but their data (with changes in hip position/angle) revealed the same trend.

Question: Cyril Rokui wrote: "With so many variables to consider, no wonder there are so many opinions about optimal seat/crank position. Even if the seat-post angle was constant for the tests, because of variation in human anatomy (big *vs.* small buttocks, tilt of pelvis, curvature of spine, length of leg bones, etc.) the hip/leg angle would be different for many riders sitting in the same seat. I wonder if a variation of 50 mm in hip-joint height would make a measurable difference—different enough for people riding the same bike to experience different levels of exertion for the same speed?

"Figuring optimal seat/crank position for upright bikes must have been trivial in comparison because of the relatively direct contact of the seat with the sit bones (ischial tuberosities) producing a much smaller amount of variability.

"I have come upon another puzzling observation. I tested my heart-rate monitor using a high-bottom-bracket (BB 215 mm above seat bottom) recumbent 'mag' trainer and an upright bike on a mag trainer. On the 'bent trainer at 150 bpm I was starting to feel uncomfortable and was at my aerobic threshold at 160. I then rode the upright and at 173 bpm was not winded. I don't understand the performance difference. Could it be that the 'bent position constricted my diaphragm and reduced my lung capacity and upright position opened up the rib cage and diaphragm? I know that this is not your area of study but I find it to be an interesting observation. Maybe others with a similar setup and a heart-rate monitor would like to try their own tests and see if they get similar results."

Danny Too: Yes, it is very possible that a 50 mm variation in hip-joint height (or less) would make a measurable difference for people riding the same bike to experience different levels of exertion for the same speed. A 50-mm variation in crank-arm length will definitely have an effect on cycling performance. However, it may not have the same effect for everyone (or affect everyone to the same extent). This is the reason why research studies are conducted with groups (instead of individuals) to find a general trend (if there is one), and statistical analysis undertaken to determine what is the probability that differences in performance are attributed to chance (or random variability), or attributed to the manipulated variable.

First, you have not indicated whether you were using the same workload in both the recumbent and upright position and obtaining different heart rates (or whether these heart rates were obtained with different workloads in the different positions). Second, are the heart rates you are recording, maximal heart rates or submaximal ones? Third, I suspect your recumbent position is not only different in trunk orientation with respect to the ground, but also in joint angles and joint range of motion during the pedal cycle. If this is the case, then you have a confounding variable, and will not be able to determine whether differences in heart rate between the upright and recumbent positions are attributed to the change in trunk orientation, or joint-angle differences (affecting power production and efficiency), or both.

On the assumption that your joint kinematics are similar during the pedaling cycle in both the recumbent and upright position, then differences in heart rate (and cycling performance) would be attributed to trunk orientation and blood-flow hemodynamics. Regardless of whether this is the case, the research literature shows that heart rate will be lower when cycling in the supine position than when cycling in the upright when the same submaximal workloads were used (although no information was provided whether the joint kinematics in the supine and upright were the same). The reason? It would appear that a certain cardiac output is required to supply blood to the working muscles for a given workload. Cardiac output is a function of stroke volume (the amount of blood pumped from the heart with each beat) and heart rate (i.e., cardiac output = stroke volume ∞ heart rate). For any given cardiac output, the greater the stroke volume, the lower the heart rate. This is the reason why endurance athletes have a lower resting heart rate. For the same cardiac output at rest, an endurance athlete will have a greater stroke volume with each heart beat (when compared to a sedentary individual) and hence a lower resting heart rate (which translates into less heart beats, and work for the heart over the course of a lifetime). In a supine position, venous blood flow is facilitated and returns to the heart much more easily, fills the heart more, resulting in a greater stroke volume, and hence a lower heart rate for any given cardiac output (when compared to an upright position). The maximal heart rate also appears to be less in a supine position than in an upright position. Therefore, your heart rate at 160 bpm in a supine position may be at the same percentage (e.g., 90% of your supine maximal heart rate) as your 173 bpm heart rate in an upright position (e.g., 90% of your upright maximal heart rate).

As for whether "the 'bent position constricted my diaphragm and reduced my lung capacity and upright position opened up the ribcage and diaphragm?"

It is possible. A study by Faria *et al* (1978) comparing a top-bar and drop-bar cycling position (on an upright), reported the maximal oxygen uptake for the drop-bar position to be greater than that attained for the top-bar position. A top-bar position was described as sitting semi-upright on the saddle with the hands resting on the uppermost portion of the handlebars, while a drop-bar position was described as sitting in the saddle while assuming a deep forward lean, with the hands resting on the drop portion of the turned-down handlebars. The differences in maximum oxygen consumption was attributed to: (1) the activity of a larger muscle mass (greater use of the arm, shoulder girdle, and lower back muscles) in the drop-bar position; and (2) the greater forward body lean angle in the drop-bar position which appears to relieve the weight of the arms and shoulder girdle from the thorax. This reduced weight plus the suspended chest is believed to ease chest expansion, thereby enhancing pulmonary ventilation potential and possibly decreasing the energy requirement for respiration. So reduction of lung capacity and constriction of your diaphragm in a recumbent position is a possible explanation for a decreased work capacity. However, I have not seen any literature that has examined the accuracy and validity of this statement and explanation. It is also unknown as to whether the greater lean in the drop-bar position altered joint angles and allowed a more mechanically advantageous position to produce force when compared to the top-bar position.

If you are interested in references related to heart rate, stroke volume, cardiac output, oxygen consumption, pulmonary ventilation, and work output during rest and exercise between supine and upright position, e-mail me and I will send you an attached text file reference list.

SOME COMMENTS ON THE EFFECTS OF "INTERFERENCE DRAG" ON TWO BODIES IN TANDEM AND SIDE-BY-SIDE

Mark Drela and Jim Papadopoulos

(Editor's note: This was contributed to a mailing list "Hardcore bicycling science" organized by Jim Papadopoulos, and has been edited and reproduced here with Jim's and Mark Drela's permission. Jim opened the discussion by commenting on the relevance of data in a book to pairs of HPVs, including bicycles and riders, and Mark gave his explanation of the theoretical background. —Dave Wilson)

Jim Papadopoulos: One of the most outstandingly useful books on fluid dynamics measurements and theory is *Fluid-Dynamic Drag*, written and published by Sighard F. Hoerner. (For the uninitiated, 'fluid' includes not only water but air, so this book bears strongly on the aerodynamic resistance of a bicycle and rider.)

Recently, I chanced on chapter 8, "Interference Drag", and wanted to share a little of what I found there. Note that the measurements relate to idealized, smooth-surfaced shapes, and not actual riders. But I think they are valuable for suggesting what might possibly happen, perhaps to a different degree, in the real world.

For example, figure 1 concerns two disks, broadside to the direction of travel, with one sheltered behind the other (drafting). Although the drag force on the forward disk is not affected by its follower, the follower is actually 'dragged along' if it is fairly close (1.5 diameters). If the two disks were connected together, for example like riders on

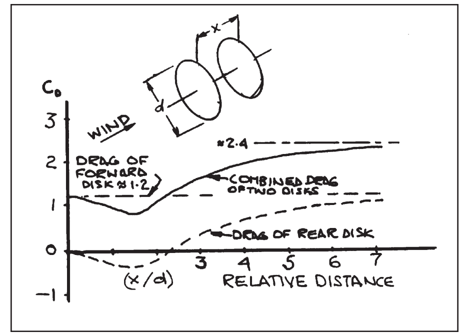

Figure 1. Interaction between two disks placed one behind the other.

the same tandem, the second would effectively perform a streamlining function for the first. If the analogy (of a 'disk' to a 'rider') held good, a tandem would need less power to propel than a single bike.

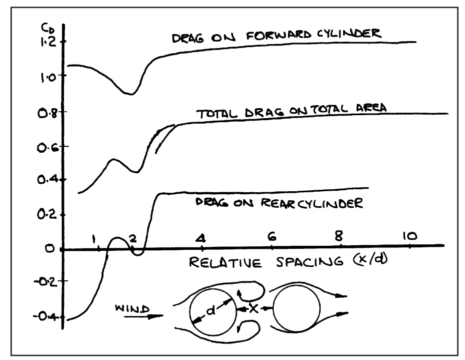

Figure 2. Drag coefficients of two circular cylinders, one placed behind the other.

Figure 2 relates to two round cylinders, roughly like one very tall runner following another. When the gap is about two diameters, the lead runner actually experiences about a 15% reduction in drag. The rear runner, in that position, experiences approximately zero drag. When the separation increases to four diameters, the lead runner loses any benefit, while the rear runner's drag is about 25% of the solo-runner value.

In figure 3, streamlined cylinders (like airplane tails or upright HPVs) are treated. When they are close, the drag on the rear

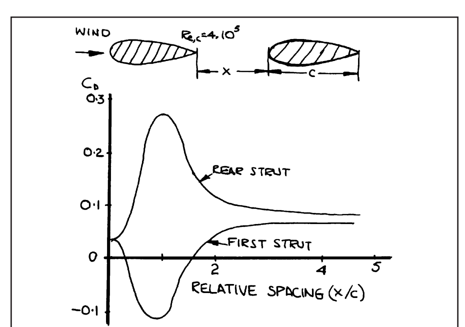

Figure 3. Drag of a pair of strut sections, one behind the other, in tandem.

unit is quadrupled, while the front receives a push, equivalent to the increase in rear-unit drag.

Figure 4 has little relevance to HPVs and is not given here. In figure 5, side-by-side cylinders show a 25% increase in drag when touching, and a 15% decrease when they are almost exactly one diameter apart.

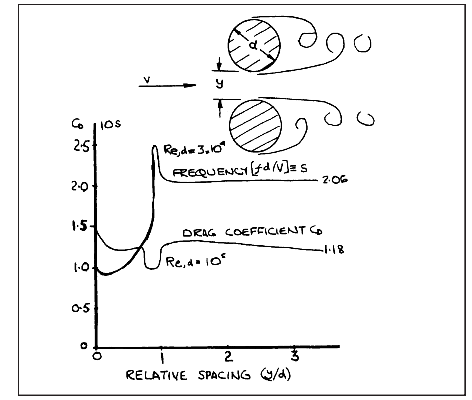

Figure 5. Drag (and vortex-street frequency) reference of a pair of circular cylinders placed side-by-side.

Finally, in figure 6 streamlined cylinders each experience as much as a tripling of drag when they are side-by-side. The gap must be greater than three widths for the effect to be negligible.

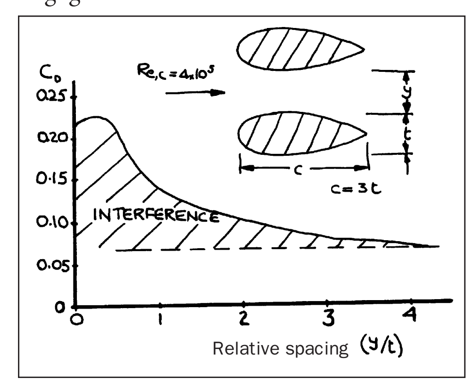

Figure 6. Drag of a pair of struts, one beside the other.

The speed and size range for fluid flow results are usually defined by "Reynolds number", which is the "speed" times "size" divided by "kinematic viscosity". Kinematic viscosity for air at room temperature is about 1.6E-4 ft^2/s or 1.5E-5 m^2/s, depending on whether you measure in feet and feet/s, or meters and m/s.

Most of the results were from experiments in the range of E5 to 5E5—this is just below that of high-speed cycling. (Evidently, they were aimed at small [30-mm] airplane parts travelling at high [150-200 m/s] speeds.) A cyclist's 'size' is from 600 mm (width) to 1200 mm (height); when streamliners are considered, the length (2 m) is typically used. A Reynolds number of 5E5 would correspond to a speed of 12 m/s (27 mph) for a 600-mm size, or 6 m/s (13 mph) for a 1200-mm size.

—*Jim Papadopoulos (bicyclengr@newrock.com)*

Mark Drela: In various two-body combinations that can be considered likely for two bicyclists, HPVs, or components such as frame tubes and wheels, there are no fewer than seven distinct physical effects that can conspire in various proportions to produce the "anomalous" drag behavior indicated in Hoerner's valuable book, *Fluid-dynamic drag*. I've summarized these effects below, and their primary influence on each body's drag. To reduce words, let's call the bodies in a tandem configuration A and B, where A is the upstream body and B the downstream. When bodies are in a side-by-side orientation they will both be C. Here are the seven cases.

Case 1: B sees reduced dynamic pressure in A's wake. The consequence is that B has reduced drag, while A is unaffected.

Case 2: A and B feel each other's pressure field. The changes in forces experienced by the two bodies depend a lot on details of body shapes and proximity—we can't generalize. This is a "short range" effect that acts only within a body dimension or so.

Case 3: B's laminar boundary layers are forced to become turbulent by the turbulence in A's wake. In this case, A is unaffected, while B has decreased drag. (B must be subcritical— the Reynolds number must be sufficiently small, but large enough that turbulent boundary layers are possible).

Case 4: B's laminar boundary layers are forced to become turbulent by the turbulence in A's wake. This seems to be the same as case 3, but in this case A is again unaffected, while B has increased drag. For this to occur, B must be laminar initially, and the Reynolds number must be sufficiently large.

Case 5: B suppresses A's vortex shedding. (This applies only to the subcritical Reynolds-number range.) The consequence is that A has reduced drag, while B is mostly unaffected.

Case 6: C's surface pressure is changed by its partner. Such a change is likely to cause increased boundary-layer separation and higher drag.

Case 7: C's vortex shedding is changed by its partner when in the subcritical Reynolds-number range. This is likely to reduce drag, since disturbing the vortex shedding in any way usually reduces drag.

We are reproducing simplified versions of Hoerner's graphs. In figure 1, two disks in tandem, we see cases 1 & 2. In figure 2, two circular cylinders in tandem, the effects of four cases, 1, 2, 3 & 5, can be seen at different points, resulting in somewhat wild behavior. In figure 3, two airfoils in tandem, cases 2 and 4 occur. In figure 5, two circular cylinders side-by-side, cases 6 and 7 are found. Lastly, in figure 6, two airfoils side-by-side, just case 6 applies.

Here are my guesses on the dominant effects on cyclists.

Case 1 is normal drafting with two bikes or HPVs. Cases 1+2 are found in close "drafting" on a short tandem. Case 6 is close side-by-side riding

The effects on bike components seem to be that cases 1+3 would apply to A being the down tube and the front wheel, and B the seat tube. Cases 1+2+5 would apply for A being the seat tube, and B the rear wheel. Case 6 would be for C as a round fork blade or a seat stay close to a wheel disk, or for C being a leg adjacent to a Trimble frame (one with the tubes connected by a membrane).

In the last two items, the major problem is likely to be with the boundary layer on the surface rather than on the cylinder.

—*Mark Drela (drela@orville.mit.edu)*

IHPVA RECORD WIND RULES: A PARTICIPANT'S PERSPECTIVE

by Paul Buttemer

In late July 1998, Team Varna, consisting of builder George Georgiev and riders Sam Whittingham and Paul Buttemer, traveled to a track in Blainville, Québec, Canada, to attempt to set new records in various categories as recognized by the IHPVA. (For those interested in Team Varna's results, and a description of the venue, see http://www.ihpva.org/com/Varna.) The biggest consideration in choosing our dates was the weather. Historically, the Blainville area enjoys, in late July, weather that is most conducive to cycling—warm, low-wind conditions, and humidity that is lower than at other times in the summer. However, in this particular July, the Blainville area suffered from unusually high and consistent wind conditions (El Niño after-effect?), with only occasional windows that were within

IHPVA wind requirements for records. We have experienced the same type of wind problems in other events, such as the Colorado Speed Challenge in September 1993.

On arriving in Blainville I was dismayed to find out that the IHPVA requires wind conditions of less that 1.67 m/s for all records, including long time trials. I have participated in a number of long events on closed-loop courses (1-hour, 12-hour and 24-hour TTs) where either new records were set, or it was advertised that a new record would be recognized, but I was never aware that any wind-measurement data were being collected. My assumption has always been that the wind rules apply only to the sprint events.

There has been much debate recently about new altitude rules, a major point being that those without reasonable access to high-altitude courses suffer a significant disadvantage. After our experience in Blainville, I believe the same can be said for the current wind rules. Personally, I would like to see some revisions made to the wind rules, or at least spark some discussion about this. As a starting point, I will first make some suggestions, then defend them.

1. We should increase the maximum allowable wind to at least 5 m/s (18 km/h) for the sprint events.

2. We should increase that maximum allowable wind to about 8 m/s (30 km/h) for the shorter time trials (such as 4 km and 10 km) on closed courses.

3. For the sprint events, we should also take the direction of the wind into consideration and allow records to be set when the wind exceeds the stated limit, but when within a certain angular offset of a direct headwind (say 30 degrees). This would require more sophisticated equipment than we currently use, and more diligent wind measurement than we currently make.

4. The wind rules should be abandoned completely for the longer time-trial events (one hour, 100 km, 12 hour, 1000 km and 24 hour), except to say that a record attempt could be halted by an official observer if in his or her opinion, the wind conditions present an unacceptable risk to the safety of the rider(s).

Many times I have seen published in IHPVA journals articles that conclude that our maximum allowable slope, 2/3 percent, gives up to ten times the advantage of our maximum allowable wind assist. Our practical experience corroborates a figure of this order. In Blainville, the back straight-away of the track slopes downward at a 0.08 percent grade, that is, in 1650 meters it drops about 1.3 meters. This amount of slope is not even discernible visually. But, when riding "down" this slope into a 8 m/s (30 km/h) headwind we would go more than 2 km/h faster than riding "up" the opposite straight-away with an 8 m/s tailwind! There has been discussion about making the slope requirements more stringent, but I believe that this would eliminate too many good record courses. I think that the 10:1 inequity between the advantage given by slope and that given by wind should be brought to within at least a 2:1 ratio.

Certainly, a headwind can do nothing but slow down even the most streamlined HPV. However, the "sail effect" may contribute, theoretically, to forward propulsion. But how many have had the experience of riding a streamliner at 80 km/h in a crosswind? I can tell you, beyond the shadow of a doubt, that any "sail effect" is more than offset by handling considerations. Even in a steady crosswind, one must lean the vehicle into the wind, counter steer and concentrate to remain rubber side down. The front wheel scrubs; it feels the same as going around a fairly sharp corner at speed, and you always go slower. A gusty crosswind (the most common type) is much worse. I do admit that the "sail effect" is a real phenomenon, and that at speeds lower than needed to break the 24-hour record, some forward propulsion can be obtained. But I say that if anyone is brilliant enough to build a vehicle that can actually realize a speed gain from the "sail effect" when traveling at 80 km/h, then s/he deserves whatever extra speed s/he can get.

The effect of nullifying a record due to wind in the longer time-trial categories (one hour and up) can be devastating to a rider and his or her team. In Blainville, I broke the 100-km record several times, but on more than one occasion, the wind reading ended up being over the legal limit, so these attempts could not be recognized. After such an attempt, I need about five days to properly recover, but who knows if there would be another possible window in five days? Time and money run out. I ended up re-attempting before I was properly recovered, and I feel that my performance suffered to some degree as a consequence.

Now, take this to the extreme, and imagine a cyclist being on schedule to break the 24-hour record, only to have the wind ruin it in the last hour. It could be many months, or even a year, before that athlete might be ready to duplicate the effort. A year's worth of training and planning, not to mention a sizable amount of money, would be wasted. So far, I have addressed only the physical side of things. It takes a tremendous amount of mental and emotional energy to prepare for a single record attempt and, to find out at the end of an otherwise successful one that the wind has ruined it, is a large psychological blow. When attempting to break the longer time-trial records, it would certainly be desirable to have wind conditions that conform to our current standards, and I contend that on a closed-loop course any wind will always have a net slowing effect.

—Paul Buttemer, pbr@mars.ark.com

(Editor's note: Paul Buttemer set 10-km and 100-km low-altitude records, still to be ratified, in July 1998, riding a Varna Orpheus built by George Georgiev, at an automobile test track near Montréal, Québec. The 10-km time was 7 mins. 53.02 s; and the 100-km time 79 mins. 4.74 s. at almost the same average speed: 21.1 m/s, 75.9 km/h, 47.1 mile/h.)

REVIEW

CONTINUATION: REVIEW OF THE EIGHTH CYCLE-HISTORY CONFERENCE

Hans-Erhard Lessing presented "The evidence against "Leonardo's bicycle", a supposed discovery of a sketch of an amazingly modern bicycle drawn by a pupil of Leonardo Da Vinci and attributed to the master's inspiration. This has been reported widely in the popular media and even in the learned journals as if it were fact. Lessing shows that it is a crude forgery. He puts the blame on a desire on the part of some nationalists to stake a claim for a fundamental invention for their country (jingoism) (as if Leonardo had not established enough "firsts"!). Others have criticized Lessing for this view. What amazed me was the casual way in which the restoration of the relevant part of Da Vinci's work, the *Codex Atlanticus*, seems to have been organized: there were many opportunities for people to meddle with his priceless work. It makes exciting reading.

In "As if on horseback", Roger Street discusses Denis Johnson's 1818 patent in

EDITORIALS

GUNTER ROCHELT

Gunter Rochelt, well known to HPV enthusiasts for his remarkable prize-wining aircraft Solair I, Musculair I and II, Schneidair and most recently the Solair II, died on the 27th of September 1998 after a short illness, at the age of 58.

Rochelt, born on the 23rd of September 1939 in Kamnitz (Bohemia), was professor of design at the Hamburg University of Fine Arts. He had gained a world-wide reputation as a leading expert in the field of lightweight construction. The Oscar-Ursinus Society gave him multiple awards for pioneering designs and Rochelt twice won the Kremer-prize for muscle-powered aircraft. Gunter Rochelt's Musculair I and Solair I and II and Schneidair have earned themselves places as permanent exhibits at the German Museum in Munich. His first Kremer prize was won with an aircraft that had no energy storage, although energy storage was permitted by the rules and was used in competing machines, and in addition it had no bracing wires. It was the first human-powered aircraft to carry a passenger (Gunter's son took his kid sister along for a flight).

(With thanks to Ernst Schoberl and Aerokultur for assistance.)

HUMAN POWER NUMBERING AND INDEXING

The only index we've had for *Human Power* was done in 1994, and it was rather crude ("A poor thing, but mine own"—Shakespeare). I found that the earlier numbering system was extremely haphazard. Volume 1 had six numbered issues, but then someone found another issue that came out between vol. 1 no. 3 and vol. 1 no. 4: I called it "3x". Volume 2 had only one issue, for spring 1982. I took over as editor in spring 1984, and resolved to have one volume a year with four issues per volume starting with volume 3. Something went wrong with volume 4, however, which had only one issue, strangely number 4. We didn't manage to keep to four issues per year because people were simply not delivering contributions.

To keep up the momentum we named the two "source guides" produced in early 1988 and late 1990 as issues of *Human Power*. We kept to the scheme of having four issues per volume, although we had a double issue for volume 9, 3 + 4. We kept thinking about the need for a new index, and worrying a little about continuing the earlier four-number sequence of volume/issue/year/page, given the confusion about earlier issues. We also want to bind *Human Power* into sets of around a hundred pages each, and we needed to arrive at a rational method of dividing up the past issues.

Recently a volunteer, Cyril Rokui, has offered to organize a new index. We were grateful, and we scrambled to try to put our house in order. Jean Seay made the eminently sensible suggestion that we scrap the volume system and just use issue numbers. We will adopt this system. Also, because *Human Power* is for information of long-term interest, we will exclude the source guides, which were very valuable before we could go to publications like *Recumbent Cyclist News* and to the internet, but would be out of place in bindings of a technical journal. This issue becomes number 46 (see the table, next column) that shows the old and new designations of past issues. Thank you, Cyril and thank you, Jean!

—Dave Wilson

THE RENUMBERING OF HUMAN POWER

Vol	#	Season	Date(s)	New #
1	1	Winter	1977–78	1
1	2	Winter	1979 (actually '78)	2
1	3	Summer	1979	3
1	3x	Fall ?	1979	4
1	4	Fall ?	1979	5
1	5	Winter	1981 (actually '80)	6
1	6	Fall	1981	7
2	1	Spring	1982	8
3	1	Spring	1984	9
3	2	Winter	1984	10
3	3	Spring	1985	11
3	4	Summer	1985	12
4	4	Fall	1985	13
5	1	Winter	1985	14
5	2	Summer	1986	15
5	3	Fall	1986	16
5	4	Winter	1986–7	17
6	1	Spring	1987	18
6	2	Summer	1987	19
6	3	Fall	1987	20
7	1	Spring/sum	1988	21
7	2	Fall/winter	1988	22
7	3	Spring	1989	23
7	4	Summer	1989	24
8	1	Summer	1990	25
8	2	Spring (!)	1990	26
8	4	Winter	1990–1	27
9	1	Spring	1991	28
9	2	Summer	1991	29
9	3/4	Fall & winter	1991–2	30
10	1	Spring/sum	1992	31
10	2	Fall/winter	1992–3	32
10	3	Spring/sum	1993	33
10	4	Fall	1993	34
11	1	Winter/spring	1994	35
11	2	Spring/sum	1994	36
11	3	Summer/fall	1994	37
11	4	Fall/winter	1994–5	38
12	1	Spring	1995	39
12	2	Fall	1995	40
12	3	Winter/spring	1996	41
12	4	Spring	1997	42
13	1	Fall	1997	43
13	2	Spring	1998	44
13	3	Summer/fall	1998	45

compiled by Dave Wilson, 8 Jan 1999

London of a velocipede that has always seemed suspiciously similar to Karl von Drais' machine, patented in Paris a year earlier. It therefore gives me a little ethical relief to read that Johnson's patent application acknowledges that he is taking action "in consequence of a communication made to him by a certain Foreigner resident abroad." This interesting short paper is a preview of a book since finished by Street: "The pedestrian hobby-horse: Britain's first bicycle", which I am looking forward to buying and reading and perhaps reviewing here.

There are 12 other enjoyable chapters—not reviewed here because we are short on space. Here are the titles and authors.

The Scottish school of cycle design, by Loren Hufstetler; The cycling new woman, by Bronwen Edwards; Some steps in the long march of the "Bloomer Brigade", by Les Bowerman; Shimano AX and the aero tradition, by Alan B. Bernstein; Huret: a notable French derailleur, by Raymond Henry; Out for a spin: the flaneur on wheels, by Glen Norcliffe; Who put the working man on a bicycle? by John Pinkerton; The social impact of cycling as a technology-based sport, by Ross D. Petty; Some facts about the history of doping in cycling competition, by Rüdiger Rabenstein; The beginnings of trans-Atlantic bicycle racing, by Andrew Ritchie; Is it about a bicycle? by Valerie Hawkins; Piet Pelle op zijn gazelle, by Gertjan Moed.

The hardbound volume is published by Van der Plas Publications, San Francisco, and has the ISBN 1-58068-000-3.

—Dave Wilson

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