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Additional copies of Human Power may be purchased by members for $3.50 each, and by nonmembers for $5.00.

We are indebted to the authors and Marti Daily whose dedicated help made this issue possible. Pat Poole

Letters to the editor

New membership

I am Wilton Shiraki of Hilo, Hawaii. I have recently been granted a U.S. patents for a Propeller Driven Surfboard, and I would like to gain membership into your association to benefit from the exchange of information that members enjoy.

I am particularly interested in human powered watercraft. I was told that the winter 1990-91 edition of your Technical Journal would be helpful. Is it possible for me to get a copy of that edition sent to me?

Enclosed please find a check in the amount of $25.00 for annual membership fees.

Also enclosed is a copy of my patent and pictures of my invention in use.

Thank you, very much.

Wilton Shiraki, 115 Maile Lau Li'i Place
Hilo, Hawaii 96720, Ph. (808) 961-6190

Human Power

Some points to ponder:

1. People in motorized wheel chairs, of whom there are more and more, are our natural allies. Sidewalk ramps at intersections are being made specifically for them, but they often use the street, like us. They can be thought of (their vehicles, that is) as *human power* vehicles – i.e., of approx. the same power that one human can supply. Their problems of access to way are the same as ours.

2. Access to reasonably safe ways is the main barrier to human-power travel, not weather or low speed. An unfaired upright bicycle is already hugely more efficient than foot travel for transportation of persons and goods.

3. The smooth power delivery and lack of possibility of standing on the pedals of recumbent bicycles means that they could use lighter gauge drive components than those based on the standard 1/2" pitch bicycle chain. Is a new standard needed?

4. A transportation system based on roads traversed by trucks, buses and automobiles is not a transportation *system.* A transportation system uses pipelines for fluids, and *guideways* for solids, which are the closest equivalent *to* pipelines.

5. The reason automobile dependence is no good is not environmental, but economic. In fact, there are five or more reasons why auto dependence is bad. In roughly decreasing order of importance they are:

A. *Economic.* On top of the $3,000 - and up - annual direct cost to the owner of depending on a car, the public (through government) supports (through taxes and other levies) the heavy costs of roads (land rent, construction, operation, maintenance) and higher medical costs, which hit two ways: i. Patching up accident victims. II. Shoring up people suffering from degenerative diseases because they do not walk.

B. *Safety.* Huge juggernauts piloted by persons of all political stripes endanger and drive off public ways legitimate users of way . . . pedestrians and users of vehicles of 1 human power.

C. *Health.* (See A. above) Walking is the natural human exercise base; when it is removed from common experience, degenerative ailments gradually develop (obesity, high blood pressure, sugar problems, clogged blood vessels, etc., etc.)

D. *Mobility.* Roads are prone to jamming with traffic, so that travel times are unpredictable. Problem gets worse with vehicle size and traffic density. Pedestrian, HPV, motorcycle travel times are more predictable.

E. *Environmental.* This is well-known; over-emphasized compared to A-D.

6. Hilly terrain and freedom to change level of effort on a recumbent bike require a 10:1 gear ratio spread (E.G. 12-120 gear-inches).

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(Continued on p. 9)

Acting Editor:
While Dave Wilson was off on sabbatical I volunteered to help out and act as Editor for this issue and the one due out in the Fall. I hope to rise to Dave's standards.

Pat Poole

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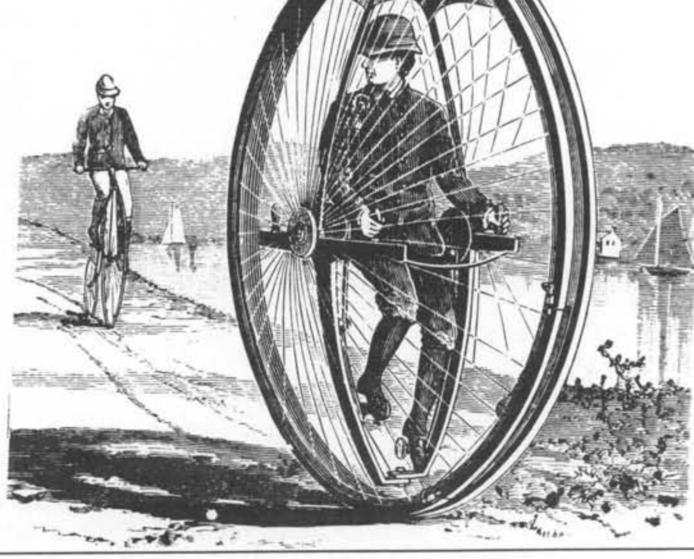

DESIGNING EXOTIC BICYCLE WHEELS FOR SUPERIOR STRENGTH

By Allan Klumpp

Summary

Rear bicycle wheels should have more spokes on the right than on the left. Asymmetric hub-flange geometry, which results from placing the sprocket cluster on the right, dictates the optimum distribution. For a typical composite hub, the optimum is two spokes on the right for each spoke on the left. Although wheels designed according to these principles were used in the 1984 Olympics, and other designs have been manufactured commercially, they are not described in Reference 1. A new aid for visualizing spoking designs is given. The reader can cut out the aid and use it to design a wheel that is optimum for his/her geometry. Six designs are given; others could be found using the aid.

Introduction

This report provides design techniques and designs for rear bicycle wheels. The designs are optimized for the hub-flange geometry and the application (racing, touring, etc.). The designs balance the spoke loads even though the wheels are asymmetric. A rear bicycle wheel normally is asymmetric because the sprocket cluster occupies space on the right-hand side. With conventional designs, the asymmetry loads right-hand spokes more than left-hand spokes, resulting in excessive spoke breakage and other problems.

Customarily, rear-wheel dropouts are symmetric with respect to the plane of the frame and the wheel is "dished" in order to make the rim plane and the bicycle symmetry plane coincide, as shown in Figure 1. Wheel dish is due to asymmetric placement of the hub's two flanges, through which the spokes pass. The flanges are placed asymmetrically in order to make room for the sprocket cluster. Wheel dish is customary for bicycles with five- to eight-speed sprocket clusters.

With standard six- and seven-speed clusters, the right-hand flange typically is only about half as far from the plane of the rim as the left-hand flange. The result is that right-hand spokes carry about twice the load of left-hand spokes. This is due to necessarily tighter stringing on the right, and also due to distributing more of the weight on the right. Excessive spoke breakage on the right side of rear wheels is due largely to this unbalance, and to the fact that right spokes apply virtually all of the pedaling torque to the rim (typically 88% according to a calculation appearing on page 136 of Reference 1). The unbalance exists in all popular bicycles sold today.

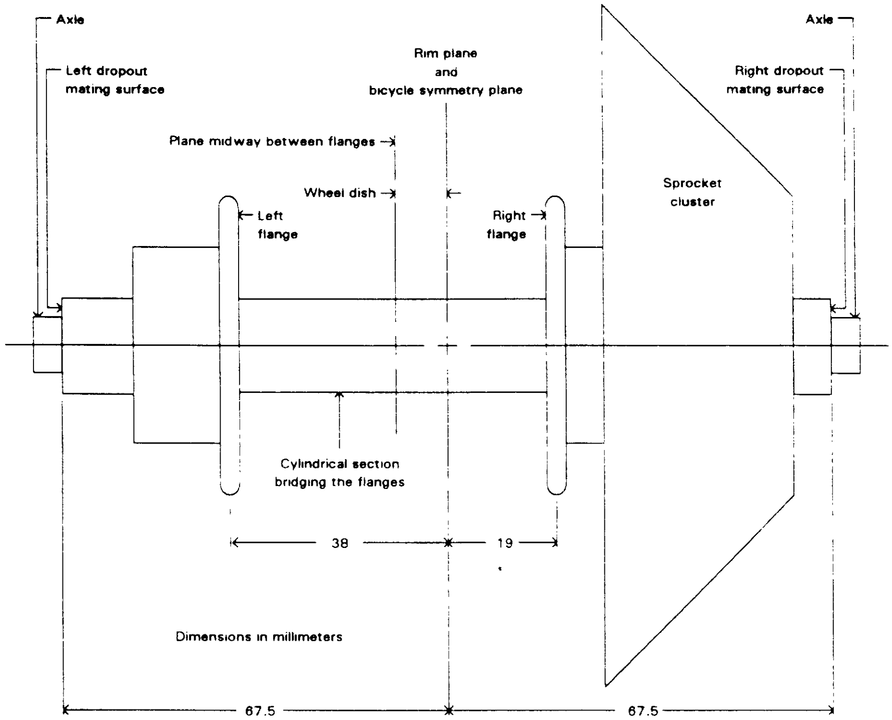

Figure 1. A Composite Cassette Hub As Viewed from the Rear of the Bicycle

Unbalanced spoke loading is unnecessary. Spoke loads can be balanced by distributing spokes *unequally* between flanges. Balanced spoke loading *requires* unequal spoke distribution. Not only is this possible theoretically, it is eminently practical. With conventional five- to eight-speed hubs and 36-hole rims, spoke loads are balanced almost perfectly when there are 24 spokes on the right and 12 on the left. The same is true for 24-spoke wheels with 16 on the right and 8 on the left. Spokes are located at uniform intervals around rim. All spokes are the same length, except that wheels with radial spoking on the left flange have shorter left spokes.

This report describes how to design such wheels, and provides several sample designs, both for competition and for carrying heavy loads. Some designs are ideally suited to the asymmetry of standard hubs; other designs are for hubs that are either more or less asymmetric than standard hubs. Anyone who can use an ordinary hand drill and can build a conventional wheel can build these high-strength wheels.

Wheels with unequal spoke distributions already are being built for demanding riders. Reference 2 reported that some U.S. Olympic road-racing candidates were using 24-spoke rear wheels with 16 spokes on the right and eight on the left. Hi-E Engineering, Inc., Nashville, Tennessee manufactures 40-spoke rear wheels with a 24/16 distribution and 24-spoke wheels with a 16/8 distribution. In 1990, Dave's Wheels, an adjunct to Campus Bike Shop in Buffalo, New York, was planning national marketing of the "Revolution" line of high-performance wheels with 16/8 and other unequal distributions, including a wheel similar to the French Roval. Trek manufactures the "Matrix" line of rims, competition and touring, with spoke holes ranging from 24 to 48. Matrix rims are available only from dealers (who should have Trek's catalog); Trek does not sell at retail.

The advantages of balanced spoke loading, cited below, are based partly on experience and partly on the known relationships between stress load, fatigue failure, and metal creep. Advantages are:

- Wheels with balanced spoke loading are far easier to align. Aligning a 24/12 rear wheel is as easy as aligning an 18/18 front wheel.

- Wheels with balanced spoke loading hold alignment far better. Rear and front wheels hold alignment equally well.

- Spoke breakage is reduced. Reduced breakage is due to a substantial reduction in the loading of the right-hand spokes. The reduction comes from making all spokes carry an equal share of the load. Balanced spoke loading is optimum: it minimizes breakage.

- With balanced spoke loading, a bicycle does not whip from side to side due to bumps or holes. In principle, conventional spoking causes side-to-side whip. Whether a rider can feel this whip is debatable. The magnitude is about half what it would be if a right spoke were broken. Some riders can detect broken or loose spokes from the side-to-side whip. A bicycle with balanced spoke loading seems to be more solid and stable, particularly when cornering on rough roads. But this sensation is subjective; it has not been substantiated statistically.

Balanced spoke loading seems to have no important disadvantage. Even though the rim supports more spokes from the right side than from the left, it is sufficiently stiff to prevent any detectable local waviness. With most rims, the left or right offset of spoke holes is so slight (a millimeter or so) that any hole can be used for any spoke without measurable adverse effect. When balanced spoke loading becomes common, spoke-hole offsets in rims should be eliminated; Hi-E markets such rims already.

For heavy-laden touring, some cyclists advocate 40-spoke wheels with a 20/20 distribution. However, the 36-spoke 24/12 distribution offers three times the improvement over a conventional 36-spoke wheel in terms of the number of additional spokes supporting the load on the right flange. That is, the 24/12 distribution increases the right-side spokes by six (from 18 to 24) whereas the 20/20 distribution increases the right-side spokes by only two. Half the additional spokes of the 20/20 distribution are wasted on the left flange, where the spokes carry less load.

Many cyclists have asked why it is that not all wheels are built with balanced spoke loading. There seems to be no reason other than habit.

Spoke Distributions

This section discusses the distribution of spokes between left and right flanges.

Spoke distribution should be chosen to balance left and right per-spoke loads. Based on a composite of several hubs (including the Shimano Hyperglide seven-speed road and mountain cassette hubs and the Maillard Helicomatic hub), left and right flanges are separated by about 57 mm with the plane of the rim about 38 mm from the left flange and 19 mm from the right. These dimensions are in a two-to-one ratio (38/19). Therefore spoke loads are balanced if there are twice as many spokes from the right flange as from the left. With a 36-hole rim, a 24/12 spoke distribution balances spoke loads, for this wheel geometry.

If the wheel is to be more dished, e.g., the rim plane will be three times as far from the left flange as from the right, then spoking will be balanced using a 30/10 distribution with a 40-hole rim or a 24/8 distribution with a 32-hole rim. No satisfactory 27/9 spoking geometry has been found. If the wheel is to be less dished, e.g., the rim plane will be half again as far from the left flange as from the right, then spoking will be balanced using a 24/16 distribution with a 40-hole rim.

> *Spoke loads are balanced if the number of spokes on each flange is inversely proportional to the distance from the rim plane to the flange.*

Balanced spoke loading balances the average spoke tension in an unloaded wheel. It also balances the per-spoke loading due to the weight of the rider plus the bicycle and any cargo. Balanced spoke loading reduces but does not balance the per-spoke loading due to pedaling torque, as will be shown in the following section.

Design Techniques

This section provides techniques for designing spoking geometries that are optimized for the hub geometry and the application. The techniques are based on visualizing the effects of wrapping the spokes around the axis of the hub. A visualization aid, consisting of a pair of cards marked to represent spoke-head locations on the left and/or right flange, is used to find satisfactory designs. A design is satisfactory when spokes do not interfere with one another. Spokes are said to interfere when the heads of two spokes must occupy the same hole in the flange, or when the shaft of one spoke must penetrate the head of another spoke (see Figure 2).

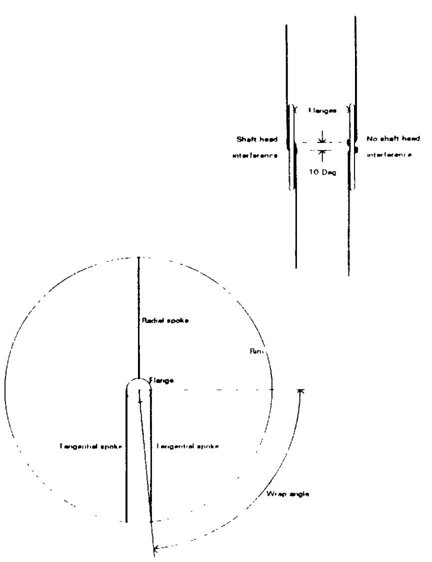

Figure 2. Spoke Geometries

As shown in Figure 1, the hub provides two flanges, one on the left and one on the right. In order to drive the bicycle forward, one or both flanges must be connected to the rim by spokes that wrap at least part way around the axis of the hub. That is, the spokes cannot be radial because radial spokes cannot transmit pedaling torque from the hub to the rim. On the other hand, when the bicycle is at rest, zero net torque must be transmitted to the rim. Since spokes are in tension even when the bicycle is at rest, some spokes must transmit torque clockwise while others transmit torque counterclockwise, thereby making the net torque zero.

A *driving* flange is one whose spokes are not radial. Half the spokes from a driving flange transmit torque clockwise, and the other half counterclockwise. When torque is applied to the pedals, the tension in half the spokes increases and the tension in the other half decreases. The differential tension applies torque to rim, driving the bicycle forward.

The left flange need not be a driving flange because the cylindrical section bridging the flanges (see Figure 1) is generally so flexible that it virtually isolates the left flange from the sprocket cluster. Thus, even when the left flange is a driving flange, it generally contributes an undetectable fraction of the driving torque. Balanced spoke loading reduces the per-spoke loading due to pedaling torque by dividing the load among a larger number of spokes; most of the spokes are on the right flange.

Torque contributed by the right flange can be detected by the change in pitch of spokes when plucked while torque either is or is not being applied to the pedals. With common hubs, the change in pitch of spokes from the left flange is undetectable.

Spokes whose tension increases due to pedaling torque are called *trailing* spokes, and those whose tension decreases are called *leading* spokes. To avoid interferences between the shafts of trailing and leading spokes, they are located on opposite sides of the flange.

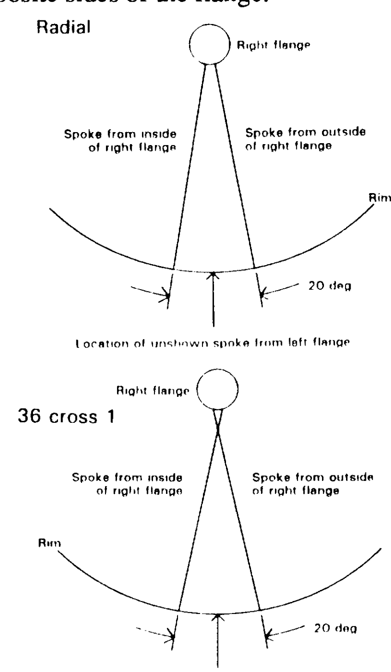

Figure 3. Radial and 36 Cross 1 Wheels Viewed from the Right Side

There is no universally accepted convention for placement of trailing and leading spokes. I place trailing spokes on the outside of the flange on the hypothesis that trailing spokes are more likely to break, and flange contact reduces flexing and consequently fatiguing of outside spokes. Inside spokes do not contact the flange.

To visualize the effects of wrapping spokes around the axis of the hub, begin with a radial geometry as seen from the right side (see the left drawing of Figure 3). Using a conventional 36-spoke wheel as a model, spokes are located every 10^{o} around the rim, spoke heads are located every 20^{o} around each flange, and, on a given flange, heads alternate between the inside and outside of the flange. Interchange the heads of adjacent spokes, so that the heads of spokes on the outside of a flange move clockwise 20^{o} and the heads of those on the inside of a flange move counterclockwise 20^{o} (see the right drawing of Figure 3). The result is a 20^{o} wrap angle (the angle subtended by the head and nipple of a spoke, as viewed from the axle, see Figure 2). Each trailing spoke (on the outside of the flange) crosses one leading spoke (on the inside of the same flange). Thus a 20^{o} wrap angle corresponds to a "36 cross 1" spoking geometry. Repeating the processes produces successively a 40^{o} wrap and a 36 cross 2 geometry, a 60^{o} wrap and a 36 cross 3 geometry, and finally an 80^{o} wrap and a 36 cross 4 geometry.

Conventional spoking geometries are named according to the number of spokes total and the number of spokes from the same flange that each spoke crosses. The 36 cross 4 geometry is common. This naming convention is inadequate and misleading for the wheels described here. These wheels are described in terms of spoke distribution and wrap angle. For example, Figure 6 describes a 36-spoke wheel with a 24/12 distribution and an 80 degree wrap angle. Each spoke on the right flange of this wheel crosses five spokes, not four as we saw in the preceding paragraph for conventional wheels with an 80^{o} wrap angle.

As an aid for visualizing the effects of wrapping the spokes about the axis of the hub, you can make a pair of circular concentric cards as shown in Figure 4. The cards represent the left and right flanges as viewed from the right side of the bicycle. Tic marks on the outer card represent possible locations of *trailing* spoke heads, while tic marks on the inner card represent possible locations of *leading* spoke heads. An L or an R next to a tic mark denotes the presence of a spoke head on the left or right flange. When the cards are in the indexed position, as in the left drawing, all spokes are radial. Rotating the outer card clockwise represents moving the heads of *trailing* spokes clockwise around the flange to the offset position shown on the right. Rotating the inner card counterclockwise represents moving the heads of *leading* spokes counterclockwise around the flange. In the offset position, both an L and an R may appear next to facing tic marks, or both tic marks may be vacant. More is said on this in the numbered notes below.

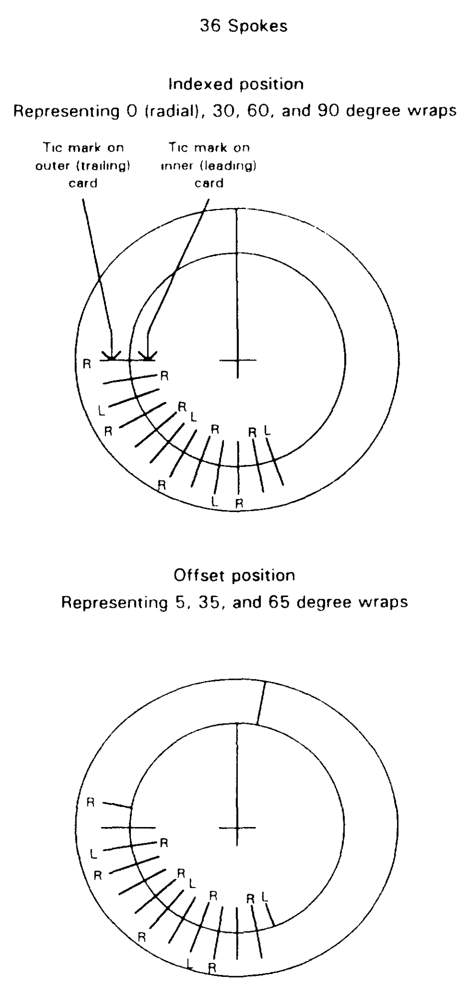

Figure 4. A Circular Visualization Aid

The problem with the circular aid is that it is hard to make. An equivalent aid that is easier to make is a pair of rectangular cards (see Figure 5). These cards can be visualized as resulting from unwrapping the circular cards. The left card corresponds to the outer card of the circular visualization aid. Thus the left card represents the trailing spoke heads. Raising the left card represents moving the heads of *trailing* spokes clockwise around the flange. Lowering the right card represents moving the heads of *leading* spokes counterclockwise around the flange.

The design process is to assign each spoke as trailing or leading, right or left flange, marking the cards accordingly while the cards are in the indexed (radial) position, and then to test the effects of all of your assignments by moving the cards to various offset positions. If left and right wrap angles are different, then left and right flanges are tested independently. If your assignments don't work, start over.

Table 1
Data on Spoking-Design Drawings

Figure number	6	7	8	9	10	11
Spokes total	36	36	36	40	32	24
Spokes	24/12	24/12	24/12	24/16	24/8	16/8
New flange holes	12/0	12/0	12/6	8/8	12/4	8/0
Unused flange holes	6/6	6/6	6/12	4/12	4/12	4/4
Wrap angle	80/0-10	65	60	76.5	50.625	75/0-15

NOTES:
1. Right-flange value/left flange value (if left value different).

2. In figures 6 and 11, left-flange spoking is quase-radial, meaning that instead of drilling new holes, spoke heads are located in the nearest existing holes, which are selected so that the nonradial spokes apply no net torque to the flange. Thus in Figure 6, six spokes are radial and six have a 10° wrap; in Figure 11, four spokes are radial and four have a 15° wrap.

Here are notes and rules for making a visualization aid and using it to design wheels. Most refer to the straight aid.

1. Spoke geometry is invariably cyclic, with several cycles per revolution of the wheel. For example, a 24- or 36-spoke wheel is likely to have six spokes per cycle and a 40-spoke wheel may have 10 spokes per cycle.

2. For the straight aid, the number of tic marks per card should be twice the number of spokes in a cycle, e.g., 12 tic marks per card for the 24- and 36-spoke sample wheels, 20 tic marks per card for the 40-spoke samples.

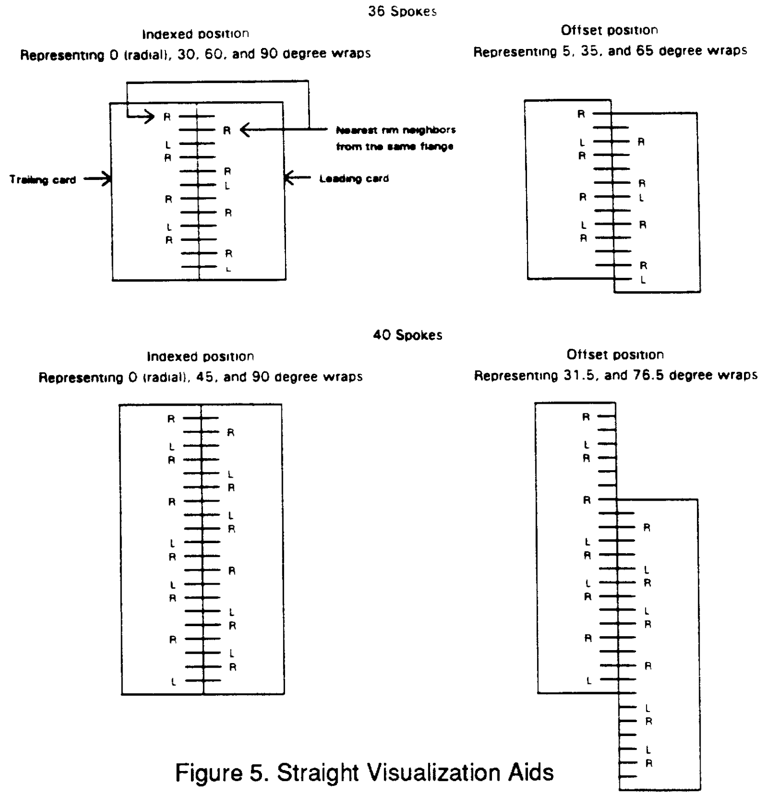

Figure 5. Straight Visualization Aids

3. When the cards are in the indexed (radial) position, mark every flange location for one and only one spoke head. Thus mark either the trailing or the leading card with either an L (left flange) or an R (right flange).

4. Half the spokes from each flange must be trailing and half leading.

5. You can adjust the offset between cards to represent any value of wrap angle (assuming flange holes could be located arbitrarily).

6. If wrap angle is a multiple of the spacing between rim holes, then flange and rim holes are aligned. With conventional wheels, this is always the case. If wrap angle is a multiple of half the spacing between rim holes, then flange holes are midway between rim holes. This is the case for the wheels of Figures 7, 9, and 10.

7. Spokes from the same flange that occupy adjacent holes on the rim are called nearest rim neighbors (see Figure 5).

8. As the left card is raised with respect to the right card, the heads of some nearest rim neighbors move together so that the spokes cross; the heads of other nearest rim neighbors move apart so that the spokes separate. In the upper example of Figure 5, all nearest rim neighbors separate. In the lower example, some cross and some separate.

9. Because of note 8, raising a card differs fundamentally from lowering the same card. Even though the geometries are cyclical, raising a card by a given number of tic-mark spaces is not equivalent to lowering the same card by the complement of that number of spaces.

10. An offset between left and right cards of one space represents a wrap angle of half the spoke interval, i.e., 7.5^{o} for 24-spoke wheels, 5.625^{o} for 32-spoke wheels, 5^{o} for 36-spoke wheels, and 4.5^{o} for 40-spoke wheels. A one-space offset represents other wrap angles also, as explained below.

11. As the left card is raised with respect to the right card, the geometry of spoke heads represented by the two cards repeats because of the cyclic spoke geometry. In terms of wrap angle, the period is half that of the spokes. Thus for 24-spoke wheels with 6 spokes per cycle, 32-spoke wheels with 8 spokes per cycle, or 40-spoke wheels with 10 spokes per cycle, the head geometry repeats every 45^{o} (the wrap-angle period). For 36-spoke wheels with 6 spokes per cycle, the wrap-angle period is 30^{o}.

12. The offset between left and right cards represents the wrap angle modulo the wrap-angle period. For example, for 36-spoke wheels with a 30^{o} wrap-angle period, an offset of one space represents wrap angles of 5^{o}, 35^{o}, and 65^{o} (see the upper-right drawing of Figure 5). For 40-spoke wheels with a 45^{o} wrap-angle period, an offset of seven spaces represents wrap angles of 31.5^{o} and 76.5^{o} (see the lower-right drawing of Figure 5).

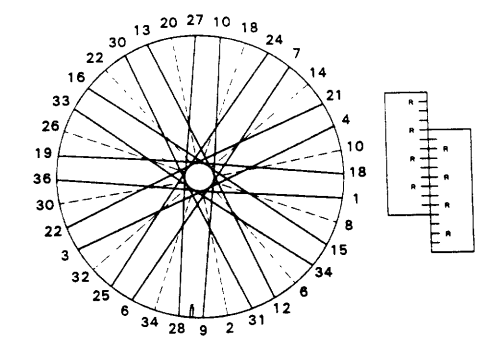

Figure 6. A 36-Spoke Wheel with 24/12 Distribution and 80° Wrap on Right

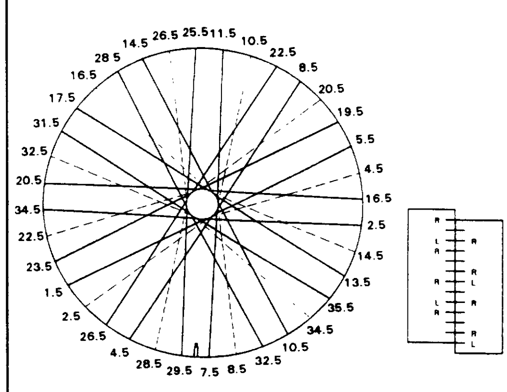

Figure 7. A 36-Spoke Wheel with 24/12 Distribution and 65° Wrap

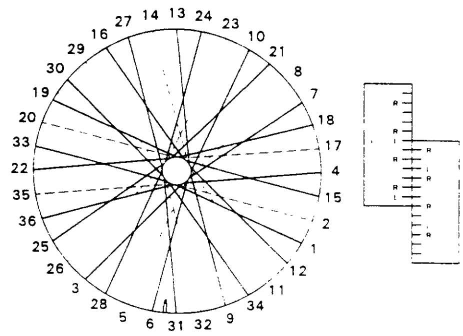

Figure 8. A 36-Spoke Wheel with 24/12 Distribution and 60° Wrap

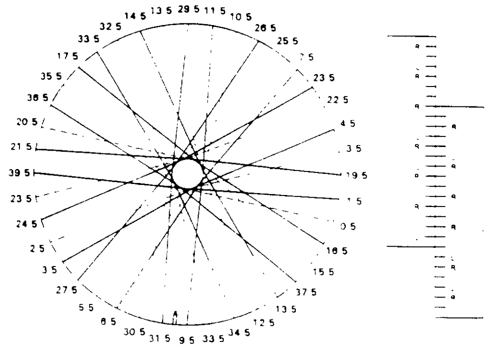

Figure 9. A 40-Spoke Wheel with 24/16 Distribution and76.5° Wrap

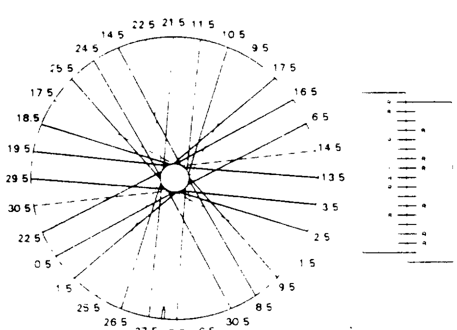

Figure 10. A 32-Spoke Wheel with 24/8 Distribution and 50.625° Wrap

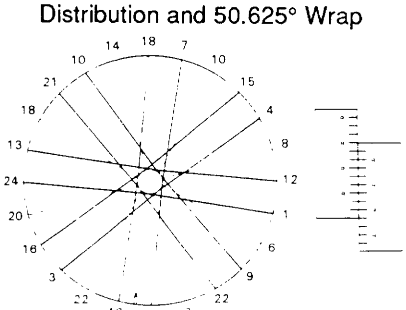

Figure11. A 24-Spoke Wheel with 16/8 Distribution and75° Wrap on Right

13. It is permissible for neither an L nor an R to appear next to facing tic marks. At this location, neither flange is occupied by a spoke head.

14. It is permissible for both an L and an R to appear next to facing tic marks. At this location, both flanges are occupied by a spoke head.

15. It is <u>not</u> permissible for an R (or an L) to appear next to both tic marks of a facing pair. This represents two spoke heads occupying a single flange hole, an impossibility.

16. It is permissible for a raised-card tic mark and the facing-card tic mark one space <u>above</u> it to represent spoke heads on the same flange. The spokes that the tic marks represent separate; therefore there is no shaft-head interference (see Figure 2). This occurs in both right-hand drawings of Figure 5.

17. It is <u>not</u> permissible for a raised-card tic mark and the facing-card tic mark one space <u>below</u> it to represent spoke heads on the same flange. The spokes that the tic marks represent cross; therefore there is shaft-head interference (see Figure 2), unless the wrap angle is small. With small-flange hubs (practically universal now), an 80^{o} wrap requires 20^{o} head separation to avoid shaft-head interference. With 10^{o} head separation, the wrap-angle upper limit is 50^{o}.

Spoking Designs

All spoking designs use standard rims (24, 32, 36, or 40 holes) with uniform spoke intervals. All use equal-length spokes on a given flange, but designs with quasi-radial left-flange spoking use shorter spokes on the left. No design requires drilling spoke holes in rims. All designs use standard hubs that come drilled to match the rim. In all designs, the spokes are distributed unequally between right and left flanges. Therefore, additional holes must be drilled in one or both flanges. New holes are always drilled midway between existing holes. This makes the minimum angular spacing between holes the same on each drilled flange as it is on the rim. Even though new holes must be drilled in a flange, some existing holes are not used because they are not in the right locations.

Every design has nonuniform spacing between right-flange spoke heads, even though all right-flange spokes in a given design have the same wrap angle. For example, every design with 36 spokes in a 24/12 distribution has right-flange spacing alternately 10^{o} and 20^{o}, giving two right spokes and one left every 30^{o}.

An 80^{o} wrap makes the spokes as nearly tangential to the flange as possible with a 36-spoke wheel. Tangential spoking minimizes spoke loading due to pedaling torque. Tangential spoking also eliminates sensitivity of spoke length to flange diameter, allowing use of the same length spokes in wheels of the same diameter, regardless of flange diameter. There is no advantage in going beyond 80^{o} wrap. Radial spoking minimizes weight, and allows all spokes to be on the outside of the flange. Radial spoking is appropriate for front wheels because only the torque needed to overcome bearing friction can be applied to a front hub, a minute amount. A common spoking geometry is 36 cross 3. Since this provides only a 60^{o} wrap, it is less than optimum in terms of spoke loading due to pedaling.

For carrying heavy loads, the best design is that of Figure 6, with a 24/12 distribution, near-tangential right, and quasi-radial left. For racing, the best is that of Figure 11, with a 16/8 distribution and similar geometry. A similar 48-spoke wheel could be built for heavy loads and tandem bicycles. These wheel designs best match the near 2/1 flange spacing of conventional hubs shown in Figure 1.

Notes on spoking-design drawings:

Locations of flange and rim holes are numbered in a counterclockwise direction with the number 1 rim hole immediately after the valve. Each number around the outside of a rim designates the location of the flange hole where the spoke head is anchored. When these numbers are integers, flange holes are aligned with rim holes; when they are decimal (e.g. 1.5), flange holes are midway between rim holes with the number 0.5 flange holealigned with the valve. Spokes from the left flange are shown dashed; spokes from the right flange are solid.

Acknowledgments

Many readers of previous versions, especially Gene Wester and Sugi Sorensen, contributed valuable improvements incorporated in the current version.

References

1. J. Brandt, *The Bicycle Wheel*, Avocet, Inc., Menlo Park, California, 1983.

2. Los Angeles Times sports section, 1984 June 27.

Allan R. Klumpp
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Allan Klumpp has been developing guidance, navigation, and control systems for missions to the moon and planets since 1959. Now with the Jet Propulsion Laboratory, he was formerly with the C.S. Draper Laboratory and, as the Draper laboratory was called then, the MIT instrumentation laboratory. He has bicycled for pleasure and as a commuter his entire career.

Letters to the Editor (continued)

Response to Tim Leier (vol. 10/2) by Peter Sharp:

When I read Tim Leier's letter in response to my article "It's Time To Change The Rules" (vol. 10/1), I realized that no one has published a paper on the use of pure human energy accumulators for land HPV. As a consequence, we lack common terms and concepts for discussing the issue. I therefore wrote the accompanying article, "A Controversial Issue: Human Energy Accumulators for Land HPV". I appreciate Tim's feedback, and I think that many members may share his opinions.

Tim agrees with me that airfoil fairings can function as "powered aerodynamic devices". His excellent article, "Aerodynamic Gains From Cross-Wind Conditions", demonstrates the large amount of power that a properly designed airfoil fairing can extract from a legal cross wind (6 kilometers per hour) in a top speed record run. Tim somehow misinterpreted me to be saying that I "condemn" fairings, so he argued in favor of them. On the contrary, I favor the use of fairings with no restrictions whatsoever on the use of wind power. My point was that since competition rule 3.1.1. unintentionally forbids airfoil fairings (by forbidding "powered aerodynamic devices"), that rule should be amended so that it does not prohibit airfoil fairings.

Tim also misinterpreted me to be saying that I wish to increase the inertia of HPV in order to make them better energy accumulators. On the contrary, I am advocating the use of accumulators for the purpose of overcoming the inertia of HPV.

Tim prefers to leave rule 3.1.2. as it is. That rule forbids all energy storage, with no exceptions — not even for the kinetic energy that any vehicle must store in order to move. Leaving that rule as it is strikes me as taking a considerable risk for no apparent reason. Anyone with an elementary knowledge of physics can see that the rule is flawed, and that no HPV can comply with it. Marti Daily, our esteemed President, has mentioned that the fragile structure of the IHPVA could be endangered by legal disputes. Anyone could claim the world's fastest HPV. If the IHPVA were to contest that claim, it would not have a leg to stand on since its records of at least the last 10 years have been in violation of its own rule. That would be a sorry day. I agree with Tim that we have to live with the fact of inertia, but we don't have to live with a rule that actually ignores inertia. It is a legal liability, contrary to basic physics, and a bit embarrassing as well, especially for an organization with so many engineers as members.

Tim argues first against more rules (to regulate accumulators) and then argues in favor of rules (to exclude accumulators). Besides the obvious contradiction, the number of rules is not the issue. The issue is that of eliminating a major and unnecessary restriction (it is the policy of the IHPVA to do so). If eliminating a major restriction requires only the addition of a couple of simple rules, then the tradeoff is an excellent one.

Tim wants to establish a separate competition for hybrid vehicles. Let me raise a tricky issue. He has shown that airfoil fairings produce considerable power in competition legal cross winds (6 kilometers per hour). That would make them hybrid vehicles. Would he place airfoil fairings in that separate hybrid competition? He would probably argue that such fairings produce no net thrust; they are only a way to reduce drag, and are therefore not hybrids. But that is like saying that a gas engine used only to overcome aerodynamic drag is not a hybrid device. Obviously, using non-human power — either wind power or a gas engine — to overcome drag creates a hybrid vehicle. We should acknowledge the obvious and then make a sensible decision about what to do about it, such as declaring wind power hybrids to be completely legal. In effect, the IHPVA does legitimize the use of hybrid wind power by not banning it. But for record attempts the IHPVA also sets a wind speed limit. That limit serves to minimize the amount of available hybrid wind power. That is a sensible compromise. However, practical HPV should make all possible use of wind power. Oddly enough, that means that someday we may have practical HPV which regularly make use of wind power to attain speeds well above the official top speed records of the IHPVA.

While I very much appreciate Tim's work on airfoil fairings, the super gas mileage contest seems to me an odd sort of contest because it requires its vehicles to be inefficient (no crabbing or wing sails) in order for them to claim to be highly efficient. That is also what the IHPVA does when it bans human energy accumulators from the top speed events. It is the definition of efficiency itself that needs to be reconsidered, and I attempt to do so in my accompanying article. I hope that my article will speak to Tim's concerns, and that he will reconsider his assumptions from the perspectives I present.

Editor's Note:

Peter's excellent paper is included in this issue beginning on page 19.

NFA Vehicles

by James Donohue

NFA VEHICLES is researching a practical HPV design based on a regular bicycle without any welding. It is a 12-speed bicycle with a streamlined fairing. It can be loaded with gear for touring and commuting. A roof was put on one of them for comfortable riding in rain and cold weather. The conclusions of this article are based on thousands of miles of testing.

NFA VEHICLES is entirely owned by JAMES DONOHUE. All of the vehicles are production bicycles with prototype fairings. To identify which one is being discussed they have been given numbers instead of names. If you read my article "How to Build an Aerodynamic Bicycle Fairing" in the Summer 1987 HUMAN POWER, the vehicles discussed in it have been designated "Type-1", "Type-2", and "Type-3". This new article deals with "Type-4", "Type-5", and "Type-6". Also mentioned is the new "Type 7A".

"Type-4" was a fiberglass versions of "Type-5". The "Type-4" was built of the less expensive material because it was built to be used in a barrier impact test to learn about the Crumple Zone. Lead ballast was used to simulate the mass of a rider and a running man pushed the vehicle at running speed into a concrete wall. A video camera was in position to record the impact. Playing the tape in slow motion revealed more than the eye could perceive. Material has been added to the crumple zones to absorb more impact based on the video tape.

Three "Type-5" units were built. One was modified to "Type-6". One is in storage and one is used every day with 10550 Km (6550 miles) on the odometer. The other vehicles had approximately 5000 kilometers (3000 miles) each.

From 1988 to 1990 testing was done to determine how much faster the vehicle would go compared to an unfaired bicycle. Coefficient of drag (Cd) was determined to be .56. The lights which have been added to the fairing disturb laminar flow. The current vehicle has lights, batteries, toolbox and a heavier fairing and as a result someone can probably go faster on a good unfaired racing bike; but the "Type-5" is good or excellent for touring and commuting.

The electrical system is 12 volt with 13 amp hours of storage. The system includes a 75 milliamp photovoltaic solar panel. The headlight is a 20 watt halogen floodlight. A special fixture was built so it could be mounted inside the fairing. The built is only 2 inches or 5 centimeters in diameter so cutting a hole in the fairing was not a problem. Two 3 watt tail lights also serve as side marker lights. There is a brake light and blinking directional lights.

For auditory awareness, the vehicle has a horn from an Oldsmobile which consumes 60 to 75 watts.

The bicycle used is out of production. It is a 12 speed bicycle with 507 mm (20 inch) wheels and a 100 cm (39 inch) wheelbase. Other frames with this size wheel have a shorter wheelbase for children. This one had a 100 cm wheelbase so an adult could ride it. If anyone has a Hutch GP-1 or "HPV Superbike" frame they want to sell, please contact the author.

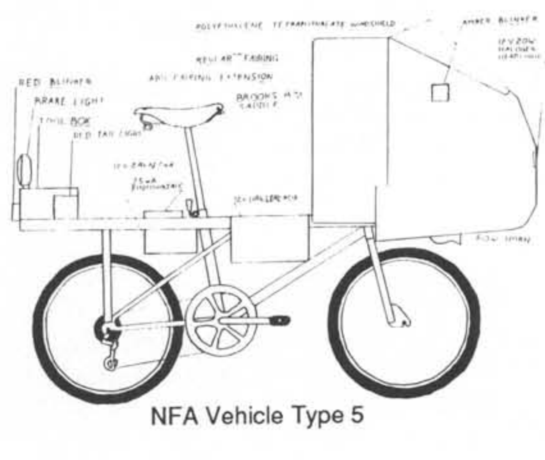
The vehicle has two aluminum tubes clamped to the top tube of the frame to support the fairing. These frame rails are clamped on with stainless steel hose clamps. The fairing is clamped to the tubular frame rails with U-bolts. Previous attempts of drilling holes in the tubes and bolting the fairing on resulted in cracks starting at the holes. Now the vehicle has gone 6500 Km without any metal fatigue. The "Type-5" is performing well. It could use a better paint job, but it works.

The "Type-6" is a "Type-5" with a roof. The roof is high enough for the rider to stand with feet on the ground. The roof is strong enough for someone to stand on top of the roof! It is made of four layers of fiberglass and two layers of Kevlar and also has two longitudinal aluminum tubes and a bulkhead. It served well for running errands in rain and freezing cold weather. It is currently in storage in New York until the author returns from military service in 1995.

"Type-7A" is a fairing which mounts on the handlebars of a mountain bike. The paint is better than earlier fairings because it is for sale. Being handlebar mounted like the "Type-3", experience has dictated that it be smaller and not enclose the front wheel because of crosswind instability with "Type-3". Aesthetics was considered more important than Aerodynamics for the "Type-7A". It was built to look nice in the photographs and carry a few things in its glove compartment, but there is no claim of increased speed. The clamps which mount the fairing to the handlebars are so heavy that a simple two point mounting system is possible rather than a complicated 5 point system like some fairings. The fairing is fiberglass/polyester. "Type 7-A" does not fit drop handlebars so "Type-8" is on the drawing board.

NFA VEHICLES prototypes have proven to be practical. "Type-5" is durable and does a lot of work carrying things. "Type-6" proved that an all weather bicycle can be built and used. "Type-7" is available.

All of the fairings have been built with the experience of building the previous fairings. The fairings are built to carry things for utility and also have good aerodynamics.



NFA Vehicle Type 5

Bendy Propeller Shafts for Human-Powered Boats

by
Shields M. Bishop

Small-diameter, high-strength shafts which are operated while bent to an arc shape are one way to position the propeller adequately deep and rotating with its axis parallel to the water flow under the boat. The bent shaft transmits power to the propeller more smoothly than Cardan-type universal joints, while providing a flexible connection for easily retracting the prop for beaching and car-topping.

In the case illustrated (Fig. 1), the shaft and the strut would be retracted up into a hollow skeg or an arrangement similar to a centerboard well in a sailboat. On a catamaran, the strut is supported by a bracket between the two hulls.

Bent shafts for power transmission have been used successfully in the past. One application was the drive shaft of the 1960-1962 Pontiac Tempest automobile. The bent shaft in that case transmitted the power from the forward engine to the rear tanaxle, which helped lower the drive shaft tunnel. The bearings at the ends of such shafts are arranged so as to impart a constant bending moment on the unsupported portion of the length so that the bent portion is a true circular arc. The shaft is a classical case of a beam loaded as shown in Fig. 2. where lengths ab=cd and the loads A=B=C=D. The beam then deflects as shown in Fig. 3.

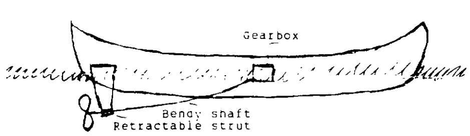


Figure 1

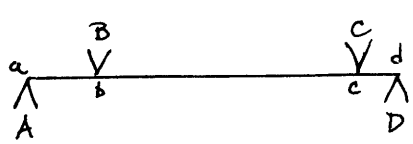


Figure 2

Figure 3

The stress calculations at the end of this article show that a good grade of spring-tempered steel will withstand the torsional and bending stresses. In this case, I rather arbitrarily selected a handy shaft diameter and useful bend radius that I have used on several pedal-powered boats, but people with a bent for computer programming might want to refine my efforts to provide the optimum combination of shaft diameter, bend radius and torsional loading as functions of the boat parameters and the fatigue -life data for the selected shaft material. When planning the layout of the components for retracting the prop, keep in mind that the number of stress cycles for this will be much smaller than the number of cycles during operation, and therefore much higher bend stresses and much smaller bend radii can be tolerated.

For my shafts I have used type 630 (17-4 PH) stainless steel in the H-1100 temper[3] because it is rust resistant and has very high fatigue strength.[2] A good grade of 1095 spring steel, through-hardened and tempered to about Rc 50 would be acceptable, but the cost of corrosion protection would probably nullify the material cost savings over stainless steel. Other materials such as high-strength composites might be considered to save weight and to reduce shaft diameter.

For attaching the small-diameter shaft to the gearbox on the boat and to the propeller, I use a long-sleeve epoxy joint. To do this, bore out a 13mm (.512 in.) diameter 6061-T651 aluminum alloy rod to 6.5mm (.256 in.) ID. Allow at least 10X shaft diameter for the bored hole depth, (approximately 65mm). Clean the surfaces to be bonded very carefully and use a good grade of water-resistant epoxy adhesive. You now have a larger-diameter easily drilled end on your shaft which will take cotter pins, etc. This joint will not fail. The shaft fails in torsion at about 40 Nm (350 lbf.-in.) torque while the joint remains intact. The joint may be disassembled by heating it to about 200 C (400 F). This heating does not harm the metal components. Note that there is about 0.15mm (.005 in.) clearance between the OD of the shaft and the ID of the bore to accommodate a film of adhesive. Be sure to provide a small air hole at the end of the bore if its is a blind hole.

The ball bearings which I have used are 6.35mm (1/4 in.) ID, 12.7mm (1/2 in.) OD miniature stainless-steel bearings which are sealed to retain grease. Notice from the bending-moment calculations that only about 5.32 Nm (46 lbf.-in.) moment is required to impart a 3m (118 in.) radius to the shaft. This means that bearings located 127 mm (5 in.) apart must take only about 42 N (9.4 lbf.) load which is fully acceptable for these bearings at speeds like 500 rpm. The ball bearings also take the driving thrust of the prop. The 3m (118 in.) is typical for some of my boats, but I have used shafts bent to even smaller radii.

I have used both vertical, up-and-down retraction and swing-up-to-one-side retraction with equally good results. It is important to provide a way to lock the shaft strut down into its running position because it has a tendency to climb toward the surface.

References:

1. Standard Handbook for Mechanical Engineers, 7th Edition Baumeister and Marks, McGraw-Hill 1967.

2. Metals Handbook Ninth Edition Vol. 1 ASM 1978.

3. Heat Treater's Guide ASM 1982.

Shield's sample calculations are given on the next page. –ed.

Calculations of Stress in Bendy Propeller Shafts for Human-Powered Boats
(For the formulas used here see ref. 1, chp. 5)

The radius of curvature of an elastically bent beam is:

$$R = \frac{EI}{M_b}$$

Rearranging,

$$M_b = \frac{EI}{R}$$

Where, E = Young's Modulus
I = Moment of Inertia of beam section
M_b = Bending Moment

Then, for a circular cross-section beam:

$$M_b = \frac{\pi E d^4}{64R}$$

where d = circular diameter.

Let's consider a 6.35 mm (.00635m) (1/4 in) diameter shaft.

For, R = 3m (118 in.)
E = 2 x 10^5 Mpa (2.9 x 10^7 psi) for type 630 stainless steel.

$$M_b = \frac{2.0\times10^{11}\times\pi\times.00635^4}{3.0\times64} = 5.35 \text{ Nm (46.29lbf-in)}$$

Let's assume that we want to transmit 1 kw (1.34 hp) @ 500 rpm.

Then 1 kw = 1000 mN/sec = 60000 mN/min.

Therefore, the torque on the shaft will be:

$$M_t = \frac{60000}{2\times\pi\times500} = 19.099 \text{ Nm (169 lbf-in)}$$

The torsional stress in the shaft is given by:

$$S_{xy} = \frac{16M_t}{\pi d^3}$$

$$= \frac{16\times19.099}{\pi\times0.00635^3} = 379.89 \text{ Mpa (55000 psi)}$$

The bending stress in the shaft is given by:

$$S_x = \frac{32M_b}{\pi d^3}$$

$$= \frac{32\times5.32}{\pi\times0.00635^3} = 211.64 \text{ Mpa(30696 psi)}$$

Now, from ref. 1, p. 5-28 the combined alternating stresses in the shaft in service in a pedal-powered boat while transmitting 1 kw (1.34 hp) and bent to a radius of 3m (118 in.) are:

Normal stress, $S_n = \frac{1}{2}(S_x \pm \sqrt{S_x^2+4S_{xy}^2})$

$$= \frac{1}{2}(211.64 \pm \sqrt{211.64^2+4\times379.89^2})$$

$$= +500.17 \text{ Mpa (+72544 psi)}$$

$$= -288.53 \text{ Mpa (−41848 psi)}$$

Shear stress, $S_s = \pm\frac{1}{2}\sqrt{S_x^2+4S_{xy}^2}$

$$= \pm\frac{1}{2}\sqrt{211.64^2+4\times379.89^2}$$

$$= \pm394.35 \text{ Mpa (±57196 psi)}$$

At 500 rpm (500 cycles per min.), if we expect 10^7 cycles fatigue life, which is conservative for this material, the shaft is good for over 333 hours service. If you could keep them pedalling 8 hours a day at 15 km/hr, four strong pedallers could cross the Atlantic in about six weeks.

Cheetah Sprints to World Record

Last September, Cheetah, a lightweight composite bicycle fitted with an aerodynamic fairing, set the world speed-cycling record of 68.73 miles per hour. The bike's developers, three mechanical engineering graduates, modeled the bike after a high-speed cycle they designed while in college.

Steven Ashley
Associate Editor

We have been given permission by Jay O'Leary, the editor of MECHANICAL ENGINEERING, the magazine of the American Society of Mechanical Engineers, to use this superb article by Steven Ashley on the Cheetah. It gives more technical background information than we have had on any previous land HPV. The ASME has done a great deal to foster HPV construction and competition at universities, and we greatly appreciate its further courtesy in being allowed to reproduce this article. We wish that we could have produced it in its original color.

Dave Wilson

The brilliant sun had just ducked beneath distant mountains when a dark streamlined shape streaked down the flat roadbed snaking across the arid floor of Colorado's San Luis Valley. The dead calm of the high-desert evening was barely broken by the whoosh of the slippery shape as it sped by a small knot of observers and a larger audience of sagebrush and rocks. Soon wild cheers reverberated against the valley walls as the two-wheeled vehicle finally slowed to the muted sounds of heavy breathing and turning gears rising from under the black wing-like canopy. As the top of the thin fairing was removed, the panting but beaming rider inside pointed to the cockpit speed indicator, which read 69 miles per hour. "Cogito ergo zoom!" he exclaimed. Translation: "I think, therefore I go fast!"

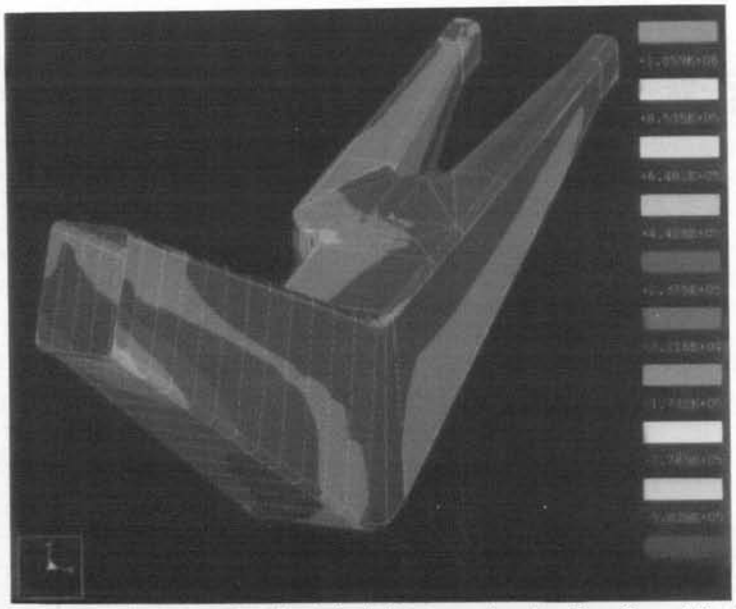
Climbing out of the oddly shaped contraption was 1989 U.S. Pursuit Champion Chris Huber of the Coors Light cycling team. Huber had just ridden into the record books aboard Cheetah, a semirecumbent composite bicycle fitted with a lightweight aerodynamic fairing. Cheetah neared its feline namesake's top velocity when the high-tech bike clocked an average speed of 68.73 miles per hour through a 200-meter speed trap on that desert road on September 22, 1992.

The new world speed record for cycles was certified by two witnesses from the International Human Powered Vehicle Association (IHPVA), the official sanctioning body for these events. The previous record of 65.48 miles per hour was set in 1985 by Gardiner Martin's Gold Rush cycle (Watsonville, Calif.), which won the Du Pont Prize for reaching 65 miles per hour.

Yelling the loudest at the finish were Cheetah's designers and builders – a trio of mechanical engineering graduates of the University of California (Berkeley), who had been working on the bike for four years. They are Kevin Frantz, the coordinator of the Cheetah project, currently a sales engineer for General Electric Supply Automation Group (San Jose, Calif.); Jon Garbarino, responsible for the composite fabrication and molding, now a mechanical design engineer at computer disk-drive maker Western Digital Corp. (San Jose, Calif.); and James Osborn, who handled precision machining operations and computer-aided design, finite element modeling, and aerodynamic analysis of the bike, now a computer engineer at Lawrence Berkeley Laboratory (Berkeley, Calif.).

Cheetah is a refinement of a high-speed cycle design developed by the trio and other mechanical engineering students at the University of California beginning in 1987. At 29.5 pounds, Cheetah is heavier than a good racing bike, but its 8-pound aerodynamic shell makes it much faster.

The project to develop Cheetah, which cost some $50,000 not including donations, was sponsored in large measure by the Adhesives & STructural Materials division of Dexter Hysol (Pittsburg, Calif.), which provided composite building materials, adhesives, and space to work. Cheetah's precision lightweight bicycle components were donated by Italy's Campagnolo Corp. Additional contributions were made by Continental Tire Co. (Hannover, Germany) and Supracor (Sunnyvale, Calif.), makers of thermoplastic honeycomb materials.



Analyzing stresses. Stress analysis representation, using the Applied Structure package from Rasna Corp., of Cheetah's composite frame shows combined tensile, bending, and shear stresses when the front edge is fixed and the rear forks are loaded vertically in opposite directions.

Concept Z

When the three engineers were undergrads at the University of California in 1987, Garbarino directed a core group of seven engineering students who developed a two-wheeled human-powered vehicle for a collegiate competition. The composite cycle, called the Concept Z, placed in several ASME and IHPVA races, posting a top speed of 52.45 miles per hour.

Before designing the Concept Z, "we evaluated other record-setting cycle designs and tried to integrate their best features into our bike," Frantz said. "At first, we were interested in a tricycle layout, but we dropped it in favor of a two-wheel configuration because a three-wheeler couldn't use standard bike components, which are optimized for light weight and good operation. Aerodynamics, weight, and rolling friction considerations also led us to a bicycle design."

"You're shooting for a reduction in the cycle's frontal area," Garbarino explained. For example, some bikes have the rider in a reclining position. This configuration is a trade-off between the amount of power a rider can generate while reclining and the degree of control he or she has over the vehicle, since the rider's position affects the bike's center of gravity and the rider's ability to balance. The bike's general configuration comes down to a subtle compromise among the issues of vehicle handling, rider comfort, and aerodynamic efficiency.

To establish the frame geometry for the Concept Z, Garbarino said that his team built a steel prototype that featured an adjustable wheel base, seat location, steering location, and pedal position. An earlier study on bike stability showed that handling is largely determined by the angle of the head tube and the wheel offset–the distance from the wheel axis to the head tube axis. For this reason the steel bike had an adjustable head tube axis and adjustable-offset forks as well.

"We compromised on a semi-recumbent seating position, which is a typical cycling position rotated to the rear about 90 degrees," Garbarino said. "This means that the bottom of the pedal circle lines up with the rider's rear end and that the rider's feet and legs are in line with the rest of the body." This makes the rider's required muscle development similar to that needed for a standard bike, reducing the amount of special training.

After graduation in 1989, the three cycling enthusiasts submitted a proposal to Dexter Hysol asking that it sponsor a project to build a better-performing bicycle based on the Concept Z. Hysol officials agreed to support the trio's proposal and work began to reconstruct the Concept Z with special emphasis on stiffening high-stress points and constructing an improved seat.

Though Cheetah was completed in June 1990, testing continued at a slow pace. Repeated bad weather and scheduling difficulties with riders and sanctioning officials led to long and frustrating delays before the successful run was undertaken last fall.

Designing Cheetah

Cheetah's carbon composite/aluminum structure was designed on the ME10 CAD package from Hewlett-Packard Corp. (Palo Alto, Calif.) and evaluated by Applied Structure finite element analysis software from Rasna Corp. (San Jose, Calif.). The frame is a prepreg carbon fiber layup over a removable urethane foam mandrel that resulted in a stiff stressed-skin structure weighing only 3 pounds. "Though the frame is locally flexible, its enclosed box section has precision-machined aluminum parts glued into the open ends to give torsional stability," Garbarino said. The aluminum inserts bonded into the ends with Hysol EA 9309 adhesive also distribute high localized stresses into the composite frame.

"We learned on the Concept Z that the area around the intermediate gear had to be reinforced, because the torque transmission there caused the graphite skin to twist," Garbarino said. "For localized stiffening near key components such as the intermediate gear, the rear brake mount, and the steering damper, we added thin (0.030-inch) carbon reinforcing panels."

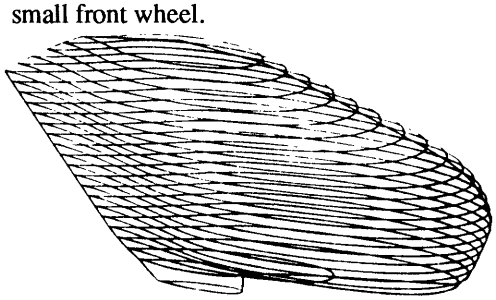
The bicycle's custom-fitted seat is a one-piece hollow carbon/honeycomb structure that weighs 1.5 pounds. Supracor honeycomb material serves as seat padding. The new seat geometry provides a substantial improvement over the Concept Z seat, which impeded blood flow to the riders' hamstring muscles, causing pain.

The Cheetah has a fixed loop of bicycle chain in the front that runs from the pedals to the center intermediate gear. This front loop upgrades the gearing ratio before the power is transmitted to a seven-speed derailleur gear in the rear. "Chris Huber, being a fast spinner, pedaled at about 113 rpms during the run," Frantz said. "He was pushing the equivalent of a 93-tooth chain ring in the front and a 12-tooth ring in the rear with a standard-size rear wheel," he noted. A standard bike has a 52-to-12-tooth-ring ratio.

The wheels, both from Campagnolo, were off-the-shelf products featuring reduced weight. To lower Cheetah's frontal area, a small 20-inch-diameter 18-spoke wheel with sidecovers was used in the front. In the rear, a monocoque-type tensioned disk wheel (700c-size) reinforced with high-specific-strength aramid radial strips in tension rather than conventional wire spokes was installed. Both wheels were fitted with lightweight aramid tires developed by Continental Tire for the latest Olympic cycle races. The low-rolling-resistance racing tires were pressurized to 140 pounds.

Aerodynamic Fairing

The prospect of riding at high velocities while enclosed in a fairing brings special difficulties to the rider. "When you're going almost 70 miles per hour in a crosswind, riding with the fairing on is like holding a sheet of plywood in the wind," Garbarino said. "Riders get off-balance because they tend to overreact to wind gusts. That's because the fairing stops the rider from feeling the wind on his or her face, which is the prime clue a rider uses to gauge how much to compensate. We installed a damper off of a mountain bike in the front of the frame to damp the rider's response to the wind so that he or she can adjust without overreacting." The damper also damps out road vibrations from the small front wheel.



Aerodynamic fairing. Cheetah's aerodynamic fairing–the secret of its record-setting speed–is composed of a series of drag-minimizing airfoil sections stacked one on top of the other.

The controversial approach the team took in developing the shape of the Cheetah's fairing was to reduce drag and increase its aerodynamic design. "Gold Rush, the former record holder, had a fairing that fit as tightly as possible around its rider and frame in an attempt to minimize frontal and surface area, but we felt that its bulbous form-fit configuration didn't offer the lowest drag co-efficient," Garbarino said.

"Though we didn't have the time or money to test fairing shapes in a wind tunnel, we did work with Michael Selig, now an assistant professor of aeronautics and astronautics at the University of Illinois (Urbana-Champaign). Selig has become known for his work in low-speed aerodynamics," Frantz said. Selig had been obtaining good results developing airfoils for California radio-controlled glider enthusiasts preparing for world competitions, which is why Frantz asked him to get involved with the Cheetah project. Selig agreed to coach the team through the aerodynamics part of the project. This was a big boost, considering the difficulty the team had finding information on low-speed aerodynamics. The bike's Reynolds numbers are in the 4 million range. "Our previous airfoils had been plucked from old NACA (National Advisory Committee for Aeronautics, NASA's predecessor) books, which concentrate on high-speed airfoils," Frantz said. "Under Selig's influence, we decided to optimize the fairing's aerodynamics rather than minimize its surface area."

"The optimization of the fairing shape was almost a blind design on Selig's part," Garbarino said. "Essentially, he was designing wing sections based on e-mail messages that gave him target ranges for Reynolds numbers and the physical dimensions we'd taken from the rider and the frame regarding interference points. He knows how to tweak an airfoil shape to get the results he wants."

From earlier experience, Selig had established that if the thickness-to-chord ratios of the airfoils used in shaping the fairing were kept under 15 percent, drag could be kept to a minimum. "The Concept Z had a 20 percent thickness-to-chord ratio, which yields a drag coefficient for an airfoil section of about 0.008," Selig said. "By using a longer shape with a 15 percent ratio, the drag coefficient could be dropped to 0.0055."

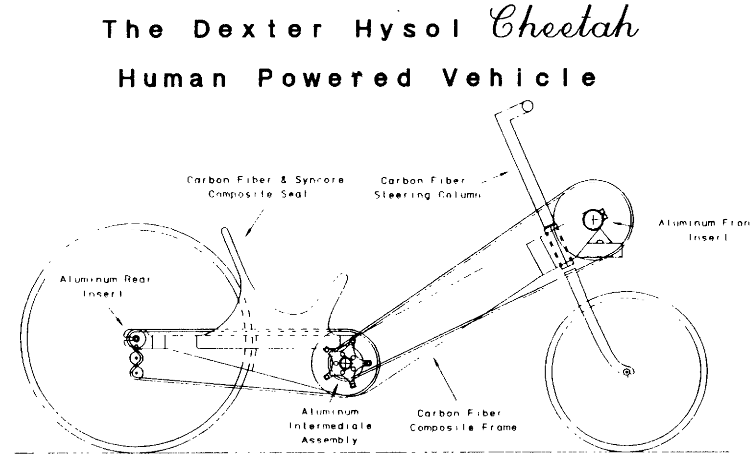
"I used the Eppler inverse design code to find the required airfoil shapes based on velocity distributions," Selig said. "Then, using another part of Eppler, I analyzed the resulting airfoils to find their drag characteristics. Finally, I confirmed those drag calculations with Mark Drela's (of MIT) ISCS analysis package," he explained.

The airfoil shapes transmitted by e=mail were maintained by the engineers using graphical scaling and fitting software Osborn wrote for the Macintosh computer. The computer translated these shapes into a series of dimensional coordinates along the airfoils, each of which represented a horizontal slice of the fairing outline. Osborn used the ME10 CAD software running on an engineering workstation to create a spline fit. Full-scale plots of the plans were printed out on one of the Berkeley lab's electrostatic plotters.

"We glued the full-size plots onto lengths of 2-inch styrofoam, which we cut with a hot-wire table," Garbarino said. "After stacking the layers like a wedding cake to form the general shape of the fairing, we shaved away the excess, ending up with a smooth shape." The foam surface was stabilized with a thin cover of fiberglass and smoothed with automotive finishing compound to produce the male master molding plug. "We molded the female molds for the fairing in four quarter sections, which we bolted together to form top and bottom 'tubs'," Garbarino said.

"Then we did a wet layup of 5-ounce carbon fiber material (one layer at 0 and 90 degrees, the other at ±45 degrees) with Hysol's low-viscosity long-pot-life EA 9396 adhesive to form a fairing about 0.012 inch thick," Garbarino said. A vacuum bag was applied at this juncture to the inside of the mold to wick out excess resin and consolidate the layers as the layup underwent room-temperature curing. Finally, the engineers installed internal ribbing of carbon fiber-covered styrofoam in the shell and glued in the Lexan window. The fairing has three attachment points, which are taped to aluminum frame members with duct tape during speed trials.

Although it has attained its stated goal, the Cheetah cycle is not necessarily retired from action, the three engineers said. There is a possibility that Huber and the bike could attempt to break the one-hour distance record of 47 miles, an event that is conducted on a closed-loop course. Indeed, Cheetah's current short-distance speed record may be challenged some time in the future by other vehicles. In particular, the previous record holder, the Gold Rush cycle, could be modified to run several miles per hour faster with a new optimized fairing.



Record Breaker. This side view of the Cheetah cycle depicts major substructures.

RIM TEMPERATURES DURING DOWNHILL BRAKING

by David Gordon Wilson

SUMMARY

A simple model has been developed that appears to represent closely enough the rise in temperature experienced by bicycle-type wheels with rim brakes during downhill braking. The model shows that only by going very fast or very slow can high temperatures be avoided on steep hills. If dangerous devulcanization of tubes and patches is to be prevented, brakes must be applied evenly to both wheels. It is desirable that the rim strip cover the whole metal surface, and not just the spoke nipple heads, to provide thermal insulation.

INTRODUCTION

This work had several precursors. One was my experience crossing a pass over the Austrian Alps on a heavily laden bicycle. The steep descent was marked by the wrecks of cars the brakes of which had obviously failed. I felt very superior, because my rim brakes were working beautifully. I stopped at a lookout point to admire the view and to show off among the motorists. Both my tires immediately deflated, along with my ego. The rims were too hot to touch. When I removed the tires, all the patches on the tubes had lifted.

The second factor in the present work was a pair of high-speed front-tire failures, one happening to me and one to Scott Martin and his captain[1] on a tandem who happened to be being photographed at the time. He wrote that they were doing the Davis Double Century and were going down a "big hill with lots of curves. They are doing about 35 mph . . . . ." The photographs show that they have just rim brakes: there is no additional disk or drum brake. The captain appears to have the brakes on at the point the tire came off the rim and exploded. Nasty injuries resulted, even though all participants just hit the road surface. They could have been dumped on to rocks or into the path of a motor vehicle.

The third precursor was a delightful 1200-km, 750-mile, trip among the Southern Alps of New Zealand in April 1993. Ellen and I were graciously offered the loan of a superb Gary Fisher mountain-bike tandem, with Shimano cantilever brakes and 26x2.25" tires on aluminum-alloy rims. The mountain passes were generally precipitous, requiring the lowest granny gear for the ascents and full white-knuckle application of the brakes for long periods for the descents. There were almost no guardrails to prevent a wayward vehicle from plunging over the side of the road into rocky gorges along much of the road. There was often at most a very narrow soft shoulder. One had to steer close to the road edges when one of the very occasional motor vehicles passed. I knew that if the front tire deflated suddenly we would be dumped suddenly and unceremoniously, and we would be unlikely to go straight ahead. Letting the brakes off to stop heating the rims would lead to a totally impracticable speed. Having the brakes on harder and harder to produce a slower speed did not seem intuitively likely to reduce the rim temperature. I applied the rear brake more than the front and produced a sudden flat in the rear tire during a precipitous descent when the tube split along the seam line. The heating of the high-expansion aluminum-alloy rim also seems the likely cause of a rear-spoke failure that was impossible to fix in the mountains. I resolved to study the rim-heating problem through a thermal model when I returned.

The model has confirmed my intuitions as being reasonable. It also suggests some measures to alleviate the problems coming from rim heating.

THE MODEL

On a steep downhill the rider presumably does not pedal. The force down the slope is mg sinα, where α is the angle of the hill and sin α is equal to the slope in percent divided by 100 (figure 1). A twenty-percent hill gives sinα = 0.20. In British usage this would be a "one-in-five" hill.

The restraining forces up the hill are the rolling resistance, mg.cosα .Cr, where Cr is the rolling resistance; and the aerodynamic drag, Cd.Ax.ρ.C^2/2, where Cd is the aerodynamic drag coefficient, Ax is the frontal area, ρ is the air density, and C is the relative velocity of the air. The following analysis is for still-air conditions, so that C is the vehicle velocity.

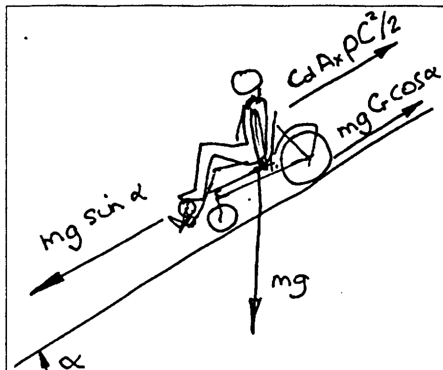


Fig. 1 Forces on a slope

The analysis is also for steady-state conditions. The vehicle is considered to be in dynamic equilibrium, the brakes providing the balancing force, and in thermal equilibrium, which will be closely approached after about a minute of steady braking.

The power dissipated in braking will be the braking force multiplied by the vehicle velocity. At zero velocity the power dissipated is zero. The dissipation rises linearly as maximum braking is multiplied by the increasing velocity with negligible aerodynamic drag until relatively high velocity is reached. Then the braking dissipation reduces fast as the terminal velocity is reached.

The full steady-state power-dissipation relation is:

$$\left[mg.\sin\alpha - mg.\cos\alpha.Cr - Cd.Ax.\rho.C^2/2\right]C$$

The (unbraked) terminal velocity is shown as a function of road-surface slope for a typical single bicycle, a tandem and a streamlined HPV in figure 2, with the

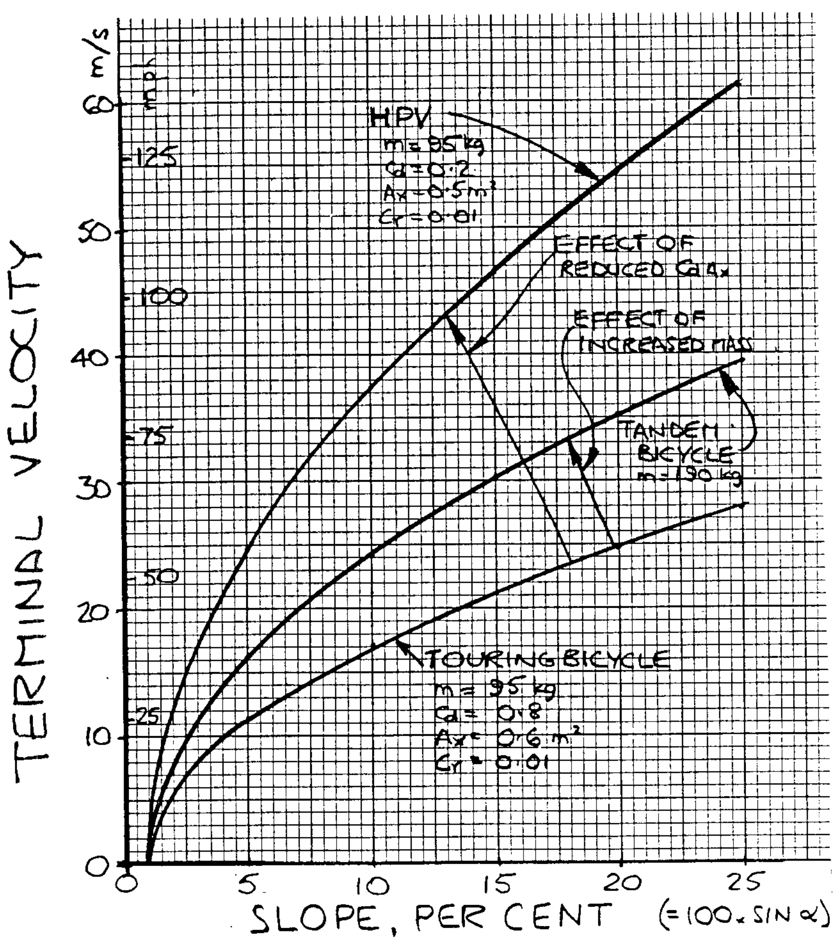


Figure 2: Terminal velocity versus slope.

choices made for the various parameters shown on the graph.

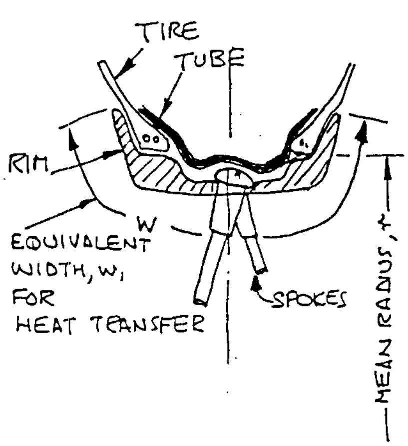


Figure 3: The thermal model

The thermal model

Braking energy is virtually all dissipated in heat at the brake-block-rim interface. Because brake blocks have generally a low thermal conductivity and small surface area, the heat will be conservatively assumed to be all transferred to the rim.

The rim will get hot, and some heat will be conducted to the spokes and to the tire, and some heat will be radiated to the surroundings. Again, these paths will be conservatively approximated as zero. In steady state, all the heat generated by the brakes will be considered to be transferred from the rim by convection.

The convection relation

The air flow around a bicycle wheel is extremely complex, even in still air. Wheel rotation pushes the stagnation point, around which the air parts to go "up and over" or "down and under" or around the tire, well below the midpoint of the front part of the wheel. The flow that goes around the tire on to the rim will usually separate in eddies, rather than stay attached to the surface. Separated flow performs less cooling than attached flow. The rear part of the front wheel and the whole of the rear wheel operate in already disturbed flow. Tires vary enormously in their shape and roughness; rims are of very different shapes; and the presence or absence of fenders (mudguards) affects the flow. There is, therefore, no chance of finding either a theoretical model, or a correlation of experimental results that will accurately represent a real flow.

The best I could do was to find a paper that considered the fluid dynamics and the heat-transfer characteristics of turbine disk[2]. Among the test data was a curve for the friction-moment coefficient, Mf, of an unshrouded disk versus Reynolds number, Re. The range of Reynolds number applicable to HPVs was well away from the given data. I therefore extrapolated the curve and linearized it in the region of interest. The resulting relation is:

$$M_f = \left[0.6 + 1.125\left(1 - \frac{Re}{10^6}\right)\right]\frac{1}{100}$$

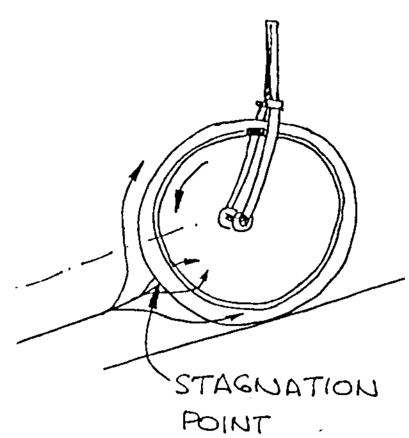


Figure 4: Flow around a wheel

$$\Theta = \frac{100\,\mu\left[m.g\,(\sin\alpha - C_r.\cos\alpha) - \frac{1}{2}\rho.C_d.A_x.C^2\right]}{4.r.w.k.\rho\left[0.6 + 1.125\,(1 - Re/10^6)\right]}$$

We can insert typical sea-level 20C (68F) values for air properties:

$\mu = 1.9\times10^{-5}$ kg/m.s., $k = 0.025$ W/m.degK

$\rho = 1.2$ kg/m³. Then

$$\Theta = \frac{9.81\,m\,(\sin\alpha - C_r.\cos\alpha) - 0.6.C_d.A_x.C^2}{100.r.w\left[0.6 + 1.125(1 - 0.0632.r.C)\right]}\ \text{degC}$$

for m in kg; A_x in m²; r & w in m; & C in m/s

Re is the Reynolds number of the disk or rim, defined as:

$$Re \equiv \rho C r/\mu$$

$\rho \equiv$ air density, kg/m³

$r \equiv$ mean rim radius, m

$\mu \equiv$ air viscosity, kg/ms

Bayley and Owen also give a heat-transfer relation for disks as a function of the friction moment based on an earlier analyst's application of Reynolds' analogy between fluid friction and heat transfer:

$$\frac{\pi \dot{Q} r}{2\pi r w k \Theta} = Re.M_f$$

$\dot{Q} \equiv$ heat transfer, W/m²

$k \equiv$ air thermal conductivity W/m degK

$\Theta \equiv$ temperature difference, rim-to-air, deg.C

Reynolds' analogy works well where flow is attached, but the heat transfer is low where the flow is separated[1], as it is likely to be around much of the wheel-rim surface. Therefore this model may underestimate the temperature rise of the rim. On the other hand, I have neglected the beneficial cooling effect of the cross-flow.

The best that can be hoped for with this model is that it fairly represents the trends of change with velocity. We can adjust constants as data become available.

The full relationship is shown in the box above.

RESULTS OF TRIALS

I made one basic set of trial calculations, for a "regular ten-speed" single bicycle and a big rider (total mass 95 kg, 209 lbm) with other factors chosen as shown in table 1.

The results of the calculations are shown in figure 5. The basic curve is that of the standard bicycle on a 20-percent slope. I chose an overall multiplier to give a peak temperature rise (over the temperature of the air) of about 80 deg C, 150 deg F for even heating of both wheels, because that seemed to agree with experience. The peak temperature was given at about 5 m/s, 10 mph, and drops at higher speeds. The power dissipation rises with speed, but so does the cooling and so does the air drag that reduces the rate of increase of power dissipation. At the terminal velocity for the slope the dissipation is zero.

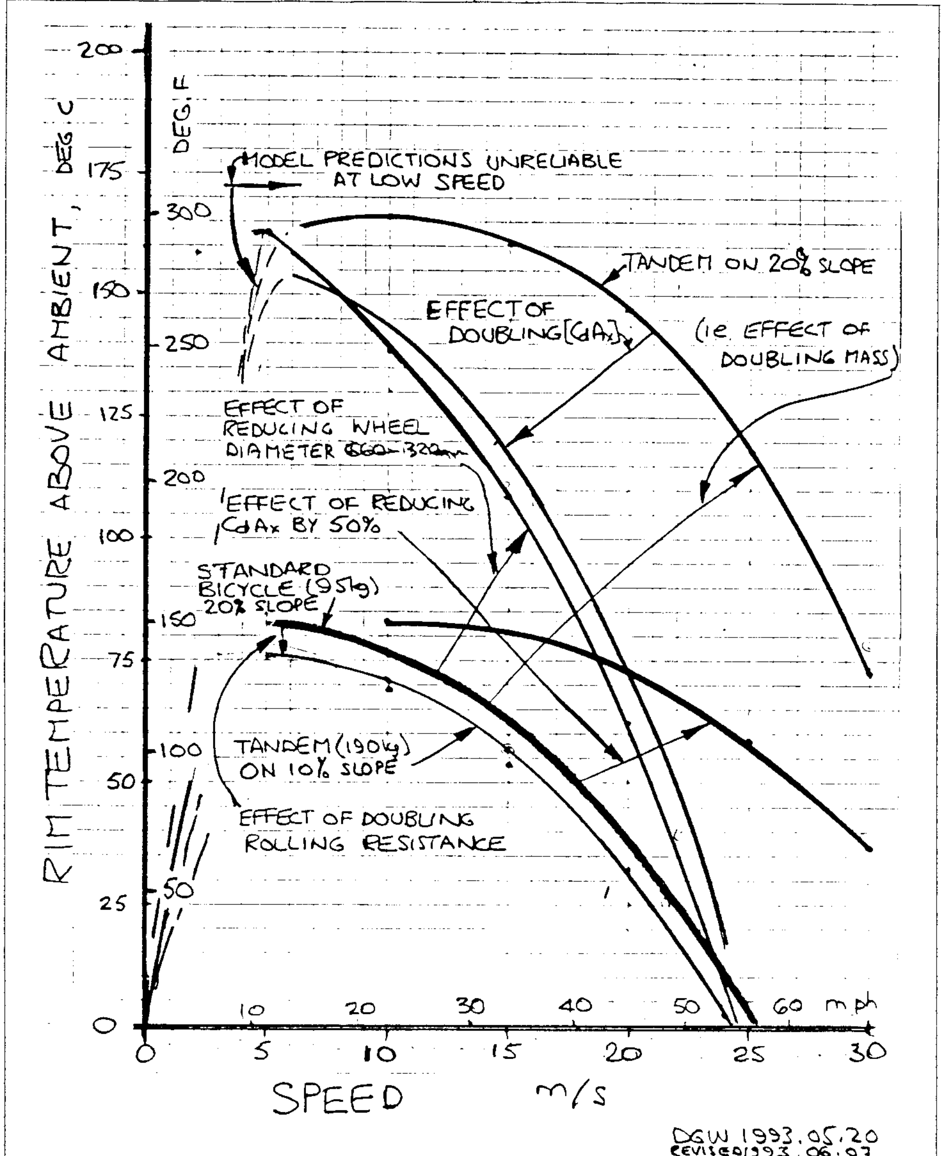


Figure 5 Temperature rise of rims during long braking on hills

Table 1
TRIAL PARAMETERS FOR CALCULATIONS

	TANDEM	SINGLE
Cr	0.01	0.01
Cd	0.78	0.78
r, wheel	0.33m	0.33m
Ax,	0.6 m^2	0.6 m^2
w	0.05m	0.05 m
m	190 kg	95kg

Cr is the rolling-resistance coefficient (rolling drag is Cr . weight or normal force)
Cd is the drag coefficient based on frontal area
r is the wheel-rim diameter
Ax is the frontal area of the machine plus rider
w is the effective width of one wheel rim
m is the combined mass of rider(s), machine and luggage.

There is, therefore, some advantage in going fast downhill in reducing rim temperatures. However, it can be seen that the reduction in temperature is small until a very high speed, eg 20 m/s, 45 mph, is reached. If at such a speed one had to brake sharply to reduce speed to get around a bend, for instance, the heating will be greater than that shown in these steady-state calculations, and the danger of tube and tire failure would be great.

The effect of doubling the mass, which is the only difference between the calculations for a standard bicycle and a tandem, can be seen to have very serious implications for tandem riders. Temperatures can be reached at which tire failure would be very probable unless special precautions are taken (see later).

Almost identical curves were produced for a tandem on a ten-percent slope and for a standard bicycle on a 20-percent slope but with tires having twice the rolling resistance. Therefore it is easy to conclude that slope has a large effect on rim temperature, but that the effect of changing the rolling resistance is negligible.

Reducing the wheel size on a standard bicycle has implications almost as serious as for tandem riders. Owners of Moultons and of recumbents with small front wheels should be especially careful.

Streamlining, shown as reducing the product of frontal area and drag coefficient to 50 percent of that of a standard bicycle, increases speed greatly on downhills but has no effect on peak rim temperature. However, at a given downhill speed, eg 20 m/s, 45 mph, the HPV rim temperatures will be far higher than that of the standard bike because aerodynamic drag is contributing very little to braking.

DISCUSSION OF RESULTS

Despite the various uncertainties in the calculations, the results agree broadly with experience. On standard bicycles, rims can get above 100C on slopes of roughly the indicated steepness as evidenced by the hissing and vapour given when the wheels are run into wet grass.

The results do not have to be accurate, however, to indicate the possibility of hazards and of ways of avoiding them.

Hazards

The calculations are for absolutely equal braking on two wheels of equal size and in equivalent airflows. In practice, the very different friction characteristics of front and rear brake cables (generally reversed on recumbent versus conventional machines) means that it is very difficult for a rider to exert equal forces at the brake pads. If one rim is braked harder than the other (regardless of which is taking the greater load) it will get proportionally hotter.

Vehicles with small wheels will experience higher rim temperatures than those with larger wheels. Recumbent riders on machines with a smaller front wheel should, on long steep downhills, apply a larger braking force to the larger wheel.

Corrective measures

Brake cables should be lined with low-friction material and well lubricated, and all unnecessary bends should be eliminated, so that even braking can be applied to both wheels.

Where there is a choice of rim-shape, streamlined shapes will run cooler than bluff shapes both because separation of the flow is inhibited and because a greater cooling area is presented.

Drum and disk brakes are greatly to be preferred in machines on which long downhills are taken because they completely eliminate rim heating from braking dissipation. Rim heating is more serious in tandems, and many such machines incorporate a rear disk or drum brake particularly for use on long downhills.

The easiest palliative - not to be regarded as a complete safeguard - is to wrap the inside of the rim with rim tape to a thickness that will provide thermal insulation. The glue must withstand temperatures that might reach 150C in extreme conditions.

CONCLUSIONS

Rim heating due to braking on long downhill runs can lead to serious danger from sudden tire deflation. A model has been developed that approximately represents actual conditions. It leads to recommendations, made above, for various measures to lessen the hazards.

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A CONTROVERSIAL ISSUE: HUMAN ENERGY ACCUMULATORS FOR LAND HPV

by
Peter A. Sharp

INTRODUCTION

Human energy accumulators may be the next step in the evolution of human powered vehicles (HPV). Like multiple gearing and aerodynamic streamlining, they promise improved efficiency, greater utility, and more driving fun. Within a decade or two, we may come to regard HPV without accumulators as part of a passing paradigm just as many of us now regard conventional road bicycles as part of a passing paradigm. But the future of accumulators may be significantly accelerated or delayed depending upon the competition rules of the IHPVA. The purpose of this paper is to help clarify the issues, to suggest a better rule compromise than the current one, and to urge its adoption.

My present goal is to work toward the development of HPV and hybrids which can realistically compete with automobiles for the majority of trips (which are less than 5 miles). If developing countries, such as China and India, emulate the U.S. and Europe in their use of automobiles, the resulting pollution, social degradation, and fossil fuel dependence could move the world more quickly toward the impending environmental crisis point (which is being precipitated by over population and inappropriate technologies). As a citizen of the Earth, it is my obligation to work toward sustainable and environmentally benign technologies.

HPV INEFFICIENCY

Current HPV, even the most efficient ones, waste large portions of their available energy. Matt Weaver's torpedo-like "Cutting Edge" won the criterium two years in a row, and defeated the superb "Fast" Freddy Markham (once in Gardner Martin's "Gold Rush", and once in the "Gold Rush Le Tour"). Commenting on the races, Weaver wrote, "I reached 56 mph on the straights, and averaged 42.4 mph for 20 miles while coasting and braking for nearly half the distance due to the many turns." And, "...I managed to smoke out a pair of conventional brakes as I blasted into one of the sharper turns.... I would choke until the burnt rubber smoke cleared out of the fairing. This problem was amplified by both the higher speeds, and also because the very low riding position allows me (to) brake much harder — long after a conventional bike would have done a flip." ("The Cutting Edge Streamlined Bicycle", Cycling Science, Sept. and Dec. 1991)

A human energy accumulator would have allowed Weaver to recover much of his energy lost to braking, and would have enabled him to accumulate pedaling energy during the periods of time he spent coasting. He also could have continued to pedal while braking, since a regenerative brake would have recovered that energy as well. An efficient accumulator would have increased both his average speed and his top speed significantly. The purpose of a human energy accumulator used as part of a land HPV is to conserve and distribute the driver's available energy as efficiently as possible over a given route or purpose.

CHARGING AN ACCUMULATOR

An HPV accumulator can be charged from at least three different sources: regenerative braking, regenerative suspension, and pregenerative pedaling. Regenerative braking accumulates braking energy instead of dissipating the vehicle's kinetic energy as heat ("thermal braking"). The braking energy is accumulated and then used to help accelerate the vehicle. Regenerative suspension accumulates the energy that would otherwise be dissipated as heat in the shock absorbers of the suspension system ("thermal suspension"). Mountain bikes, for instance, might benefit from regenerative suspension. Pregenerative pedaling is used to accumulate the driver's pedaling energy that would otherwise not be generated during short periods of no pedaling or reduced pedaling — such as while coasting, while traveling at less than the preferred cruising speed, while braking, and while stopped en route. (I refer to this type of pedaling as "pregenerative pedaling", in contrast to regenerative braking.) Perhaps the main reason we do not already have accumulators is that conventional bicycles can not be pedaled while they are stopped. This is an obvious consideration if accumulators and pregenerative pedaling are to be used for HPV.

The simplest accumulator system would include either regenerative braking or pregenerative pedaling or regenerative suspension. The most sophisticated accumulator system would integrate all three of these. The simplest accumulator system for using pregenerative pedaling would be one that allowed pregenerative pedaling only while the HPV were moving. A more versatile accumulator system would permit pregenerative pedaling while the HPV were stopped. (One technique would be to charge the accumulator by pedaling backward.) Road racing HPV might use only regenerative braking, since pregenerative pedaling could be fed to the accumulator by applying the brake. Practical vehicles might use only pregenerative pedaling, since they do relatively little braking, but rather a lot of standing and coasting. Top speed record attempt HPV, and HPV dragsters, might charge their accumulators using either pregenerative pedaling or regenerative braking, but pregenerative pedaling would be more convenient. As with any technology, there are many tradeoffs to be considered. Different accumulator designs would be better suited to some charging techniques than others.

PREGENERATIVE PEDALING

A practical HPV typically spends a portion of its time waiting for signal lights to change, or moving at a speed which is less than its optimum cruising speed. The driver generates no power, or less than her preferred power, during those periods. An accumulator would permit the driver to pedal continuously at her chosen power level (appropriate for the route and/or purpose), thereby maintaining a higher average power output. The energy which is not generated during periods of reduced or no pedaling may be regarded as lost or wasted energy. It is "ungenerated" energy. (If regenerative braking or regenerative suspension is not used, then the lost energy is "unregenerated" energy.)

The "human engine" has a very limited output and there is often no way to make up for ungenerated energy. Lost pedaling time becomes lost pedaling energy. We are not in the habit of recognizing ungenerated human energy as wasted energy. That is probably because we drive automobiles with enormously oversized engines. If our gasoline engines were limited to about 1 hp, we would soon realize that we needed to run them continuously at their optimum power level, while using an accumulator to distribute their power most efficiently. The energy in the accumulator would be used when more power was required, such as when accelerating from a stop, passing, or climbing a hill. We would then take it for granted that all HPV need an accumulator as well, and that to not use one would be absurdly inefficient.

HPV EFFICIENCY

Because of the very low power that humans are able to generate, it is critically important to maximize the use of all available energy. (This problem is presently analogous to that of power plants. They must often operate at lower power levels than they were designed for because they have no way to store energy that could be used later during peak load periods.) Together, the human energy that could be pregenerated or regenerated, plus the human energy that is actually generated (at the chosen power level), is the "available" energy. A streamlined

HPV equipped with an efficient accumulator would enable the driver to utilize most of his available energy (by reducing ungenerated and unregenerated energy), and thereby maximize the "route efficiency" of the HPV.

The efficiency of an HPV is usually discussed in terms of instantaneous efficiency. For instance, we measure a person's instantaneous output using an ergometer. Or, we measure how well the driver's output is converted directly into vehicle speed. But instantaneous efficiency is only part of a more inclusive kind of efficiency, which I refer to as "route efficiency". Route efficiency is a measure of how well an HPV conserves and distributes all of the driver's available energy to complete a given route or purpose. A higher route efficiency will usually result in a higher average speed or a higher top speed, depending upon the characteristics of a specific route or purpose. To determine the route efficiency, all of the available energy must be included in the calculations. Consequently, other things being equal, an HPV with an accumulator will usually have a higher route efficiency than an HPV without an accumulator. An exception would be when the intended route or purpose includes few or no opportunities for pregenerative pedaling, regenerative braking, or regenerative suspension, such as a one hour time trial on a smooth track.

HPV ACCELERATION

The most important use of an accumulator is to help the driver accelerate more quickly up to cruising speed. Streamlined, recumbent HPV usually accelerate more slowly than a standard racing bicycle until a speed is reached where superior aerodynamic streamlining becomes the most important consideration. This is primarily because recumbents weigh more and have a higher inertia. Streamlined fairings add considerable weight, and thus increase the problem. There are often other contributing factors as well, such as frame flex and seat flex, a too open body bend angle (between trunk and legs), and less freedom to employ or adjust the various muscle groups. This is demonstrated by the fact that most recumbents must use a lower gear and a higher spin rate when climbing steep hills.

The problem of relatively poor initial acceleration is especially apparent when streamlined HPV are compared to automobiles. With an average driver, streamlined HPV are capable of maintaining or exceeding the standard urban speed limit of 25 mph on level roads. But because they can not accelerate as fast as automobiles, they are less able to stay with the normal flow of traffic, and therefore miss more lights and spend more time waiting at intersections. An accumulator would enable a streamlined HPV to better stay with the normal flow of automobile traffic, and to achieve lower trip times. The advantage of using an accumulator for a practical HPV would depend to a great extent on the specific characteristics of a route or purpose. On some routes, such as one with many stops, an accumulator could achieve a significant reduction of travel time. On other routes, such as a long stretch of highway, an accumulator would be of little benefit.

The value of an accumulator, and the better acceleration it provides, lies not only in its ability to reduce trip times or to improve an HPV's chances of winning certain competitive events, such as drag races, road races, and top speed events. Fast acceleration is fun. And not only fun. Fast acceleration tends to confer power and status upon the owner of the vehicle. For that reason, good acceleration is a strong selling point for many vehicles. Most automobiles with high acceleration make little practical use of that acceleration. High acceleration is desirable primarily because it enhances the driving pleasure of the vehicle. Consequently, accumulators may eventually become the critical factor in making HPV appealing to a large percentage of consumers. An accumulator would typically enable an HPV driver to increase his power output during acceleration to two or three times his maximum output, or many times his average output. An HPV with much higher acceleration than a conventional bicycle would have considerable sales potential. It would be a "hot bike".

DEFINING AN ACCUMULATOR

An HPV might also use its accumulator for storing excess wind power generated by a wing sail, or an airfoil fairing. Technically, however, using wind power would make the HPV a hybrid vehicle, and storing wind generated energy would make the accumulator a hybrid device. The same accumulator device could be used for multiple functions ranging from pure human energy accumulation to pure non-human energy storage, with many types and combinations of accumulation and/or storage in between. For instance, a battery with a motor/generator could be used in any of these ways. To qualify as a pure human energy accumulator, as opposed to a human energy storage device, an accumulator must satisfy two conditions. First, the source of the energy must be the driver (and/or the stoker on a tandem). Second, the energy must be accumulated during the intended route or purpose. The second condition requires an arbitrary definition of the distance and/or time interval for the intended route or purpose. The second condition is an especially important consideration when writing rules for HPV competitions.

An HPV using pure human energy accumulation is not a hybrid vehicle. The energy is still human generated energy, but an accumulator permits that energy to be distributed more efficiently. It provides a better route efficiency in a similar way that multiple gearing provides a better route efficiency. An accumulator may be regarded as an extension of an HPV's transmission. It may serve to integrate both brakes and suspension into the transmission system as well. All HPV are aggregates of human energy accumulators of one kind or another. All HPV accumulate energy. (For instance, kinetic energy is accumulated simply by moving). An HPV which is also able to accumulate regenerative braking energy, pregenerative pedaling energy, and/or regenerative suspension energy is a more efficient HPV.

TYPES OF ACCUMULATORS

A human energy accumulator's cycles are typically short intervals that alternate between charging and discharging within seconds or minutes. In that sense, an accumulator is analogous to, or sometimes identical with, an electrical capacitor. There are many types of devices which could be used as human energy accumulators, such as batteries, capacitors, flywheels, compressed gas, springs, rubber bands, etc. In Bicycling Science by Whitt and Wilson (pages 316-317), there is a list showing the energy-storage capability of various materials, and the energy density of different types of systems (fuel, battery, mechanical, etc.).

An ideal accumulator would rank very high in efficiency, capacity, and practicality (quick charging and discharging, safe, low cost, light weight, etc.). But as yet, no system does. For instance, advanced flywheels might provide adequate efficiency and capacity, but they are difficult to build, require complex transmissions, and are potentially dangerous. Battery systems would seem to be the most versatile choice, but they tend to be only about 25 to 35 percent efficient when used as a pure accumulator. However, it may be possible to build a very efficient accumulator if we are willing to accept a low capacity. There seems to be only one circumstance in which a practical human energy accumulator requires a large capacity. That is when braking while descending a long, steep grade. If we are willing to sacrifice a large capacity, then there are at least two good candidates.

One option is the use of rubber bands.

Paul MacCready, with his extensive experience in building rubber powered model airplanes, pointed out that one pound of rubber bands would be enough to accelerate a 200 pound HPV up to about 20 mph (proceedings of the Second Annual HPV Symposium, 1984, pg. 114). Jim Papadopolous has designed an elegant accumulator which uses rubber bands, and which is entirely contained within a rear bicycle wheel (personal communication). Many rubber bands are arranged parallel with the spokes. He has calculated that some synthetic rubber compounds could store more energy than indicated in Bicycling Science.

My own preference is for the use of a vacuum contained in a cylinder with a piston (this option was not considered in Bicycling Science). A vacuum has a very low capacity — about 2,117 ft-lb. of potential energy per cubic foot at standard temperature and pressure. But it would have a very high efficiency — theoretically, 100 percent. It is actually a gravity device, since it displaces and lifts the atmosphere. It also has the minor advantage of providing a constant input and output force. In order to minimize weight, the vacuum cylinders might double as the frame of the HPV. There may be a way to make very light vacuum cylinders that can resist atmospheric pressures without collapsing.

Given the differences in efficiency and capacity between different types of accumulators, sophisticated systems might eventually combine two or more types. For instance, a high-efficiency/low-capacity type could be combined with a low-efficiency/high-capacity type so as to produce the best overall efficiency. The former would handle stop and go driving, and the latter would handle long descents, and probably hybrid power as well.

THE IHPVA'S COMPROMISE

For HPV competitions, the use of accumulators must, of course, be regulated. Used without regulations, they would lead to over specialized and highly impractical racing vehicles which would resemble automobiles. Fortunately, accumulators can be regulated very easily without banning them from any event. The simplest way to do that is to limit the time available for energy accumulation. To encourage relevant innovation, the time limit should be consistent with the time periods typically available to practical HPV for energy accumulation.

Other things being equal, a multipurpose HPV equipped with an accumulator is more efficient than an HPV without an accumulator, just as an HPV with multiple gearing and a streamlined fairing is more efficient that an HPV without them. Nevertheless, the IHPVA prohibits the use of accumulators in the drag races, the top speed event, and top speed record attempts. To not permit the use an accumulator in these events is to deliberately handicap the vehicles by requiring them to be less than optimally efficient. If the goal is to make a pure HPV go as fast as possible, then using an accumulator is both an obvious and a necessary choice. Accumulators are permitted in the annual Human Powered Speed Championship (HPSC) road races, which demonstrates that in the eyes of the IHPVA there is nothing inherently illegitimate, or hybrid, about using an accumulator. Why then a seemingly contradictory and counterproductive prohibition against using them in the drag races and the top speed events?

As far as I have been able to determine, the reasons are more historical than rational. In fact, there seems to exist an actual prejudice toward the use of accumulators, as if using an accumulator in those events somehow would be "illegitimate" or contrary to the "spirit" of those events, and any record would be "contaminated" or "impure", and not a "real" record. That belief seems to be directly analogous to the negative attitude of the UCI toward the use of deliberate streamlining. It seems to be based on traditional assumptions about what constitutes a "real" or "pure" HPV. For some IHPVA members, accumulators still seem contrary to the "purism of the bicycle", even though they have already accepted streamlining as consistent with that purism. It is ironic that an organization founded on freedom of innovation would legitimize the same type of bias that it intended to overcome.

In 1982, under then President Lynn John Tobias, the IHPVA reconsidered the issue of whether or not to allow the use of regenerative braking (and therefore accumulators) in competition. Members were asked for their opinions on a questionnaire, and the result was that members were evenly divided for and against. Tobias proposed the present rule format as a compromise (Human Power, no. 1, Spring 1982). It is not clear whether there was an informed debate on the issue. Permitting accumulators in the road races, but not in the top speed events, may have seemed like a reasonable compromise between those who favored accumulators and those who opposed them. Without knowing the full context in which that decision was made, I am hesitant to criticize it. But we can at least reconsider that "compromise" in terms of outcomes rather than intentions.

As far as I know, no land HPV equipped with an accumulator for regenerative braking or pregenerative pedaling has ever been raced in an HPSC event. Nor have I found even a single research paper in Human Power on the subject of pure human energy accumulators for land HPV. So it seems that the effect of the past and present rules has been to completely discourage research and development on accumulators for land HPV. After 10 years, it is now clear that under the current competition rules there are insufficient incentives to encourage the research and development of accumulators for land HPV. It now seems fair to say that to continue with the current competition rules would be to deliberately suppress the development of accumulators.

A BETTER RULE COMPROMISE

Those who argue that anyone is free to build an accumulator if they want to, and use it in a road race, are closing their eyes to the enormous power of records as incentives. That power is clearly evident in the remarkable advances made in the design of aerodynamic fairings, which is a direct consequence of their use being sanctioned for IHPVA records. So I would like to propose two very simple competition rules that would encourage the development of accumulators while respecting the concerns of those members who still believe that accumulators are somehow inconsistent with the "purism of the HPV". These two rules are the "six minute rule" and the "one minute rule". The "six minute rule" would apply to top speed record attempts. The "one minute rule" would apply to HPSC events. The goal of both rules is to provide approximately one minute for energy accumulation in their respective events. A period of approximately one minute for energy accumulation is generally consistent with the longest time periods that practical HPV spend waiting at intersections or moving slowly while waiting for slow traffic to resume speed. Consequently, accumulators developed to utilize a one minute charging period would have direct application to practical HPV.

THE SIX MINUTE RULE

The rule for top speed record attempts would be the "six minute rule". Six minutes would be allowed, when using an accumulator, to make a top speed record run from start to finish, including any and all human energy accumulation. HPV not using an accumulator would have no time limit in which to complete a run — the same as now. To use an accumulator for a record attempt, an HPV would be required to make two runs, one using and one not using the accumulator, and with the accumulator device on board during both runs. Both speeds would be recorded as part of the official record, and the official record speed would be determined by taking the average of those two speeds. It would be the responsibility of the competitors to convincingly demonstrate to the record officials that no energy was accumulated prior to the six minute period, nor during the run not using an accumulator. The top speed using an accumulator could be included or not at the option of the competitor. Therefore, using an accumulator could help, but not hinder, a top speed record attempt. It is important to note that using the average speed would retain the current dominant emphasis on streamlining, and it would prevent any lopsided attention to accumulators.

My reason for choosing the "six minute rule" (360 seconds) is to allow time for acceleration to 70 mph, plus one minute for energy accumulation. Acceleration profiles would differ with different vehicles and different drivers, so some approximation is required to determine the appropriate time period. For instance, as calculated by Matt Weaver in his article (see his Fig. 2), the time required for him to accelerate his vehicle to 70 mph would be approximately four minutes and 25 seconds (265 seconds), and would require a distance of about 3.7 miles. However, Fast Freddy Markham, in the Easy Racer Gold Rush, reached his record speed of 65.4 mph in approximately 2 minutes and 25 seconds (145 seconds), and required a distance of about 2 miles ("Simulation of the Gold Rush 200M Sprints, by Danny L. Pavish, proceedings of the 3rd IHPVA Scientific Symposium, 1986, pg. 98). This contrast implies that Weaver's calculations may be conservative. Six minutes might therefore provide more than one minute left over for energy accumulation. However, as speeds increased, the time period left over for energy accumulation would continue to decrease. That would encourage the constant improvement of accumulators along with all other components.

The use of an accumulator in top speed record attempts would eventually increase speeds perhaps 5 to 10 mph, but only half that when averaged. This "six minute rule" would provide valuable comparisons and it would not "contaminate" the top speed that could be achieved without an accumulator. I consider this to be a much better compromise than the current one that was adopted ten years ago, and which has proven itself to be no compromise at all, despite its good intentions. This compromise would provide an orderly transition to the use of accumulators.

If, after many years, accumulators became common components on both racing and practical HPV, we then could consider the option of using only the top speed with an accumulator as the official record speed. In the mean time, as speeds increased, the initial difference between the two speeds would diminish because less and less time would be left for accumulation. If necessary, we might eventually consider adding more time to the "six minute rule" so as to insure at least one minute for accumulation. These issues would cause no confusion for the competitors. They would need to know only one simple fact: the time available to complete a record run. They could use that time in any way they wished.

As top speeds increase, it is becoming increasingly difficult to find locations which provide enough distance for record attempts. For example, Matt Weaver mentioned not having had a chance to test his machine on an ideal course for top speed. The use of accumulators should shorten the distances required for acceleration and therefore would make many more places appropriate for record attempts.

An interesting question is what kind of acceleration profile would best utilize this 6 minute period. For instance, would it be theoretically possible to accumulate energy for the first 3 minutes and then accelerate up to 75 mph. within the last 3 minutes? A shortened period of acceleration would reduce the amount of energy normally consumed by drag while slowly building up to top speed. But could anyone invent an accumulator that could be used to implement such a strategy?

THE ONE MINUTE RULE

The top speed event during the annual HPSC races would use a specific distance for acceleration, as it does now. In addition, it would use the "one minute rule". Energy accumulation would be permitted only during the one minute prior to the start of the run. The other procedures would be the same as those described for top speed record runs. The top speeds in this event have leveled off due to the lack of sites with a sufficient distance for acceleration. As a result, it is difficult to demonstrate the true potential of a superior aerodynamic design. Using the "one minute rule" would have about the same effect as providing another mile for acceleration. This would increase speeds sufficiently so as to enable superior aerodynamic designs to demonstrate their superiority.

The rule for a road race or a criterium would be the same "one minute rule". If a flying start were used, energy accumulation would be permitted only during the pace lap. If a Le Mans type start were used, no prior accumulation would be permitted.

Since accumulators would be of little benefit during time trials, no use of accumulators would be permitted. This would enable us to better compare the ranking of vehicles when using accumulators (road race and criterium) to their ranking when not using accumulators (time trial).

Drag races would use the same "one minute rule". However, since the most important function of an accumulator is to provide increased acceleration, the top speed and elapsed time with or without an accumulator would be recorded as the official speed and time. By using the "one minute rule", the accumulator technology developed for the drag races would be directly applicable to practical HPV. The "one minute rule" would lead to exciting drag races because it would enable unfaired HPV that used accumulators to become competitive with fully faired HPV that did not use accumulators.

The policy of the IHPVA is to encourage innovation by reducing competition restrictions to a minimum. We must not abandon that policy. These two simple rules (the "six minute rule" and the "one minute rule") would remove an inappropriate major restriction from the existing rules. These simple rules should provide the necessary incentives to encourage the research and development of human energy accumulators without diminishing in any way the progress of aerodynamic innovations. I encourage members to consider these two simple rules and to improve upon them if necessary.

RELATED ISSUES

Some people might suggest that we place accumulators in a separate top speed category. We know, however, what the effect of that would be. The UCI did that with recumbents, and the result was very little development for almost 40 years. History teaches us that "separate, but equal" almost never is. The only category in which accumulators belong is the open category. It would make no more sense to put them in a separate category than it would to put multiple gears, recumbents, or streamlined fairings in a separate category.

To those who would insist that the IHPVA maintain its tradition of "no stored energy", I would point out that human energy accumulation is easily differentiated from energy storage, and that the IHPVA legitimized human energy accumulation 10 years ago. It is important to note that the use of pure human energy accumulators does not contradict the tradition of "no stored energy". If anyone were to insist that the tradition must be interpreted to mean that accumulators should be excluded forever from top speed record attempts, then I would argue that the tradition was poorly conceived to begin with, and that the time to reconsider that tradition is long overdue.

AN ANALOGY

The use of accumulators in the top speed events is analogous to the use of synthetic materials for vaulting poles. (I happen to consider vaulting poles to be one of many possible types of human powered vehicles.) Wooden poles had always been used. Synthetic poles were developed precisely because they are excellent human energy accumulators (their extreme bending accumulates considerable energy). But when they were first introduced, many people considered them to be "unnatural", "cheating", "not in the spirit" of pole vaulting, "contrary to the tradition" of pole vaulting, and not "real" vaulting poles. Those people were concerned that synthetic poles would tarnish or "contaminate" the glory of previous records, and that new records would be "impure", or even "meaningless". They insisted that using synthetic poles would be changing the rules in the middle of the game. But the ruling to allow synthetic poles turned out to be an excellent decision. Records have reached amazing heights, new techniques have been developed, the old records are still highly respected, and synthetic poles are now considered to be entirely legitimate and appropriate. To now suggest a rule that permitted only traditional wooden poles, with their limited capacity as human energy accumulators, would be considered pointlessly regressive.

CONCLUSION

The IHPVA must now decide, by decision or default, whether or not to allow the use of accumulators in the top speed events. It can do so without diminishing the organization or its records. Human energy accumulators are necessary components for applying the human muscular potential most efficiently. An accumulator is an essential part of an efficient HPV, in the same way that multiple gears and streamlining are essential for achieving maximum efficiency. That is why accumulators may be the next step in the evolution of the HPV. To retain the IHPVA rules as they are would be contrary to the IHPVA's policy (unrestricted innovation) and purpose (to be "devoted to the study and application of human muscular potential to propel craft through the air, in and on the water, and on land"). We must not abandon both our policy and our purpose merely to avoid admitting an honest mistake - that we should never have banned human energy accumulators from the top speed events in the first place.

Now is the time to act. By banning human energy accumulators from record attempts, the IHPVA inadvertently contradicted its own policy and purpose, and contributed to a 20 year delay in the development of human energy accumulators. By updating our rules, we would have everything to gain and nothing to lose. Our organization is a world leader in the development of energy efficient vehicles, and the IHPVA has initiated what may eventually become a world wide transportation revolution. The world is very much in need of fast, highly efficient, and clean running personal vehicles. We must not continue to inhibit an important part of their development. We have come to a fork in the road, and we must decide upon the direction our organization will follow. It is our social responsibility, and our opportunity, to choose the path of innovation with the fewest restrictions and delays, and with the greatest rewards for society. I urge my fellow members to seize the day and vote for innovation. By decision or default, the choice is ours.

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