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HUMAN POWER

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

We are indebted to the authors, to Marti Daily and to Julie Drennan, whose dedicated help made this issue possible.
Dave Wilson

In this issue

A FOLDABLE MWB RECUMBENT BICYCLE (a study of the language of design)

A photo of Nick Abercrombie Andrews' recumbent appeared in the newsletter of the British Human-Power Club. I immediately wrote to find out more. For this bicycle was not only foldable, but had a medium wheelbase: the crank axis was over the front wheel. The Velocar of the 1930s was a MWB, and I have long felt that we need to explore this variant. You will be intrigued and impressed with Nick Andrews' ingenuity and design and construction skills.

A PASSIONATE PLEA FOR AHPVs

Peter Ernst believes strongly that we are being too uncompromising in rejecting all fossil-fueled machines. The assisted-human-powered vehicle in which one has to pedal to get power-assist can make major contributions to reducing fuel use and emissions, and even win some converts to HPVs. He believes that John Tetz's AHPVs and Yamaha's introduction of the PAS bicycle could signal the start of a revolution.

FACTS ON FAIRING FLOWS

Andreas Fuchs defines and discusses the effects of the center of gravity, center of pressure, center of friction and other arcane aspects of faired-vehicle design. He tells you how to become a more-stable person.

REVIEWING A RECUMBENT TANDEM

John Allen finds much fine design and componentry in Steve Delaire's compact and attractive Rotator double-recumbent tandem, and some unpredictable slow-speed-handling characteristics. Explanations and advice from the gurus of bicycle stability would be welcome.

CALIBRATING AN ERGOMETER

Most of us are content to accept the readouts from an ergometer - after all, the name implies scientific measurement. However, a scientific maxim is "don't trust anything." John Raine and his co-authors from New Zealand approach the problem of calibrating an ergometer with typical scholarly care and precision. If you need to know whether or not your ergometer or exercise machine is telling you the truth, this paper will show you how to find out.

Kazuo Ohishi taking off to capture the Japan HPA distance record on Dohgasa beach, December 30, 1993 in Team Aeroscepsy's new HPA "Gokuraku Tombo Special" Span 32m, length 7m, area 28.4 m², weight 32 kg, pedal rpm 87, prop. rpm 145; Power requirement 250 W at 30 km/h cruising speed.
Photo Aeroscepsy; contributed by Toshio Kataoka

HUMAN POWER

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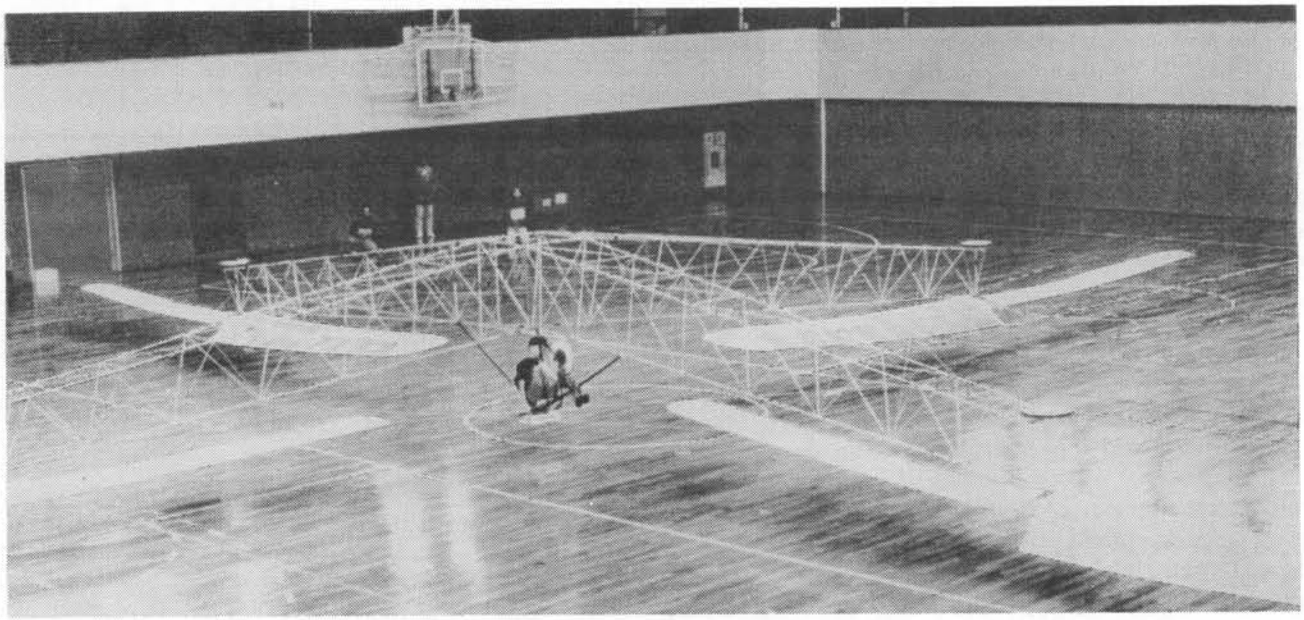


HP helicopter "YURI": official flight March 12, 1994 - 19.46 sec. at 500-600 mm. Pilot Norikatsu Ikenchi. Four rotors 10m dia.; truss frames of Japanese cypress; weight 32 kg, Kevlar wound-rope drive. Photo: Yomiuri Shinbun

Editorials

Congratulations Lec. Naito!

It is easy for your editor to be very enthusiastic about the hover flight of the HP helicopter Yuri. We have learned from several HP articles and from the many attempts at the American Helicopter Society Sikorsky prize that this is the most difficult of the HP contests to date. Akira Naito is the equivalent of professor emeritus at Nihon University and has a stunning record of success in HP aircraft and early helicopters working with students. From what I could tell from the videotape sent by Toshio Kataoka, Yuri is virtually all his own work, and certainly his own radical design. May we all be as creative and productive in our retirement.

Are intelligent vehicle-highway systems intelligent?

"With federal funding for Intelligent Vehicle/Highway Systems approaching $1-billion within the next few years, the IVHS industry is on the verge of explosive growth". Such was the gloating opening of a pitch for "Inside IVHS", a very expensive newsletter. IVHS are electronic methods of letting motorists know how to maneuver around traffic jams. Once the government has subsidized automobile travel to the point where it is such an apparent bargain that the roads choke up, it subsidizes it more. I would call that a highly non-intelligent way of spending my tax money.

Colophon

This is a word I had to look up. Journals have started including colophons to state how they are produced. We always want to join the trend. If someone wants to produce draft articles in as close to the final format as possible, use this.

Human Power is produced on a 486/33 PC using Lotus' Ami Pro on a modified newsletter three-column style. The standard font is Adobe Times New Roman PS 10-pt; headings in Helvetica bold, 11 & 12-pt. With this issue we are printing on Julie Drennan's Hewlett-Packard DeskJet 1200C. We send our camera-ready text plus separate illustrations to Marti Daily who, with her team, puts them all together and sends them out to you.

The new arrangement of the journal pleased those who wrote. I've followed the same arrangement as introduced in vol. 11/1, principally in having all articles start on a page and continue until finished, with letters, reviews etc. acting as fillers. This time I had everything set up except for the editorial page and this page 3, and I combined them.

I'm starting work immediately on vol. 11/3 because I have a happy backlog of good papers (and because this issue is a little late). There will also be a delightful bonus in the next issue. Chet Kyle, who founded and edited Cycling Science until it was sold last year, offered HP two magnificent past articles: Arnfried Schmitz on the Velocar origins, and Matt Weaver on The Cutting Edge.

Dave Wilson

DESIGN AND PERFORMANCE OF A DYNAMIC CALIBRATION RIG FOR A BICYCLE ERGOMETER

by
J.K. Raine, H.P. Trolove & H. Beveridge

ABSTRACT

The Department of Mechanical Engineering at the University of Canterbury, Christchurch, New Zealand, was commissioned by the Physical Education Department to build a dynamic calibration rig for an air-braked Repco cycle ergometer. The project was part funded by the Hillary Commission on the recommendation of the New Zealand Sport Science and Technology Board. This paper describes the conceptual approach and embodiment design of the apparatus and presents calibration test results.

1. INTRODUCTION

Athletes and coaches in most sports are interested in the measurement of performance, and athletes need a reliable quantitative means of assessing their level of fitness or the effectiveness of a training programme. Methods of assessment range from field performance tests, through trials on laboratory exercise machines, to biochemical tests on muscle condition. Quality control is a basic requirement of any method used, i.e. instruments must be accurate and experimental data reliable.

Air-braked cycle ergometers are extensively used in training and in sports-science research for various athletic disciplines in New Zealand, eg. ice racers, skiers, cricketers and triathletes. At present it is difficult to compare performance data from one athlete using the same assessment exercise programme on different ergometers as the various institutions have not had the means to check the accuracy of their equipment. It was seen that a common means of calibrating ergometers accurately would facilitate coordination of sport-science support services nationally.

The Physical Education Department at the University of Canterbury has a number of exercise machines, including a cycle ergometer, which may be used for training and for experimental studies of human physical performance, and to measure physiological response to graded exercise. The Repco HP5209 Air-Braked Cycle Ergometer contains a built-in power meter of limited accuracy, and it was decided to design and build an accurate calibration system for this machine to enable indicated instantaneous power readings from the ergometer to be converted to true readings.

This project aimed to provide a portable system that would allow sports-science researchers in various locations to calibrate their air-braked cycle ergometers, thus establishing a common standard throughout the country. A secondary aim was to provide a rig which could be evaluated as a speed-controlled system for load application and measuring for incorporation into athletic training machines.

The calibration system now commissioned is portable and can be used by sports scientists throughout New Zealand. This will allow comparison of performances of the same athlete tested on different ergometers in the future.

The operation of the equipment is based on precision measurement of the torque and speed of a geared electric motor that drives the pedal shaft of the ergometer. This paper describes the background to the project, details of the design, and calibration test results for the Repco ergometer.

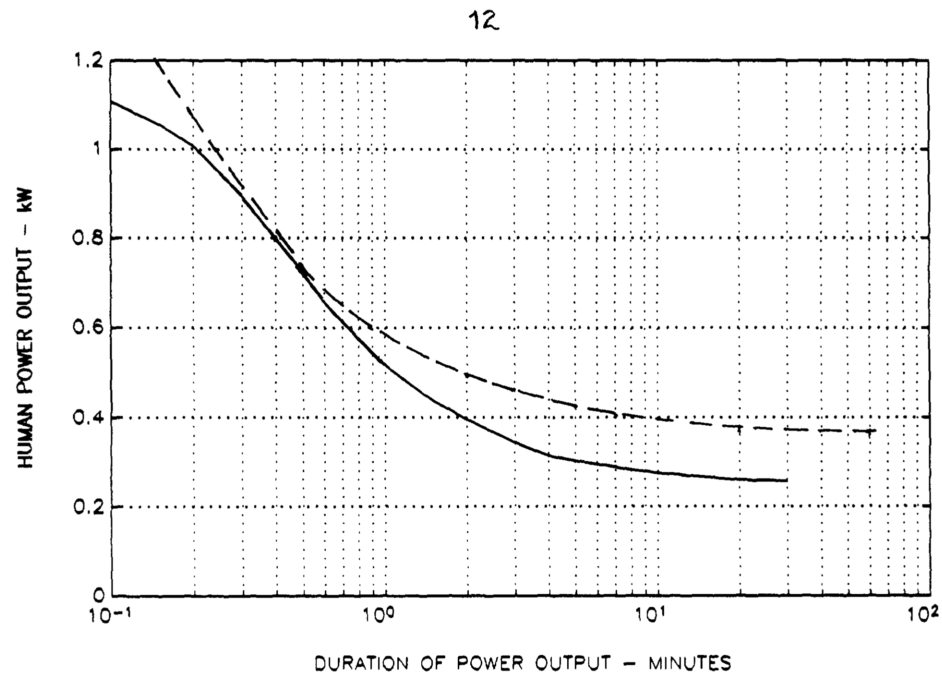


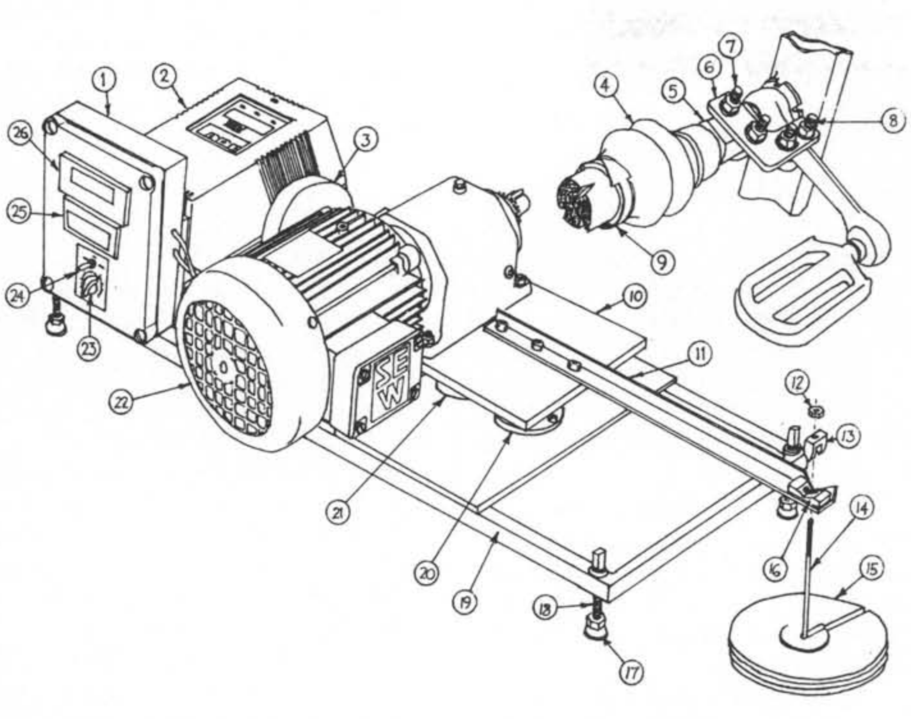
Figure 1 Human power output for various durations

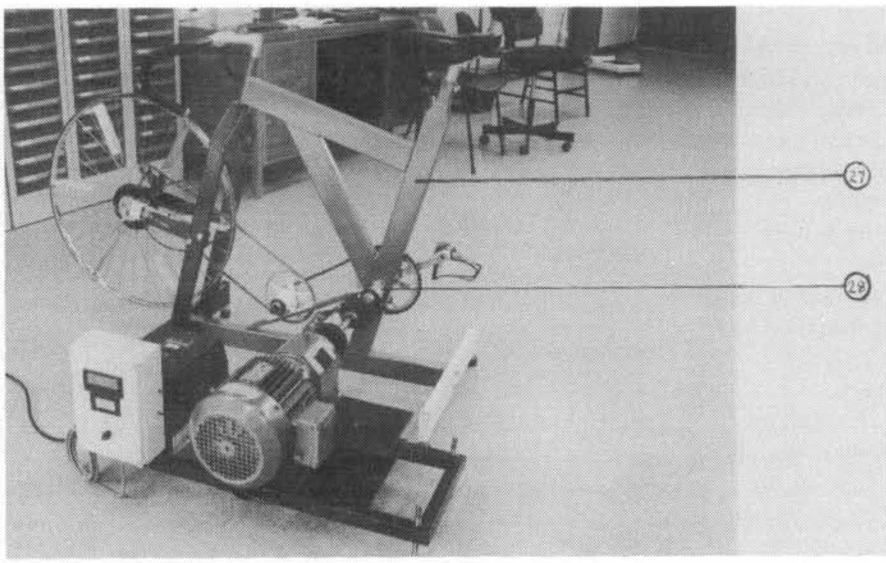
2. CONCEPTUAL DESIGN

2.1 Cycle ergometry - background

The cycle ergometer is one of the most commonly used devices in exercise testing and, depending on the level of instrumentation, can measure torque applied at the pedals, instantaneous human power output, or energy expended over a test programme by integration of power output with time. Cycle ergometers are available at relatively low cost and in some configurations offer precise and accurate control of the load (**1**). The present project focuses on an air-braked ergometer which allows steady speed measurement of the rider's power input. The calibration rig developed could, however, be used to calibrate an unbraked inertial flywheel ergometer such as described by Kyle and Mastropaolo (**2**)

The quality of the power meters on these machines is often unsatisfactory for laboratory research work. Lack of repeatability over time means that frequent calibration checks are required (**3**). Telford (**4**) claims that authors should describe the calibration system of their ergometers with the same care they describe the calibration procedures of the gas-analysis equipment. Cumming and Alexander (**5**) state that ergometers





Key to Figure 2:

1	Display box	15	Calibrating masses
2	AC-motor controller	16	Knife-edge support
3	Counterbalance mass	17	Rubber-bottomed foot
4	Flexible coupling	18	Stand
5	Shaft adaptor	19	Frame
6	Crank clamp	20	Load cell
7	U-bolt (shaft)	21	Viscous damper
8	U-bolt (crank)	22	Motor
9	Direction of shaft rotation	23	Speed-control dial
10	Motor supporting plate	24	Control switch
11	Calibration arm	25	Torque readout
12	Lock nut	26	Speed readout
13	Knife-edge device	27	Ergometer
14	Load hanger and tray	28	Input position at crank axle

Figure 2 Complete test rig

used for research purposes should be calibrated by dynamometer two or three times per year and it is noted they also refer to electronic ergometers where calibration appears no more reliable.

Few authors have reported calibration techniques used with air-braked cycle ergometers. The limitation on this type of ergometer is that the load torque can be varied for the rider only by changing gears, there being otherwise no means of varying torque at a constant speed as on band-, hydrokinetic- or eddy-current-brake ergometers. Telford (**4**) describes an air-braked Repco bicycle ergometer and states the subject's power output is proportional to the cube of the wheel rev/min. The calibration procedure for such ergometers is found in research reports as commissioned by the manufacturers.

A further aspect which must be considered is what exactly is being measured. Kyle and Caiozzo (**6**) report ergometry trials on a number of machines, and note that on a Monarch bicycle ergometer they tested unmeasured losses in the chain-drive train accounted for between 1.9% and 3.5% as ergometer power increased from 98 W to 295 W at 50 rev/min pedal cadence.

As it is the rider's input power that is of interest, the authors considered the only acceptable method of calibration was measurement of input power at the pedal crank shaft. In any event, the air-braked ergometer does not lend itself to dynamometry by measurement of band-brake tension as could be used to measure net power at the braked wheel on some machines.

An early decision related to the required power of the calibration system. The literature provides a good deal of data on human power output (for example references **6**, **7**, **8**, **9**). Kyle and Caiozzo (**6**) graph duration of human power output for tests on a number of average recreational cyclists ranging in age from 20 to 47 years old. The duration curve for an average of five cyclists is shown in Figure **1**. Whitt and Wilson (**9**) survey a number of sources of experimental data on duration of human power output. Their graph of NASA data for top athletes gives a good indication of maximum power output in cycling and is also plotted in Figure **1**.

It is clear from Figure **1** that the calibration system needed to be sized on short-duration output and able to provide

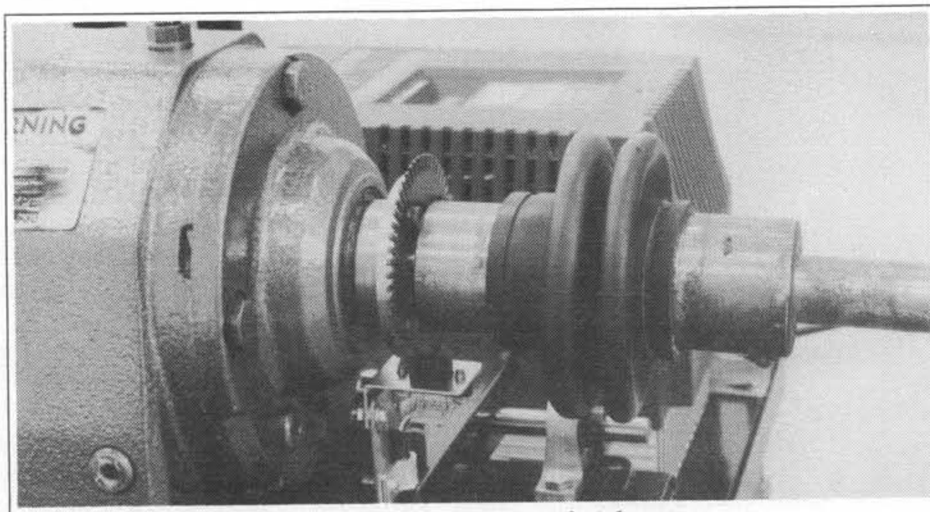


Figure 3 Toothed-wheel and optical-sensor speed pickup

a power of at least 1 kW for several seconds. Allowing some margin for peak power output for very short periods, and recognising that the calibration rig might need to settle for some time before readings were recorded, maximum power requirement was specified to be at least 1.2 kW for a minimum of 20 seconds. A speed range from 0 to 200 rev/min was also specified to cover all likely cadences of interest.

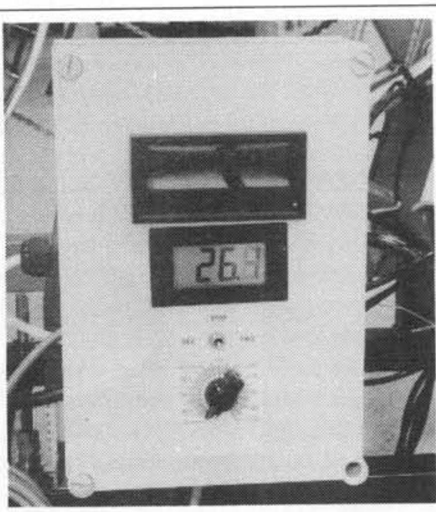


Figure 4 Electric-motor speed controller and torque-speed display

2.2 Calibration-rig design concept:

Ideally the ergometer would emulate the power input of a rider at the pedal crankshaft, providing the substantial variations in torque with crank angle as measured, for example, by Hull and Davis (10,11). The torque variations caused by time-varying human effort on the pedals would cause slight speed variations depending on system equivalent rotational inertia. Therefore, on an air-braked ergometer with power absorbed approximately as the cube of speed, the average human power input would be slightly higher at a given mean cadence, than from, say, an electric motor producing sensibly constant torque and rotational speed.

Simple calculations by Raine, based on a sinusoidal variation in crank speed (at twice pedal cadence due to maxima at each pedal 180° apart), indicated that a speed variation of as high as +/-5% (+/-10%) about the mean cadence would lead to an integrated energy consumption approximately 0.25% (1%) higher than at constant speed on a cubic power curve.

The high gear ratio speeding up the drive on the Repco ergometer greatly increases the effective rotational inertia, and it was calculated to exceed that on a typical friction brake ergometer with a braked steel wheel about 300 mm diameter. Wheel-speed variations about the mean with human power input were not measured on the Repco ergometer in this project, but subjectively felt small, as on a road-going bicycle. This may be the subject of a future study, with strain-gauged pedals, a high-resolution optical encoder fitted to the ergometer air-brake wheel, and computer integration of instantaneous rider input power.

In practice, effects of pedal crankshaft rotational speed variation will depend on current mean cadence, individual rider effort and prevailing equivalent inertia. These effects will be more marked at high effort in lower gears and quite variable. For practical calibration purposes it was judged most appropriate to use, as prime mover, the relatively uniform torque input from a geared, mains-powered electric motor with variable-speed control. This would be driven at a series of constant speeds. Later introduction of programmed torque versus time input, scaled to achieve the desired mean speed at the air-braked wheel, could be used to emulate rider pedal action more closely, increasing the versatility of the rig.

The choice of speed measurement was straightforward. Initially it had been hoped to use a frequency signal

APPENDIX
SPECIFICATIONS FOR DYNAMIC CALIBRATION RIG

Designed for use with a bicycle ergometer based on Repco Cycle Ergometer, Model No. HP 5209.

Power source: SEW - Eurodrive helical geared motor Type R43 DT90 L4, with Movitrac 1015-403-1 3-phase AC controller and remote station with digital display

Nominal and max. power output: 1.5 kW & 2.0 kW approx.

Output speed: continuously variable speed range from 3.5 to 200 rev/min.

Load cell: Bongshin strain-gauge based, type CBEK 100
Rated output (R.O.) 2mV/V ± 0.5%, equiv. to 200 Nm max.
Non-linearity 0.15% rated output (R.O.)
Hysteresis 0.15% R.O., Repeatability 0.1% R.O.

Drive shaft: 2x Fenner Universal Joints, Type C12

Displays: shaft speed or cadence (rev/min), torque (Nm), and frequency (Hz).

Overall dimensions:
length: 870 mm (1100 mm with load-cell calibration arm)
breadth : 480 mm without coupling
height: 400 mm
total_weight: 53.5 kg (including universal-jointed shaft)

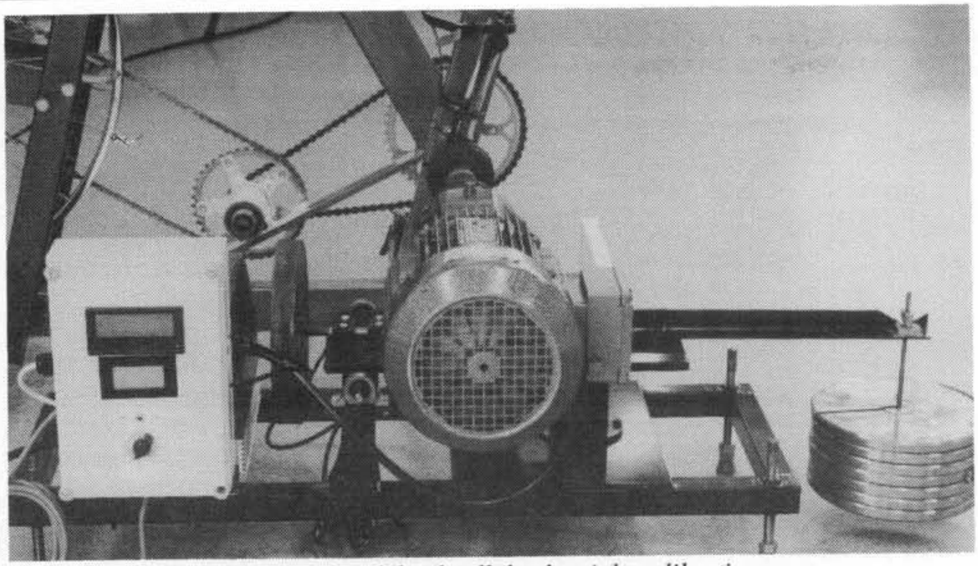


Figure 5 Calibration rig fitted with load-cell dead-weight calibration gear

from the a.c. motor controller, but slip in the induction motor ruled this out. A high-resolution optical transducer on the gearbox output shaft was therefore used.

Torque would have been most elegantly measured by using a foot-mounted drive motor and fitting a strain-gauge torque meter in the drive train from motor to pedal crank shaft. Alternatively, trunnion mounting of the motor and torque reaction through a load arm onto a load cell is another very acceptable solution, commonly used in electric dynamometers.

For reasons of cost both of these were ruled out. The solution finally chosen was to foot-mount the electric motor on a torque-reaction platform. This low-cost solution is quite satisfactory where ultimate accuracy is not required, and where very fast response of measurements to torque transients is not required of the load cell. The load-cell accuracy in the chosen system is ± 0.5% of full-scale deflection (FSD) and the overall torque-reading accuracy better than ± 0.6% FSD. This overall accuracy could easily be improved to ± 0.3% by fitting a higher-precision load cell. The higher inertia and typically lower stiffness of the torque-platform arrangement reduces the system response to higher-frequency torque oscillations, which above the bandwidth of the system response will not register on the load-cell signal.

To minimise alignment difficulties for the user, it was specified that the calibration rig should be mounted on height-adjustable feet and should transmit its power to the ergometer via a shaft able to accommodate parallel and angular misalignment.

3. SYSTEM EMBODIMENT DESIGN AND DEVELOPMENT

A view of the complete system is shown in Figure **2**. The following sections describe components of the rig.

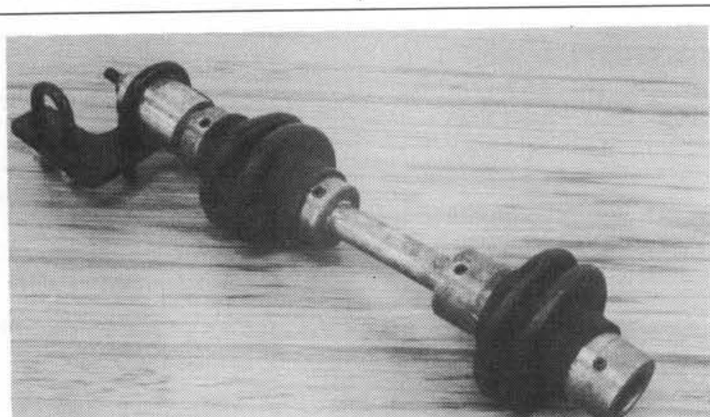


Figure 6 Universal-jointed shaft and pedal-crank attachment bracket

3.1 Electric-drive system

Initially it was decided that to keep the system compact an SEW Eurodrive helical geared calibration drive motor of 0.75 kW (1.2 kW for short duration) would be just acceptable, while not able to cater for brief superhuman efforts over a few seconds. A geared a.c. motor controlled by a variable-frequency drive connected to the 230 V mains supply was chosen for this task. With this arrangement the calibration drive output speed is continuously variable between 0 and 200 rev/min, with maximum power nominally available from 80 through to approximately 170 rev/min, covering the range of geared speeds on the cycle ergometer. The variable-frequency drive was to incorporate a digital speed display.

Subsequent development testing showed that the 0.75 kW single-phase system purchased was reluctant to deliver rated power, let alone higher powers for short durations. After some experimentation this system was exchanged for a 1.5-kW 380-415-volt 3-phase motor and inverter drive. This worked very satisfactorily and offered the added advantage of being able to meet a duty of 1.5 kW continuously, and higher powers for short term duty. The specification for salient parts of the complete system is given in Appendix 1.

3.2 Instrumentation and control

The signals from the Bongshin load cell, and the toothed-wheel-triggered optical speed pick-up (Figure **3**) pass to a signal-processing module which carries a control panel constructed to give four-digit displays of torque in Nm and speed in rev/min. This is mounted to the underframe adjacent to the variable-frequency motor-control module. This inverter-drive module also contains a digital display which indicates operating frequency (Hz) and in which of various push-button-selectable modes the controller is operating (Figure **4**). The controller enables the system to run in both forward and reverse rotations, the latter really useful only if the rig is used for some other speed-controlled rotary loading function.

3.3 Torque-reaction system

The 10-mm-thick motor mounting plate is pivoted from a shaft carried in sealed deep-groove ball bearings. At its outboard end the pivoted plate rests on

TABLE 1
INDICATED AND TRUE POWER vs PEDAL CADENCE
(atmoshperic pressure 757.45 mm Hg; temperature 20 C)

1st Gear (6.078:1) Cadence rev/min	Torque Nm	Power W
37.5	7.8	30.6
61.4	12.7	81.7
79.5	18.0	150.0
91.7	22.9	220.0
115.3	32.8	396.0
124.4	38.2	498.0
141.4	49.1	727.0
154.7	60.2	975.0
168.8	65.0	1150.0

2nd Gear (6.367:1) Cadence rev/min	Torque Nm	Power W
35.5	6.6	24.5
75.2	17.0	134.0
99.0	26.9	279.0
115.1	34.6	417.0
137.3	49.5	712.0
163.2	67.3	1150.0
181.7	87.2	1660.0

3rd Gear (6.686:1) Cadence rev/min	Torque Nm	Power W
42.3	8.2	36.3
77.7	20.6	168.0
100.1	33.3	349.0
137.6	59.5	857.0
153.4	72.3	1160.0
173.7	93.0	1690.0

4th Gear (7.038:1) Cadence rev/min	Torque Nm	Power W
37.0	8.0	31.0
69.5	18.4	134.0
88.4	28.6	265.0
101.2	36.9	391.0
118.7	49.0	609.0
142.2	71.8	1070.0
154.0	87.9	1420.0

5th Gear (7.866:1) Cadence rev/min	Torque Nm	Power W
19.3	5.6	11.3
44.3	12.7	58.9
68.3	21.0	150.2
87.9	32.7	301.0
99.4	42.8	446.0
114.9	57.5	692.0
127.8	75.6	1010.0
141.0	94.9	1400.0

6th Gear (10.286:1) Cadence rev/min	Torque Nm	Power W
21.7	8.5	19.3
47.8	25.7	129.0
64.6	44.0	298.0
80.5	70.2	592.0
92.3	92.1	890.0
100.3	104.7	1100.0
109.8	131.2	1510.0

the load button of the Bongshin strain-gauge load cell. An oil-filled dashpot is fitted between the motor mounting plate and the underframe to reduce any vibratory influence on the behaviour of the load cell. The load cell amplifier circuit also includes a low-pass filter to smooth the torque indication which is read at the instrument panel. All fabricated metal components are black painted or powder coated for corrosion protection.

To calibrate the torque-measuring load cell a counterbalanced calibration arm is attached to the underframe of the electric drive motor and loaded with dead weights to check span and linearity of the torque reading. This is shown rigged for calibration in Figure **5**. The weight hanger has a knife-edge support on the calibration arm.

3.4 Drive shaft and connection to ergometer

The geared motor output shaft is connected by means of rubber-booted Fenner universal joints and an intermediate shaft to the coupling to the centre of the pedal crank shaft. This coupling provides concentric alignment, while a torque bracket which attaches to one pedal crank transmits the drive torque (See Figure **6**). The drive-shaft overall length across the universal couplings is approximately 400 mm and is run close to truly aligned to minimise any forces internal to the drive train caused by offset of the universal joints.

It was later found that, while the calibration drive system experienced minimal vibration from misalignment, the ergometer itself tended to become unsteady and rock at high pedal cadences. To handle this some deadweights have been designed to anchor the back feet of the ergometer to the floor.

3.5 Underframe

The black-powder-coated hollow-section-steel framework sits on lockable threaded height adjusters which are used to align the driveshaft between calibration rig and the ergometer. The feet of these adjusters have rubber caps to offer quiet and secure standing on the floor.

3.6 Test procedure and data processing

After a warm-up period of, say 10 minutes, a typical test will involve driving the ergometer or exercise machine at a series of constant speeds throughout its speed range in each gear, and measuring the torque at each speed. From the measured data a calibration curve of power versus speed can be plotted for the given machine in its configuration as tested. This curve can be used to correct the indicated power on the machine's own power meter. This may be done by either

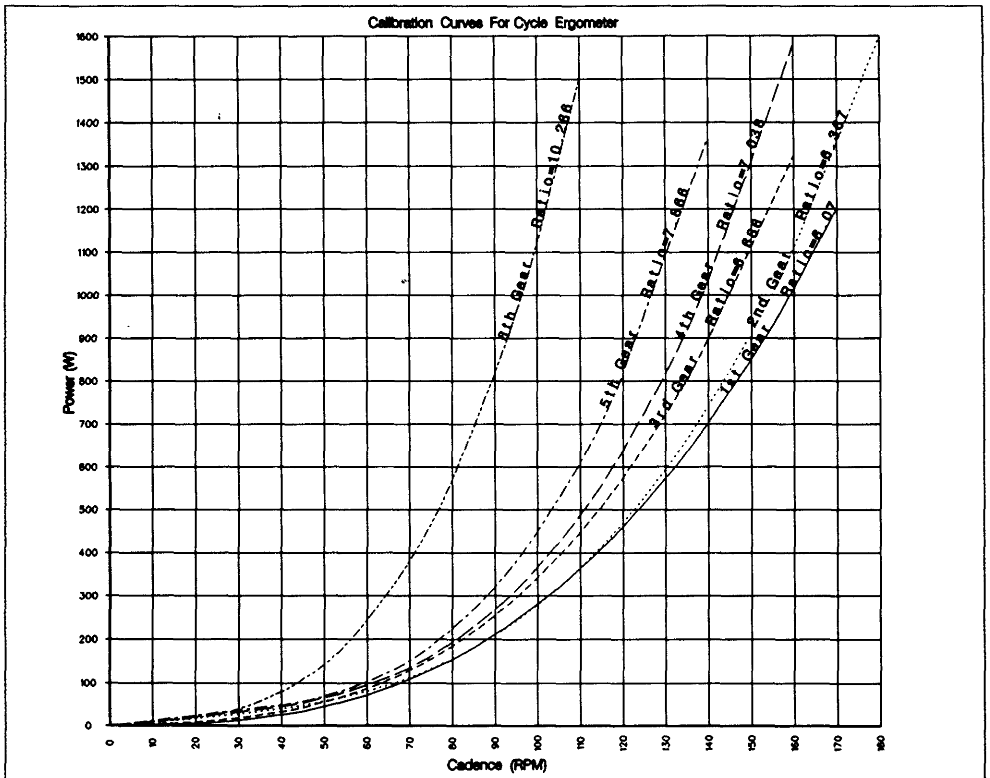
(i) using polynomial regression to fit curves through the calibration test points, so that the true power may then be calculated at each speed for each gear ratio; or

(ii) creating a look-up table which is used to read off power at any speed in any gear.

In both cases the calculation of power is easily done in a personal computer which may also be used for recording and processing data from test runs. The PC will also be used to log data of physiological importance such as atmospheric pressure, ambient temperature and humidity.

At any instant during a motoring test, power is calculated using the formula

$$\text{Power (W)} = \frac{\text{torque (Nm)} \times \text{speed(rev/min)}}{9.5493}$$



Atmospheric pressure 757.45 mm Hg; temperature 20 K

Figure 7 True power output vs pedal cadence

density affecting the aerodynamic drag of the machine, and temperature affecting transmission efficiency. It would be convenient to pre-determine a performance map of the type in figure 8 over a range of atmospheric temperatures and pressures, so that by interpolation an appropriate look-up table or set of regression formulae can be referred to under any given test conditions.

With the precision of the present load cell giving a torque read-out accuracy of +/-0.6%, and pedal-crankshaft speed being readable to +/-0.1 rev/min, overall system accuracy at a pedalling cadence of 100 rev/min is estimated to be +/-0.7%. This would improve to +/-0.4% if a high-resolution load cell with +/-0.25% accuracy were fitted. Four or five significant figures have been retained for the power calculation polynomial regression formulae in Table 2, but results of these calculations should always be rounded to recognise the limits on system accuracy. Calculated power figures in Table 1 have

4. ERGOMETER CALIBRATION TEST RESULTS

The system was first calibrated for torque and speed measurement, then run at various speeds against the Repco cycle ergometer in each gear to establish the corresponding power curves. Recorded indicated and true power data are shown in table 1, and true power is graphed against pedal cadence in figure 7. The aerodynamic drag of the Repco brake wheel does not fully load the motor at lower cadences, so there is drive capacity in hand for an ergometer with more demanding power characteristics. Polynomial regression analysis was carried out on the test data to produce a polynomial expression for the power curve in each gear. These are shown in table 2. Further tests will be carried out during working trials to experiment with the various power and control configurations of the electronic electric-motor drive.

The power relates strongly to N^3 at higher speeds, while showing some linear proportionality to N at low speeds. This is reasonable to expect, since transmission friction will be a significant component of the absorbed power at low speed.

Ambient-temperature and atmospheric-pressure variation will affect the performance of the ergometer, the air

TABLE 2
POLYNOMIAL-REGRESSION FORMULAE FOR DYNAMIC CALIBRATION CURVES IN THE VARIOUS GEARS

Calibration Curves for Cycle Ergometer
Least-Squares Regression has been applied

$P_1(N)=0.1851\times10^{-3}N^3+1.1188\times10^{-2}N^2-0.2013N+4.0798$
$0\le N\le 170RPM$

First Gear ratio = 6.078

$P_2(N)=0.3351\times10^{-3}N^3-1.8117\times10^{-2}N^2+1.2955N-4.0352$
$0\le N\le 180RPM$

Second Gear ratio = 6.367

$P_3(N)=0.3170\times10^{-3}N^3-0.1427\times10^{-2}N^2+0.4021N-0.6745$
$0\le N\le 170RPM$

Third Gear ratio = 6.686

$P_4(N)=0.5204\times10^{-3}N^3-3.1762\times10^{-2}N^2+1.6681N-3.3902$
$0\le N\le 160RPM$

Fourth Gear ratio = 7.038

$P_5(N)=0.7770\times10^{-3}N^3-5.5689\times10^{-2}N^2+2.3626N-6.3383$
$0\le N\le 140RPM$

Fifth Gear ratio = 7.866

$P_6(N)=1.2876\times10^{-3}N^3-2.6888\times10^{-2}N^2+1.0471N-1.2606$
$0\le N\le 110RPM$

Sixth Gear ratio = 10.286

therefore been rounded to three significant figures.

5. CONCLUSION

The ergometer calibration rig which has been developed should be a useful addition to the equipment available for sports-science research at the University of Canterbury, and is sufficiently portable to enable it to be easily transported to other laboratories in the country.

It is the intention of the Physical Education Department to mount a further project to develop a computer program with computation or look-up table to convert indicated readings to true power. Once the data-handling system is fully developed the rig should provide a versatile dynamic calibration facility. Ultimately, implementation of a programmed torque-versus-time input to the electric motor would provide a useful means of evaluating the effect of rider induced pedal crankshaft speed variations on the mean power consumption of bicycle ergometers.

The rig design and fully built-up systems are available to other persons, who should contact the first author.

6. ACKNOWLEDGEMENTS

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Letter

ACCUMULATOR DEBATE

This response is to the letter of William J. Moriarty (HP, vol. 11/1) commenting on the human-energy accumulator debate. Mr. Moriarty is able to convey a great many attitudes and assumptions with a minimum of words. His advice for resolving the issues, based on his authority as an engineer and a rule maker, is to do nothing until accumulators are built and have demonstrated their merit. The IHPVA would then, he confidently reassures us, revise its rules.

I assume that he would have given similar reassurances to Charles Mochet, who in 1932 built an aerodynamically superior recumbent bicycle that proved to be meritorious in the extreme, only to have it later declared a non-bicycle and placed indefinitely in its own irrelevant record category by the UCI. Mr. Mochet is still waiting patiently, beneath a Field of Dreams, for a rule revision. Of course, the present circumstances are different. The UCI did not originally prohibit recumbents, whereas the IHPVA already prohibits accumulators from its record categories, thus making it even more difficult to demonstrate merit in the first place.

Consider how ridiculous it would have been if the IHPVA had banned streamlined bodies from record competitions until after they proved their merit. The whole point of the IHPVA is to provide a context in which HPV innovations can prove their merit. Certainly Mr. Moriarty would not exclude new types of transmissions, brakes, or suspensions from IHPVA record competitions until after they had independently proved their merit. That would be absurd. But that is exactly what he recommends, since accumulators are new types of transmissions, brakes, and suspensions. So why does he treat accumulators differently? And why does he think his advice is neutral, even generous, when it is so clearly discriminatory?

Apparently, he still believes the "energy-storage myth": that a "real" or "pure" HPV does not store energy, so an accumulator is inherently less legitimate than other devices, and it therefore does not deserve equality of opportunity within the IHPVA. Obviously, I reject that myth, and I have explained my position in detail.

Continued on p. 13

THE LINK BETWEEN STABILITY AND PERFORMANCE

by
Andreas Fuchs

In the HPV-world, the basic law is "power output of rider equals the sum of all resistive forces multiplied by the speed and divided by the transmission efficiency". Since power output is limited and since the transmission efficiency is given by the state of the technical art, from this law one readily concludes that in order to maximize speed, the coefficient of drag Cd, the frontal area A of the vehicle, the mass m and the coefficient of rolling resistance Cr all have to be minimal. However, you may find, after having built your dream machine (for example with two wheels for minimum drag) while riding the vehicle into the first corner and while falling over - that there might be some hidden secrets... Of course there are: the "Power law" was written for zero angle of attack (the angle between the main axis of the vehicle and the relative wind, which in turn is the vector sum of the mirrored vehicle ground speed and the wind speed) and zero angle of slip (the angle between the direction of travel and the wheel main plane) and therefore does not explain everything.

If it is your main goal to build vehicles for maximum-speed attempts, then of course you do not mind corners and the power law in its form for in-line movement applies. But if you want to use a faired vehicle in velodrome races or for daily commuting, preferably that vehicle handles well and has good overall performance on a course with a lot of corners (even if it might not be the fastest on straights).

Cornering stability

The ratio of the aerodynamic forces to the weight of faired bikes is larger than that for cars because the bodies of the faired and much lighter bikes are more streamlined than those of cars. Therefore one could believe that it must be lift (perpendicular to the lifting surface, which in turn is vertically in the case of a symmetrical fairing) that disturbs the path of an HPV. But if a steady-state turn is analyzed, it is found that the vertical component of lift is usually small compared with the weight on either wheel and that the cornering velocity is reduced only slightly compared with the higher velocity that could be achieved without a fairing (in a lean, a component of the weight counteracts the centrifugal force as well as a component of the lift. In presence of lift, that is, with a fairing, in order that the forces balance, the lean angle has to be bigger than without lift. But the lean angle is limited due to the maximum adhesive friction of the wheels on the ground, and therefore the cornering velocity without a lift-producing fairing can be higher). Are there any other reasons for the sometimes sudden falling over of faired single-track HPVs? Yes... maybe the faired HPV experiences a stall similar to that of a stalling aircraft: the lift vanishes while the drag increases (due to flow separation).

Stalling aircraft have to dive to gain the speed that is needed to reestablish the airflow around the wings. Bikes cannot dive too much - otherwise they fall. Rough calculations show that increasing drag due to separation of the flow consumes a lot of kinetic energy. This then drops and the leaning vehicle can not return to the highest state of potential energy: the vertical position. In more technical words: to have predictable behaviour of your vehicle (essential if you ride it in dense traffic!) at angles of attack different from zero degrees, that is if the relative wind is transverse to the longitudinal axis of your vehicle, drag has to be minimal. Good handling is associated with the Cd being small over a broad range of angles of attack (and not only at zero degrees).

Roughly, from the point of view of the aerodynamicist, the existing velomobiles (European expression for HPVs) can be divided into two categories: the fin-like and the airship-like. The main difference between these two forms lies in the height-to-width ratio. This ratio is higher for fin-like designs than for airship-like designs. For this reason transverse airflow is less disturbed on airship-like forms (the Cd of a flat plate is much higher than that of a cylinder at 90 degrees to the airflow). If airship shapes are chosen for fairings, the increase of the drag with the angle of attack will be less severe than for fin shapes. Therefore, to fair practical vehicles, you will prefer round, airship-like forms over sharp, fin-like forms. For mere speed-record vehicles, both forms are feasible. If you are optimizing your minimal-drag-machine (with a given frontal area) and if you can decrease the height - do it.

Stability and performance

The fastest vehicle will be the one that suffers from the least accumulated drag along its path. Therefore it is first of all essential that the vehicle travels as straight as possible - to minimize the length of the path and thus the time needed to travel it - and second that the vehicle's wobbling is minimal because on a fairing the drag increases with the angle of attack and on wheels the rolling resistance increases with the slip angle. This is now the point where we have to

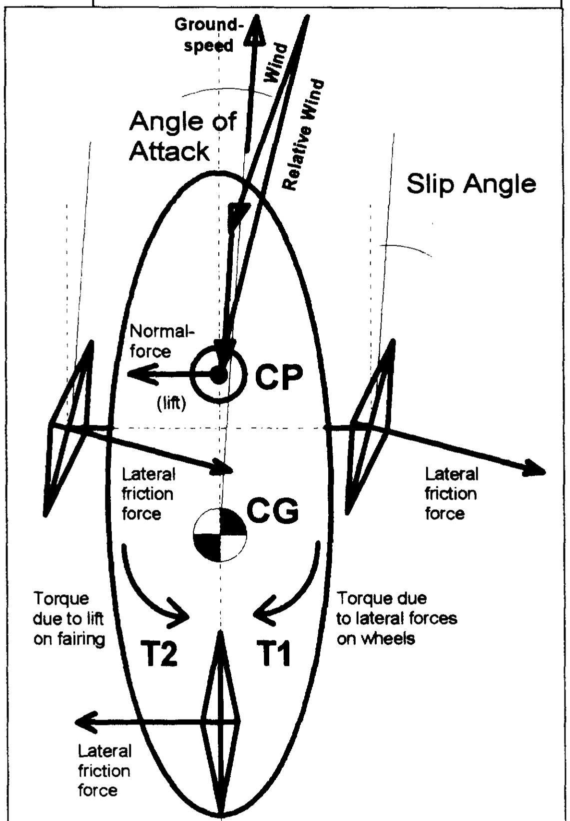

Figure 1 Typical three-wheeler: the lateral forces acting on the wheels and on the fairing. In the case shown the vehicle will return to its original path of movement (direction of the velocity) because the wheels produce a torque bigger than the torque due to lift (T1 > T2).

look at the stability of our vehicle, since the amount of stability seems to be important for its performance.

The wobbling of a vehicle is always around an axis through its center of gravity (CG) and depends on the inertia of the vehicle and the torques that are generated by the frictional forces on the wheels and the aerodynamic forces on the fairing (and to some amount also on the wheels). See figure 1.

As a reminder: torques are proportional to the forces and to the length of the levers on which these forces act. If it is known on which wheels what lateral (lateral: pointing-to-the-side) frictional forces and where on the fairing the pressure forces act, the torques may be found by summing these forces multiplied by the appropriate levers (distance of a point of attack of a force to the center of gravity). It is the same if from the distribution of each type of force we compute a total force acting on a single, respective lever (from the CG to the center of the distribution). In the case of the forces on the wheels, we call this total force "lateral frictional force" and for the aerodynamic forces we name it "normal force". The first one acts on the "center of friction" (CF) and the second one on the "center of pressure" (CP). Once all this is known, the total torque is defined and the reaction of the vehicle can be deduced.

Questions like "At this angle of attack and at this steering angle, will the vehicle return to its original path or not?" can then be answered.

Centers of friction and pressure

The CF (center of friction) and the CP (center of pressure) can be determined based on measured and tabulated data. Unfortunately, there are few data available about lateral forces produced by bicycle wheels. After all, they are not designed to take mainly lateral loads. But to guess the lateral force, one could say that at a slip angle of about ten degrees the lateral force equals the adhesive friction force. Above ten degrees, wheels will slip and then the lateral force is proportional to the slipping friction. The position of the CF depends on the locations of the axles and the numbers of wheels on them, as well as on the slip angles of the wheels. The slip angle on each wheel of a vehicle may be different, since some wheels are fixed relative to the main axes of the vehicle, some wheels are steered and if there are two wheels on the axle with the steered wheels, they have to move in such a way that the Ackermann-condition is fulfilled (Ref. 0).

More data exist about the location of the CPs of various forms, so we do not have to calculate them from pressure distributions. On faired and reclined recumbents, the CP is located at roughly one-third of the vehicle length behind the nose of the fairing ("Buelk's Rule", Ref. 1). The position of the CP is not fixed: at small angles of attack, up to say fifteen degrees, "Buelk's rule" tells you where it is. With increasing angle of attack, the CP moves backwards, until at 90 degrees it is near the geometrical center of the side view (near the middle of the fairing).

Figure 2 Andreas Fuchs with his Leitra

Stability considerations are different for single- and multi-track vehicles because bicycles move in a totally different way than do three- or four-wheelers: bikes have to lean in order to turn around a corner. Scientifically, this is called roll-yaw coupling. (Rolling is turning around the longitudinal, yawing is turning around the vertical, and pitching is turning around the lateral axis). Multitrack vehicles on the other hand, if unsprung, do not lean much in corners.

In this article, we do the stability considerations only for certain defined situations, fixed in time: we consider "static stability". There only the relative locations of the CG, the CF and the CP are of interest. Conversely, if we would consider also the accelerations and velocities, we would deal with "dynamic stability". This is a complicated task that is best tried on a fast computer and therefore this discussion is restricted to "static stability".

Yaw stability

For multitrack vehicles, the considerations are easy: if the center of pressure (CP) lies behind the center of gravity (CG), then the vehicle is aerodynamically stable (positive aerodynamic static stability). The vehicle works like a vane in the wind. If the CP lies in front of the CG, it is unstable (negative aerodynamic static stability), and if the CP and the CG are at the same position, we have "neutral aerodynamic static stability". A vehicle with positive aerodynamic static stability will keep its direction of travel even if the wheels lose contact with the ground. If one analyzes existing world-class racing machines, negative aerodynamic static stability is found in nearly all cases (Ref. 2). So these vehicles keep their direction of travel only because their wheels are in contact with the ground. And since wheels produce very high forces at a given angle of slip compared to lifting fairings at the same angle of attack, these vehicles can go straight even when the CP is far ahead of the front wheels and the CG.

What is said about the aerodynamic static stability of multitrack is also true for the single-track vehicles: if they are positively aerodynamically stable, they travel straight even without ground contact. But if the wheels are touching the ground, it is difficult to find out what happens due to the complicated manner in which bikes move, the so called roll-yaw coupling. Doug Milliken made experiments (Ref. 3) and found that if the CP is front of the CG, the bikes were easier to control (they were not aerodynamically stable, but they were easier to control) than if the CP was behind the CG. Matt Weaver used these findings to build a great racing velomobile, the Cutting Edge (Ref. 4). On difficult tracks with a lot of corners Cutting Edge outpaced Gold Rush, which for sure has proven to perform well on straights!

Is that all (?), you may ask. No! The concept of static stability allows us to compare vehicles and to find the relationships among the parameters and the handling characteristics. Ref. 2 shows how to calculate the parameters that determine stability or instability. Even if the simple means of calculating the position of the CP presented there are not absolutely precise, interesting insight can be reached when comparing different vehicles for which the behaviour in side winds or wind gusts is known from practice.

Roll stability

Here the most interesting question is whether or not it is possible to construct a faired single-track vehicle that handles well even in gusty crosswinds. From experience is known that multitrack vehicles are easy to handle over a wide range of wind speeds, but that in heavy wind faired bikes can be ridden only by experienced riders or sometimes even not at all. If your vehicle is to be a commuter, then you are interested in how much space on the street is needed in order to avoid collisions with cars (while oscillating from left to right and vice versa in case of a single-track vehicle). For multi-track vehicles, the width of the stripe needed is given by the width of the vehicle, because they do not oscillate much.

For bikes, the maximum width is the double of the sum of the amplitude of the front-wheel path and of the lean at the limit of adhesive friction and of the half width of the vehicle itself. Therefore it is found that under the worst windy conditions, a bike needs much more space on the street than a tricycle or a quadricycle. In a calm a bike requires less room, which usually is an advantage when overtaking cars in a total traffic standstill. The roll-oscillation of bikes is driven by gravity (the contact point of the front wheel is moved from one side of the main plane to the other side and gravity acting on the CG accelerates the bike from left to right and to the left again...). One could therefore consider a bike as being an inverted pendulum. Its oscillation-period then is proportional to the square root of the height of the center of gravity (Ref. 5 and 6) and this period is a measure for how long it takes for a bike - due to roll-yaw coupling - to change its direction of travel slightly.

In a gusty wind, a bike needs to change the direction of travel very often in order to stay on only one lane of the street. It is therefore concluded that the roll-oscillation period and hence the height of the CG need to be as small as possible for good handling in windy situations.

So the construction of a faired bike that handles well in gusty cross winds (in 99 percent of all situations) seems to be possible: its CG-height has to be as low as possible. Probably such bikes would even be quite safe in traffic. But when I ask people why they have not changed from normal bikes to recumbents, they most often say "Well, I feel better when I have a good overview about what happens on the street"...

Final remarks

Of course some statements made above need more proof before they could become accepted truth. For example the hypothesis that flow separation makes faired bicycles tip over should be checked in wind-tunnel experiments. Faired vehicles are often similar to forms that have been studied in wind tunnels, and that is why some statements above may be quite correct, but these vehicles and the tested forms are always not exactly identical. For precise statements we need more specific experimental data.

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Letter from Peter Sharp

Continued from p. 10

The burden is on Mr. Moriarty to justify why he is opposed to treating accumulators equally, within the constraints I have recommended.

Mr. Moriarty's advice is also backwards. Instead of using records as incentives for developing accumulators, he wants to use meritorious accumulators as incentives for changing the record rules. To make progress, one holds the carrot ahead of the donkey, not the donkey ahead of the carrot. The history of the IHPVA demonstrates that his advice has already had 20 years of no progress.

This is not an academic debate. If we believe that accumulators can improve HPVs, and that advanced HPVs are urgently needed as global short-distance alternatives to the automobile, then we must act. My understanding is that we may have that opportunity.

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BRIDLED ASSISTED HPVs, UNBRIDLED CHANCES

by
Peter Ernst

Ten years ago, when outsiders asked me why I got involved with HPVs, I became long-winded. Today, I simply answer:

"18 kJ for transporting 1 person over 1 km bicycle,
2,160 kJ for transporting 1 person over 1 km by car."

As the IHPVA prepares to celebrate its 20th anniversary, it seems fair to take stock of tangible HP-commuting results. The same picture prevails in most industrial countries. As soon as we leave the erudite circle of practical HPV adepts, what do we find but the roar of fossil-powered-displacement battles filling our streets? Hardly any HPVs are in regular use. Are they acquiring the same reputation as motorbikes, namely that of pure leisure machines? Is it sane to load our HPVs on cars before taking them for a spin?

We must not close our eyes to the possibility that present-day HPVs are not as practical, streetwise, as we would wish them to be. In most countries, tax laws nip the act of HP-commuting in the bud, since while car kilometers are tax deductible, muscular kilometers rarely are. Political innovations are long overdue, but perhaps our HPVs are unsuited for the challenges of traffic and climate. True, HP-record achievements have been outstanding. We must give high-speed protagonists credit for their help in breaking down ideological barriers. But surely, our daily salvation cannot lie in:

105 km/h = 65 mph human-powered-record vehicles,[1]
105 km/h = 65 mph wasteful, fossil-powered cars

but rather in machines evolving somewhere in between, capable of bringing the two into harmony.

How close to petrol - how close to people?

Why should the IHPVA shun assisted HPVs (AHPVs) of the **bridled** type? By "bridle" is meant "power-limited," so that motors are used solely as adjuncts to human power. Thus, they will be restrained from evolving into the behemoths on motorcycles. Bridling is discussed in more detail later in this article. In 1987, we accepted solar energy enough to keep records of genuine solar-car achievements. In 1993, board members were asked to debate the addition to our competition rules of:

- further energy forms derived from the environment
- human-powered energy accumulators

Today, our 200-m speed grapes are hanging so high that wordsmithing sophistry alone will allow specialists to climb the speed ladder further. It is time to look for the speedway exit named "Practicality." Beside huge expenditures needed for the next speed-record increment,[2] who can afford lawyers to ward off ensuing strife?

This is why I sympathize with the members who have asked our association to take AHPVs more seriously. There are immense energy savings dormant, provided the assist is somehow keyed to our legwork, for example, if:

- assist is geared so low that it can be used for climbing only (as demonstrated convincingly in 1991 by John Tetz[3])
- assist is spread, but available only beyond and in addition to human power input in accordance with the No Potatoes - No Dessert principle.
- etc., etc., (the reader's creativity is solicited)

Where could the auxiliary push come from?

Cold-pressed plant oils may be developed. Solar-produced hydrogen may be a distant goal. Yet, such substitutes are generations away. Since time is running out, we must make astute use of existing sources now. Electric drive is one option. Apart from liquid fuels, another readily available alternative seems to be natural gas in compressed form (CNH). It is clean and relatively cheap (billions worth are lost every day into the atmosphere). Micro-diesels could be rendered clean also, but their mass and bad starting habits militate against them. Stirling engines are too bulky and complex for our niche.

a) Battery-powered electrical assist

Based on my 1986 entry in the Swiss Tour de Sol of a battery-buffered solar HPV, I am inclined to delay giving electricity a serious chance, because of prevailing battery problems. Weight and deep-cycling characteristics in a tough climate are anathema to lightness and reliability.

Contrary to popular belief, hybrid bicycles with battery assist are not a new invention. In Europe, electrically assisted tandems appeared as early as 1895 in competitions.[4] Since then, dozens of commercially available electric bicycles have been launched, only to disappear again. Initial owner pride suffered because of tough pushing after battery failure, or the hard search for charging opportunities, or frequent battery-replacement costs. Interestingly enough, almost a hundred years later, the Japanese motorcycle giant Yamaha[5] opted for batteries to power its 1993 P.A.S. hybrid bicycle. It possibly did not have a small enough, de-toxed, 2-stroke engine ready for adaptation. Motor management may also have played a role in favor of electric drive. Anyhow, Yamaha plans to sell 10,000 P.A.S. units in 1994 and 100,000 in 1995 to its domestic market alone. Market feedback will tell whether the electric option meets commuter aspirations.

Basically, energy savings are possible, thanks to high-efficiency motors and electronic controls. However, even for modest operating ranges (say 15 - 30 km = 10 - 20 miles per charge), the cumulative mass of motor + control box + battery + charging unit is critical in an ultralight design. Once the long-awaited jump in accumulator technology materializes, electric drive will spread.

b) Hydrocarbons burned in I.C. micro-engines

Here, small 2-stroke engines, as conceived for portable implements, may be used. They are of simple design and have a good power-to-weight ratio. However, they do pollute and small engine makers are aware the California Air Resources Board (ARB), might extend stiff limits[6] on their products. Control of the most minute fuel injection-quantities and -times will decide whether these make the grade.

FMS of Switzerland[7] has developed 2-stroke motor management, typically applicable to spark-ignited micro-engines between 20 - 80 c.c. Their metering device, i.e., injectors plus catalytic converters, represent a logic system with soft and hardware, capable of de-toxing most two-strokers. Since metering and fine-dispersing require high pressures, it would seem advantageous to start with pressurized energy in the first place. Thus, the injection part becomes simpler, lighter, and cheaper.

Another Swiss invention, described in the rocking-piston patent,[8] proposes further savings in weight through reduction of moving parts. As shown in figure 1, the mushroom-shaped piston is of a piece with the connecting rod. Hence, its cylinder liner is waisted in the middle. The inventor reckons that it should soon be possible to build high performance, ultra-light 10 c.c. assist engines, weighing (tank inclusive) around 1.5 kg (3.3 lb).

It is instructive to note that worldwide, large sums are invested to lift two-strokers into the league of compact, light, but clean-running engines, also suitable for hybrid powered cars, using I.C. engines for charging batteries.

Assist-source comparisons

If one cares to look back, a rough trend emerges. As can be inferred from table 1 and figure 2, assisted two-wheelers seem to converge toward a total vehicle mass of:

approx. 30 kg (66 lb) for battery-assisted bicycles

possibly 10 kg (22 lb) for fuel-assisted bicycles

The latter would guarantee handiness in everyday use and ease of parking and storage. Such solutions imply high-tech materials, which are acceptable if the overall energy balance is backed by longevity and recyclability.

Most of the listed examples do not exceed 500-watt nominal assist power, with correspondingly modest speed and insurance levels: hence, no driver's licenses are required! The next higher class are heavy models, mobylettes, and mopeds, whose tiny pedals are never-used adornments, i.e., they are **not**

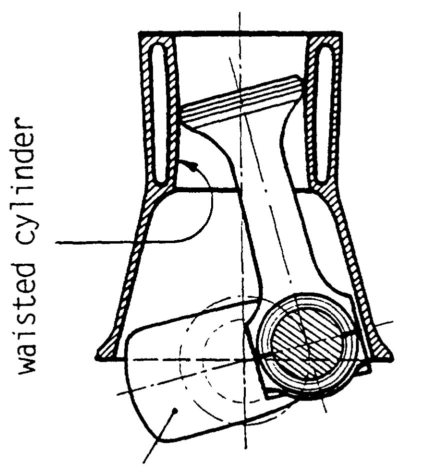

Figure 1 The rocking piston, the basis of several Salzmann patents

Table 1 ONE CENTURY OF HARDWARE DIETING

a) battery assisted cycles	kg	b) fuel assisted cycles	kg
1895 DUTRIEU stayer tandem (B)	97	1899 WERNER cyclomoteur (F)	51
1961 GARELLI elettro bici.(I)	39	1947 HONDA 49 cc mobike (J)	32
1976 HERCULES el. Zweirad (D)	35	1950 LOHMANN Diesel Zweir(D)	29
1982 SINCLAIR C-5 el.trike(GB)	45	1956 PARILLA Parillino bi(I)	27
1983 PANDORA el. bicycle (GB)	38	1964 GARELLI Mosquito bic(I)	24
1984 GEIGER Elfa el.Zweir.(CH)	33	1967 BSA Ariel-3 trike (GB)	32
1988 BINDER-Matic el.Zweir.(D)	30	1976 MX-3 SUPERBIKE bicyc(K)	19
1991 VELOCITY-Cann.el.MTB (CH)	28	1984 F+S Saxonette KATzwr(D)	26
1993 YAMAHA P.A.S.bicycle (J)	31	1991 TETZ Lightning HPV (USA)	15
		1995 SALZMANN 10 cc Velo(CH)	12

AHPVs, since constant use of the 900 (legal) watt = 1.2 hp limit is made.

I am very concerned about the psychological profile of the average mobility consumer, perhaps comparable to that of a TV-watcher behind a bag of salted peanuts. Hard as it may sound, I am **for** compulsory bridling devices, especially for the young two-wheel fans in their formative years.

Free will, versus tutelage

When I earlier suggested that leg power be augmented, I was met with opposition, "Do not patronize mature citizens!" Yet, appeals to common sense are a dead end. In traffic, man copies electricity and will always follow the path of least resistance. Hence, a "bridle or bust" tamperproof link is perhaps not so odd. Have no fear, such links do not have to be complicated. Fuzzy logic might, for example, probe the tension on the drive chain via a spring-loaded idler sprocket, which acts on engine output (throttle control or chopper amplitude).[1]

Clearly, if our AHPVs grant human power its right of way, we shall enjoy:

- better health, via cardiovascular exercise
- lower vehicle mass (rider's too, ultimately)

Indeed, hyperlight vehicles are possible, if one limits fairings, etc., as shown in figure 3. Climate control is warranted via enforced hp-metabolism. Semi-fairings also meet the exigencies of hot climates, where ample air exchange and a modicum of roof shelter are a must. It is unrealistic to think of adding active air-conditioning devices to closed vehicles intended to be moved chiefly by our own anatomy.

First production of "bridle" control, 1993

Apart from their choice of assist energy, Yamaha must be given due credit for the world's first production APHV, having a bridled assist control, without a 'hand gas' grip. In 1993 a thousand units were built to conduct national market tests. Typically, an electronic motor management senses the load of the pedals, plus the vehicle speed. If there is no human base load, there is no assist. If there is, the control unit releases auxiliary push "in proportion to pedal load" up to 15 km/h (4.2 m/s). Beyond that, up to 24 km/h (6.7 m/s) the released push will be decreased gradually.

Figure 4 confirms that the assist is about equal to the "do-it-yourself" contribution. The electric motor is of a DC-brush type and has a nominal output of 235 watts. That means that, for climbing hills, combined tractive power is about 2 x 235 = 470 watts.

Figure 5 shows that the assist is wisely meted out when most needed, namely during ascents, rides into the wind, or for load-carrying. Beyond 25 km/h (16 mph = 7 m/s), the assist quits. The brightest fitness news in Yamaha's documentation reads:

"The bike never runs on motor power alone..."

What a hope-inspiring acknowledgment by one of the world's leading mobility providers, in stark contrast to hyper-mobility exhortations made by giant car makers.[9] Yamaha reckons with a 20 km (12 m) operating range per battery charge.

The shape of future APHVs

Knowledge gained during 20 years of practical HPV-building might (eco-) logically also recognize meritorious efforts such as Yamaha's P.A.S. upright.

Not speed, but user comfort, energy requirements, and environmental effects will thereby improve. To stay legally within the bicycle category, maximum assist speed might be pegged at 25 - 30 km/h (16 -19 mph). National legislation

[1] Such a device was patented by Henri Rosen of Boston for his "Goped" AHPV - *editor*

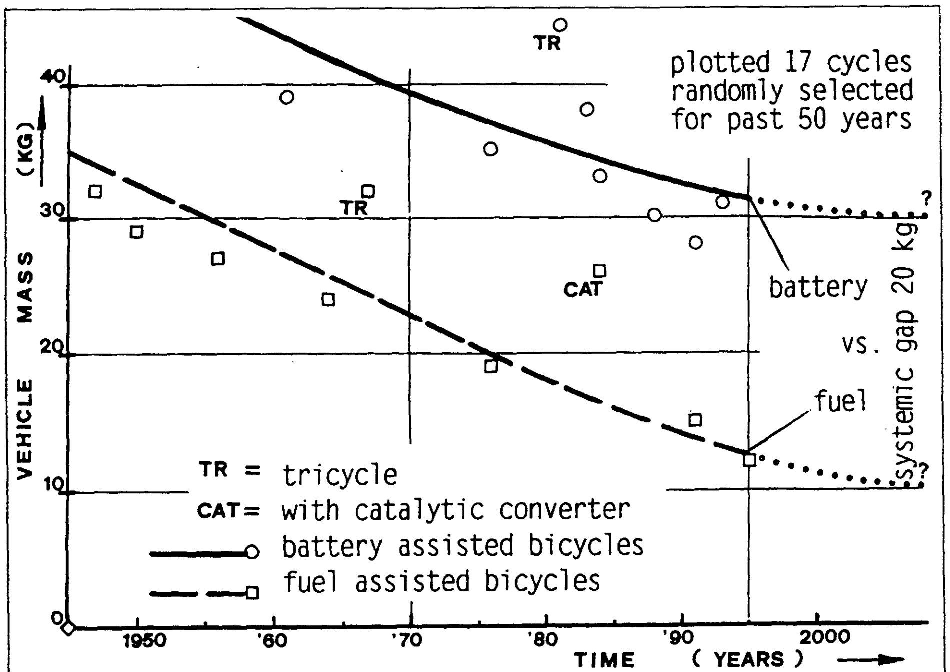

Figure 2 Trends of decreasing vehicle mass

might have to be adapted to allow low-power assist (on both two- and three-wheelers) to take advantage of cycle paths.

Since APHVs purport to replace commuting cars, they should also make second (regular) bicycles superfluous. The assist might come as a removable power pack to reconvert APHVs quickly to HPVs for flat flings. With an electric drive, this is difficult. This would be quite possible though with fuel assist, if the tank is pack-integrated and smaller than 500 c.c. (½ liter = 1 pint).

Aren't grandmas and grandpas potential AHPV users? They are little inclined to recharge batteries, nor to clip on/take off engines. For them, evolution is spelled:

OFTEN = daily/nightly, electric recharge...
SOMETIMES = weekly/monthly, filling of tank...
NEVER EVER = yearly replenishing by dealer, during maintenance, security check

Indeed, synergetics occur if the tank is a frame-member, as in fig. 3. Recumbents can easily take a central tube of 2m (6'6") length by 72mm (2.84") diameter, to hold some 8 liters (about 2 US gal = 1-¾ imp. gallons) lasting for 1 year, i.e., for about 8000 km, or 5000 miles, based on conservatively estimated 1000 km/liter, or 2350 mpg (US), or 2820 mpg (imp.), thanks to a modest assist portion of 10% only!

If CNG is used, the equivalent energy may be stored under pressure of a few dozen bars in a filament-wound tube, refillable yearly by bike-shop personnel (AHPV users won't buy via dept. store/mail order). A further CNG spin-off is an appreciable simplification of the injection metering system (because it is pressure-fed) while emission standards are more easily reached. Such AHPVs will be energy misers. Displacements of 10 - 20 c.c. will suffice to deliver the occasional peak approaching 440 watt, or 0.6 hp. De-toxing cures by FMS,[7] applied to German 2-stroke chainsaw engines, confirm that typical consumptions of 260 - 300 g/kwh are possible with innovative motor management. The first figure would correspond to a high-tech metering system, the latter to a cheaper injection version, both coping with growing environmental demands.

I visualize the ideal AHPV of the year 2000 to be a 1+1 seat three-wheeler having the banking characteristics of a bicycle. Hence, the two independently sprung rear wheels must provide suspension **and** roll freedom, i.e., I imagine the front wheel to be steered and driven. My 2000 x 750 mm (6'6" x 2'6") vehicle would be only scantily faired by an aerodynamic, close-fitting "45° apron." Shopping loads, up to two beverage crates, would be accommodated. The bottom bracket is to be adjustable, while the seat will be the part of load-bearing structure, with a child-seat attachable to its back-face, etc., etc. ...

In short, if advanced AHPVs cannot be bought soon at the friendly cycle shop around the corner, IHPVA members must heed Mike Burrows'[10] advice, "Good machines need building, not talking." To get into gear, are we passively waiting for Far-Eastern mailers to offer cheap, de-toxed, two-strokers, or possibly even do-it-yourself AHPV-building kits?

We can do better. As a first step, I invite those HP-readers with a sharp sense of things to come to write in and suggest a typical AHPV testing cycle and -circuit, with length, profile, speeds, etc. Indeed, only exact consumption and emission figures will allow us to compare energy balances of emerging AHPVs. Hardware suggestions are equally welcome in our discussion. Let's push-start them!

This does not betray our credo, but will draw the IHPVA away from protected circuits and will bring it closer to everyday road users. Without support of the latter, we and our eocsphere are bound to back the oars.

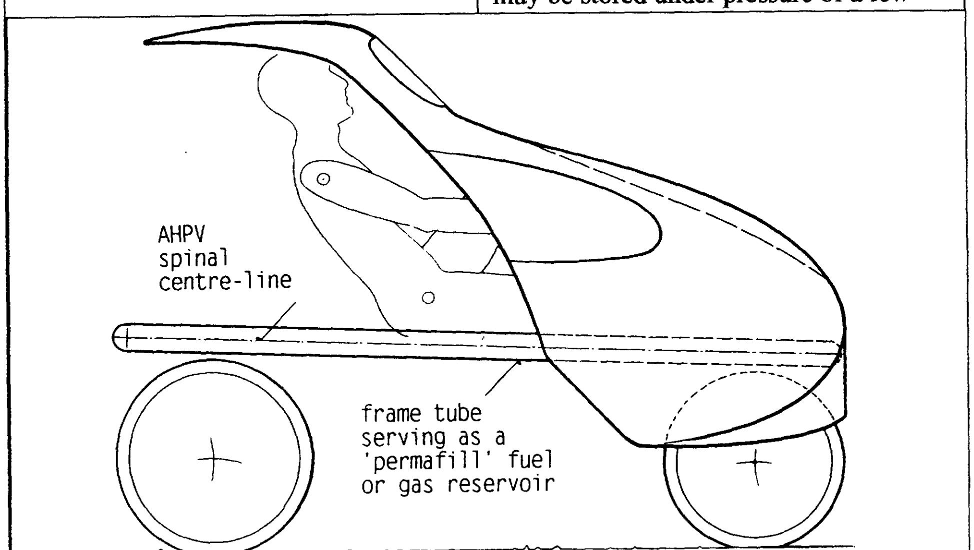

Figure 3 Partial weather protection - e.g. semi-fairing "45° apron"

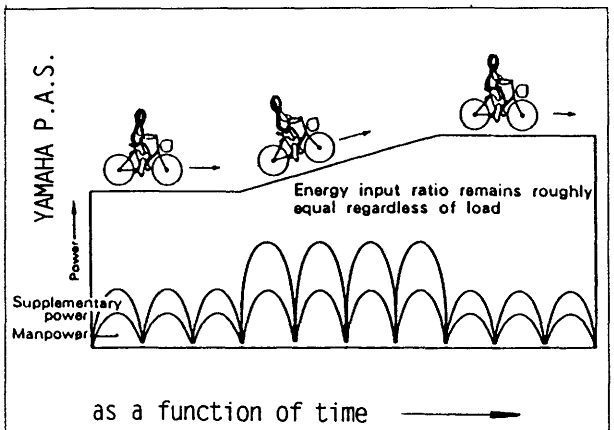

Figure 4 Power ratio (assisted/human power) as a function of time.

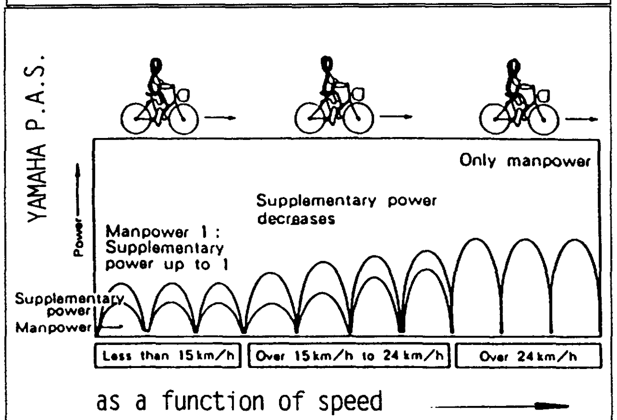

Figure 5 Power ratio (assisted/human power) as a function of speed

Peter Ernst
Alex. Moser Str. 15
CH-2503 Biel-Bienne
Switzerland

References

(1)du Pont de Nemours prize of $18,000 US, won May 11, 1986 by Fred Markham (Easy Racer team of Gardner Martin, USA) at 65.48 mph = 105.36 km/h = 29/27 m/s over 200 m

(2) At least 5% faster than current world speed record of 68.73 = 110.59 km/h = 30.72 m/s, held by Chris Huber (Cheetah team, USA), dated Sept. 22, 1992

(3) John Tetz, IHPVA member
7B Mark Lane
Succasunna, NJ 07876 USA
tel. (201) 584-6481
describing his experience with a converted Lightning recumbent HPV + 21 c.c. McCulloch engine in HPV News issue Nr 8/2 1991 and 8/5 1991 (also see his paper in the Fourth IHPVA Scientific Symposium).

(4) Helena and Eugene Dutrieu of Belgium. Successful world-class pacer racers of the last century (soon realizing that it is deadly to perform behind fuming pacemaking machines). Source: W. Gronen, Geschichte des Rad Sportes, Edit. Fuch Druck Verlag, Hausham, Germany

(5) Yamaha Motor Co., Ltd.
2500 Shingai, Iwata
Shizuika 438 Japan
tel. 81-538-32-1145
fax 81-538-37-4250

(6) Air-pollution laws by the State of California, handled by the Californian Air Resources Board (ARB)
P.O. Box 2815
1102 Q Street
Sacramento, CA 95812
USA
tel. (916) 445-4383
ATSS 8-485-4383
in particular, title 13, relating to miscellaneous engines

(7)Inventor + patent holder: B. Frey, Ing. ETH, partner in:
Fuel Management Systems, FMS Fuel Management Systems, Inc.
Eschenhof
529 N. Morris Ave.Mundelein, IL 60060 US tel. (708) 566-8820
CH-4512 Bellach
Switzerland
tel. 65-38-1797, tel. 77-31-6037
fax 65-38-3663

(8) Rocking piston and further I.C. engine patents by:
W.E. Salzmann
Dipl. Eng. ETH, MSIA, MSAE
Schoenegg 7, CH-6300 Zug
Switzerland, tel. 42-22-2235

(9) Mercedes-Benz board member Dr. Rudolf Hörnig, in Der Spiegel, weekly magazine, Hamburg, Nr 25, 1987, "We can hardly be expected to make our cars artificially slower."

(10)Mike Burrows, hon. chairman + press sec., British Human Power Club
Burrows Engineering
16 Thunder Lane, Thorpe, Norfolk, England; tels. 0603-72-1357 (wk), 0603-3-2142 (hm)
British activist, prolific designer, builder, rider of: Windcheetahs/other HPVs/UCI-approved competition machines.

Peter Ernst is a mechanical engineer, a past IHPVA board member, co-founder and first president of Future-Bike (Switzerland); team chief and designer of the world's only three-wheeled lightweight solar-powered grand piano competing in the Tour de Sol, now in the Christchurch, UK, museum for three-wheelers.

Review
1993 HPV SYMPOSIUM
Technical University of Eindhoven

It is always humbling to realise that whereas I find the production of Human Power in my own language an enormous task, my fellow editors in Europe and many other places in the world publish dual proceedings, one in the local language and one in English. Thus I am happy to be able to review the preliminary proceedings in English of the HPV Symposium held at the Technical University of Eindhoven on March 18, 1993 (W.S.V. Simon Stevin, Eindhoven, The Netherlands).

The proceedings consist of some short introductory pieces, three main papers, and a catalog of HPVs available or on display. One introductory piece described how three mechanical-engineering students tried out a FWD Flevobike while on vacation and subsequently campaigned to have the construction of Flevobikes as the principal activity of a new course. Thereby has enthusiasm for HPVs grown at Eindhoven.

The first of the main papers is a repeat of a 1978 publication by Chet Kyle: "Predicting HPV performance using ergometry and aerodynamic-drag measurements". It has a footnote indicating that Chet would bring it up to date in his presentation. I presume that the updated version will be seen in the final proceedings. In any event, it is full of useful data and correlations.

"Muscle involvement in cycling: can we get more efficient?" is the title of the second paper, by Gerrit Jan van Ingen Schenau, who is on the faculty of the university of Amsterdam. This is a paper that starts with the fundamentals of musle energy supply and action and deduces that the peak metabolic efficiency can get to around 30%. Bicycle pedaling produces 23-24% which is therefore, states the author, close to the theoretical maximum. Therefore we cannot expect to do much better.

Continued on p. 21

A FOLDING MWB TWO-WHEELED RECUMBENT

by

Nick Abercrombie Andrews

A two-wheeler recumbent need not be a "long tame monster", nor ride like a roller skate down a rutted road. It is possible to build a two-wheeler that is light and responsive, comfortable for touring and with enough foldability to make it easily portable. Putting these features together in my recumbent has required time (several years of part-time effort to date), a lot of brazing rod (I have built several earlier models in my cellar workshop), and a good supply of secondhand tubing (from a helpful local cycle dealer).

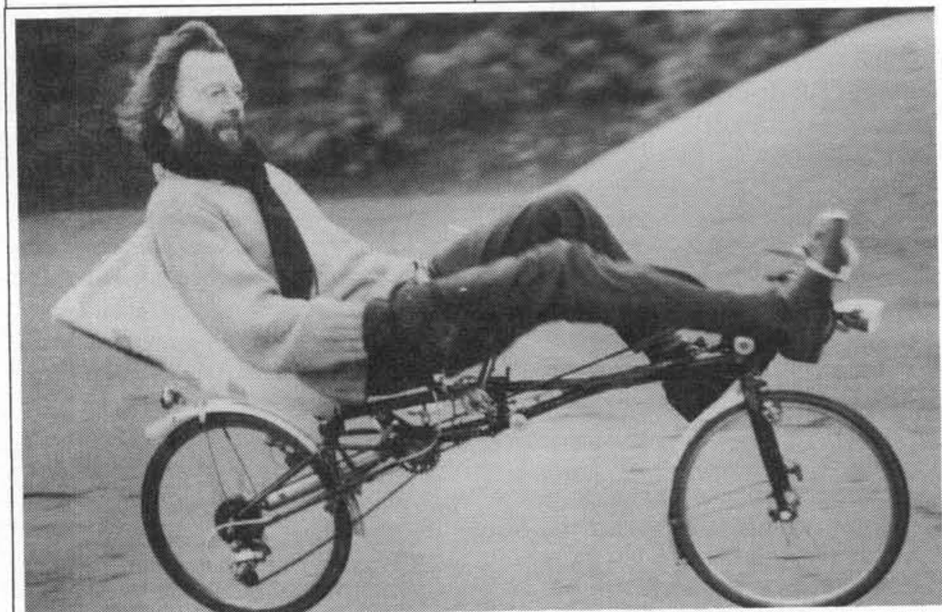

Figure 1 Nick Andrews on his foldable medium-wheelbase (MWB) recumbent

My first recumbent was a rigid short-wheelbase variant with conventional handlebar steering, which delivered a lot of road shock. To reduce this in my next bike I added suspension between the seat and the back wheel, and moved the front wheel forward. I wanted a comfortable and compact bike for touring and commuting, so I settled for a layout which put the crank axis slightly ahead of the head tube, and used small wheels front and rear.

I was very pleased with this non-folding bike, but with its cable steering and single main tube, it was asking to have a hinge added in the middle.

Making a bike which I could easily take on public transport was one of the main aims. The most important consideration in designing the hinge was not to compromise the bike's performance or safety by making it less rigid, and not to add unnecessary weight. So rather than cutting the bike in half and adding a hinge I built a new bike which in most respects was similar to its non-folding predecessor.

Steering

I wanted to avoid a handlebar arrangement that would impose any sideways bending requirement on my knees. My first idea for the MWB non-folder had been to pivot the handlebars separately inside the transverse front seat supports, and use two link rods rather than the one used with conventional under-seat steering, which to me never looks comfortable. My arrangement brings the handlebars
up at a natural position beside the body, and needs a linkage which, as normal, moves the front wheel to the left when the right handlebar goes forward, and vice versa. This can be achieved by offsetting the seat supports, crossing over the handlebar tubes, and attaching the linkage below the handlebar axis.

Instead of using solid link rods, a conventional head-tube angle allows the same motion to be obtained by a cable and four pulleys. This is the system used in the folding recumbent.

Cable steering is sometimes said to feel spongy or imprecise, but I have not found this. I think a lot depends on the diameter of the pulleys; if they are too small, not only is more stress put on the cable, but any resulting movement is magnified. Reasonable-sized pulleys are also needed to prevent cable failure. A figure mentioned by a lift engineer is a ratio between pulley and cable diameters of at least 25 or 30 to 1. The 49-strand stainless-steel cable used is 2mm diameter, high-tensile but very flexible. I used 63.5mm (2-1/2") diameter handlebar pulleys, and a 76mm (3") steering pulley, which together feel fine. A fairly strong

Figure 2 Unfolding commences

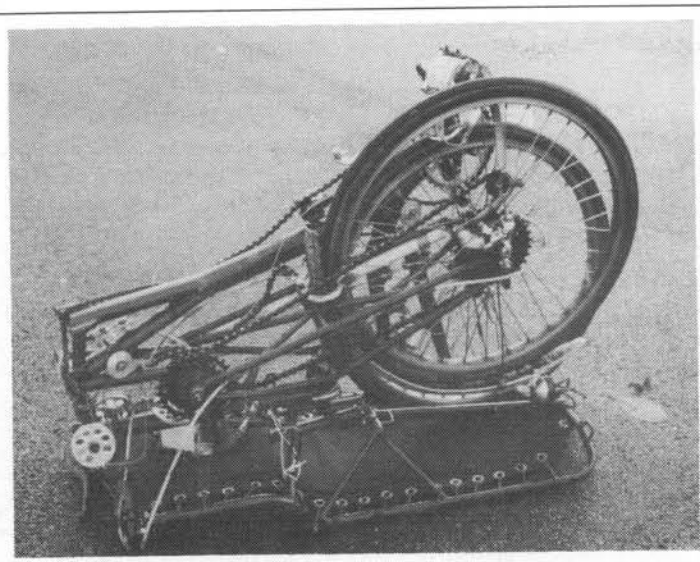

Figure 3 Fully folded

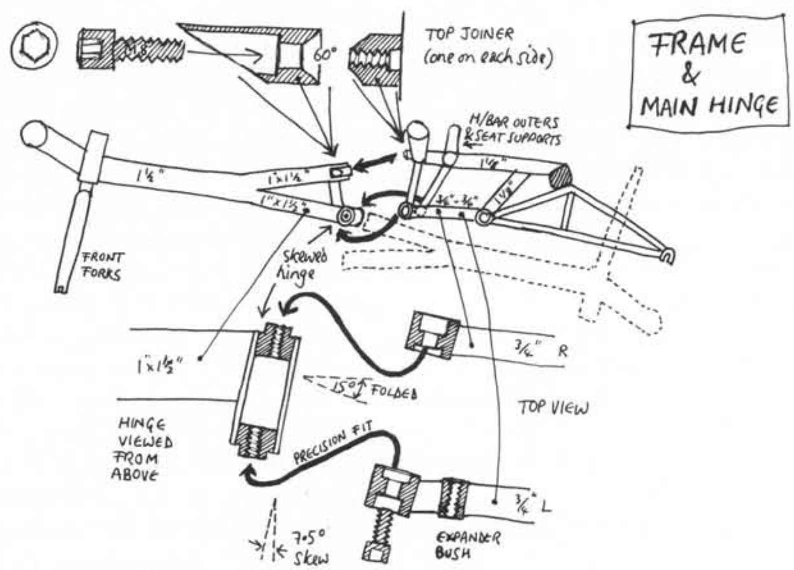

Figure 4 Sketches of frame and main hinge

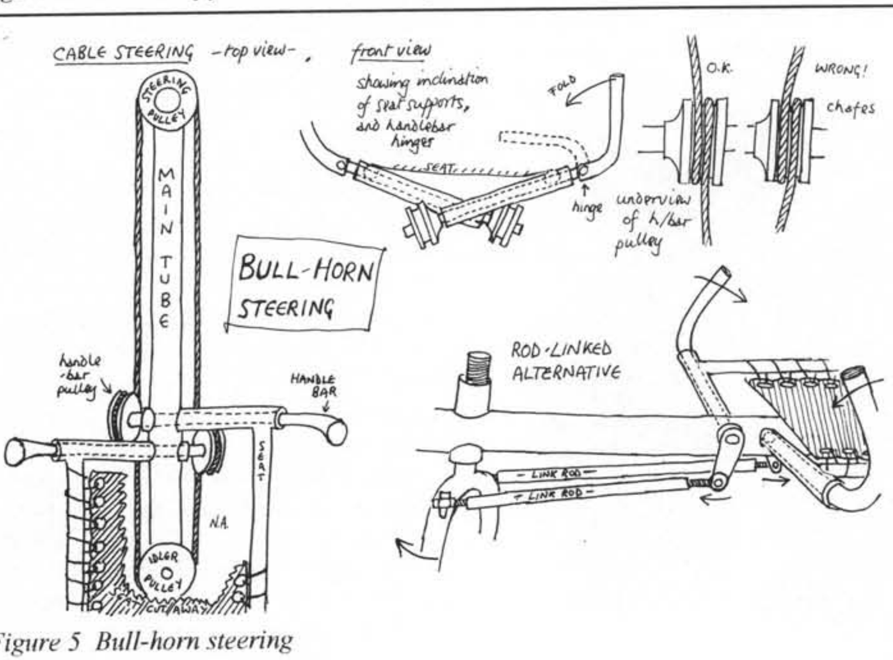

Figure 5 Bull-horn steering

tension is needed in the cable, and consequently a sturdy construction with decent bearings is required.

An in-line cable tensioner tended to bob about alarmingly over bumps, so the folding bike uses an adjuster on the front pulley, with a quick-release lever to avoid having to re-tension the cable each time the bike is unfolded.

The unconventional handlebar geometry makes riding tricky initially even for experienced recumbent riders, since the handlebars feel increasingly strange as they are moved further away from the central position. Sometimes very confident riders begin well, but then fall off when they try to turn. Most people soon relax and enjoy riding, but a few have had a long wrestling match with the machine and still been unable to stay on.

The steering arrangement has an advantage I did not anticipate; the bike balances around the handlebars and the rider can stand up to lift the bike very easily. This is a useful way of turning on the spot or negotiating a kerb, which to a large extent overcomes the occasional awkwardness of the recumbent position.

I have recently been told that a recumbent motorbike was once manufactured with similar handlebars. When the speed fell to about walking pace, two stabilizer wheels automatically dropped down, so the rider did not have the difficult task of supporting the machine's weight with his outstretched legs.

How short?

Most recumbent two-wheelers place the seat just in front of a large rear wheel, but by using a small wheel I had the option of sitting directly above it. For general use it feels about right to have one's head directly above the rear axle. The bike can negotiate BMX obstacles without doing a backward flip. Travelling downhill and/or braking tends to put more weight on the front wheel, as does carrying luggage between the wheels, and the bike always feels stable. I have not noticed any disadvantage from the unequal weight distribution apart from the inability to hold the bike stationary on steep uphills using just the front brake.

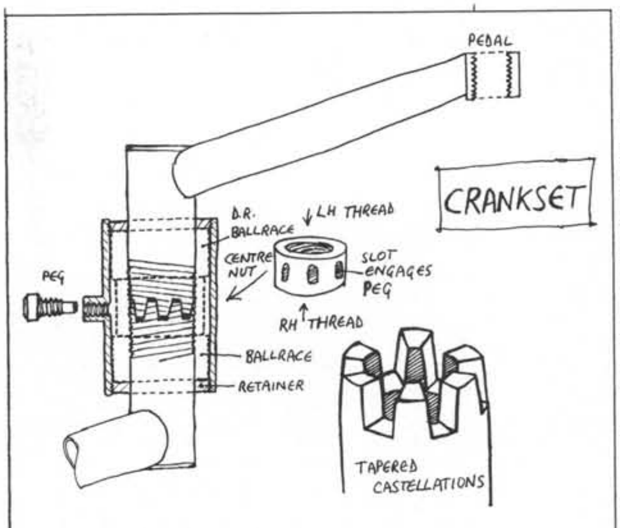

Figure 6 Crankset sketches

Crankset

Although the MWB layout gives a pedal/wheel clearance rather better than a shorter design, it needs the pedals slightly further apart, else the crank ends foul the fork crown or front brake. Rather than use a very long crank axle, I made my own cranks using cut-down fork blades bronze-welded at a slightly oblique angle directly onto an axle which is split in the middle. The inter-locking tapered castellations joining the two halves of the 22mm (7/8")- diameter axle are pulled together by a lefthand thread on one half axle, and a righthand thread on the other. Both threads engage a central double-threaded nut. A peg screwed through a bush in the housing engages a slot in this nut and holds it while the cranks are turned to screw them together or apart. The cranks revolve on double-row 7/8 x 1-5/16 ballraces, which are an interference fit in the housing.

The cranks have worked well over several years, and stay tight despite not being machined to a fantastic accuracy (I used a home-made bevel-sided mill mounted between centres on my rather flimsy lathe, which resents being used as a milling machine).

Transmission

Two chains are used, both on the righthand side of the bike, with the gears on the rear chain which is offset towards the outside. As well as avoiding a single long chain with a tendency to flap about, this arrangement has several other advantages, particularly for touring:
- it achieves a good range of gear ratios with a small rear wheel, and does not need a very large and expensive front sprocket;
- the rear wheel is undished and therefore much stronger,
- the transmission can be made more rigid in very low gears, and
- the front chain is as close to the central frame tube as possible, reducing strain on the frame as well as being further from one's trousers.

These benefits have to be weighed against the extra complication of an intermediate gear cluster. I used a five-speed freewheel turned inside out and running on ballraces. It required a certain amount of fiddling with shim washers to get the 15, 20 and 28T driving sprockets in the right places. A Campag touring front changer shifts the rear chain quite nicely over these three sprockets.

To prevent the cluster from coming unscrewed in use, I filled the thread inside the screw-on sprocket with brazing metal and machined a lefthand thread, and a complementary thread on the freewheel body. The non-folder used a sliding adjuster on the crank-axis housing to tension the front chain. On the folder I saved weight by using two pulleys on the return run. These also lift the chain above the luggage when touring. Some movement of the seat is available to trim the machine for leg length.

Suspension

Pieces of rubber collected from the roads while out cycling have proved ideal for the rear suspension. The best ones have been cylinders sized from 38 to 51mm (1-1/2 to 2") diameter, with axial moulded holes, and a piece of hard thick sheet from which I turned packing discs for height adjustment. I used string looped through the holes to keep the rubber in place and to stop the swing arm from dropping when the bike is lifted.

The rear swing arm pivots on the axis of rotation of the intermediate sprocket cluster. Some people suggest that the upper chain line should intersect the suspension pivot, so the suspension is not affected by the pulling force of the chain. I think this ignores the force exerted between the tyre and the ground, which as well as propelling the bike forwards, also tends to move the arm down.

Materials

The front half of the frame is Y-shaped, composed of a length of 38mm (1-1/2") aerospace chrome-moly behind the head tube, joined to two pieces of 32mm(1-1/4") 501 mountain-bike tube which I rolled to a 25 by 38 mm(1" by 1-1/2") ellipse. The lugless bronze-welded joint leaves just a pair of small holes, one on each side, through which I could not resist passing the twin cables for the front cantilever brake. The seat-supports are of 25mm (1") tube, with the 16mm (5/8") handlebar tubes inside. Much of the rest of the back is 29mm (1-1/8") tube, and the rear swing arm uses recycled 13mm (1/2") seat stays.

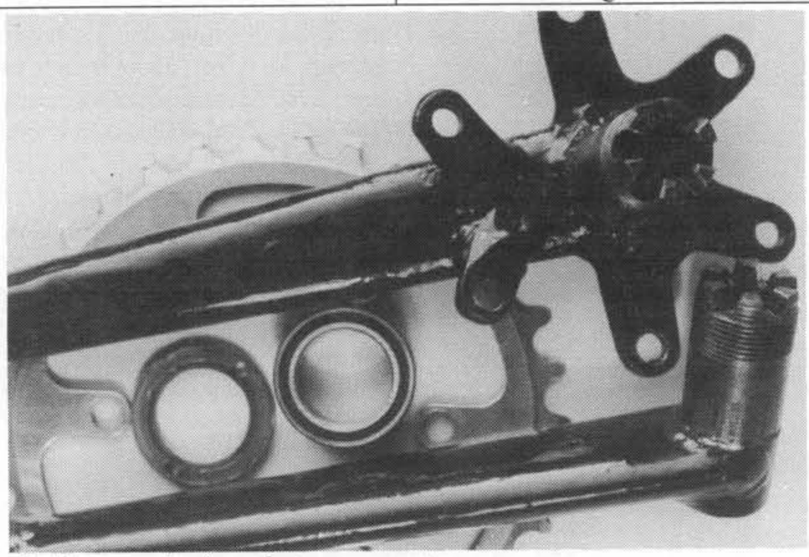

Figure 7 Photograph of crankset components

Handlebars and swing arm pivot on needle-roller bearings running on slightly oversize inners. For the steering pulleys I used a hard aluminium alloy (obtained in useful offcuts from a local scrapyard).

Hinges

The folding bike's main hinge is in the middle at the bottom, just ahead of the seat, and takes the form of a skewed tube with threaded end plugs brazed in. Bolts passing into these plugs secure two precisely fitting end caps which are part of the back section of the bike. An expander bolt can be screwed through a bush to spring the back section apart. Next time I might slant the hinge the other way, avoiding the acute angle at the left end of the tube and bringing the front of the bike to the chain side of the back when it is folded.

Two pairs of conical couplings are used at the top joint, which is compressed by the rider's weight. The handlebars fold inwards onto the seat, and the back of the seat folds rearwards using a hinge which also allows useful adjustment of the angle for riding, using a strut with some carefully placed holes.

On the road

I finished the folding two-wheeler in June 1993, and clocked just over 1600km (a thousand miles) by the autumn (when the mileometer failed during heavy rain in France). Much of this distance was covered while carrying camping gear. (Heavier items like a cooking stove were stowed between the wheels, and items needing easy access behind the seat. A rolled-up tent was carried under one side of the seat, and a sleeping mat on the other side, projecting backwards to overlap the rear wheel.)

Overall I am pleased with the machine, and while there are a few things I would change, I do not have any plans to make a substantially different two-wheeler. Personally however I doubt that any 2-wheeled recumbent can be ideal for heavy touring, and with this in mind I have recently started building a folding 3-wheeler.

Dimensions

Wheelbase	1190 mm	47"
Overall Length	1700 mm	67"
Folded Size	610x330x860 mm 24"x13"x34", approx.	
Weight	15 kg	33lb.
(with rear fairing, dynamo, mudguards)		
Gears	18 speed; 25" to 101"	
Wheels	500A rims from Pashley, Michelin tyres	
Folding Time	about 10 minutes, with out fairing	

Nick Abercrombie Andrews
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Nick Andrews is at present a PhD student researching into parsing methods for natural languages. Before returning to university he manufactured Bikeasy panniers.

Letter

Regarding "Assault on the hour" in HPV News, Dec. 93, Tim Leier threw out the rolling-resistance part of his equations to obtain a simple closed-form solution, his rationale being "rolling re sistance becomes relatively minor at higher speeds". This is true for unstreamlined bicycles but not at all true for a super streamliner. Using his figures for a streamlined and a regular bike of Cd 0.10 and 0.85, and frontal area of 0.4 and 1.0 m^2, we get a power consumed at 90 km/h of 376W and 8 kW. The rolling resistance is 85 W for each, which is almost 25% of the streamliner's power but only about 1% of the huge power required by the regular rider.

The message is that when you lower aero and other drag, the rolling drag is quite important!

Doug Malewicki, 14962 Merced Circle, Irvine CA 92714, USA

Review, cont. from p. 17 1993 HPV SYMPOSIUM Technical University of Eindhoven

I am sure that HPV designers would regard this range as huge: an increase in muscle efficiency from 23% to 24% would yield an increase in output of over 4%, enough to make a huge difference in a race or speed contest.

There are some interesting graphs showing the efficiency and oxygen consumption for various sports. Cross-country skiing seems to be top in both categories. "The more muscles, the more joy" is how the author recommends we should design transmission interfaces in future.

The third major paper was missing the first page in the preliminary copy I received, but the general topic was government bicycle and HPV policies.

The last part of the proceedings consists of 27 full-page descriptions and illustrations of various HPVs, showing the range of creativity we have come to expect in Europe. They are from The Netherlands, Germany, Denmark, Britain and Belgium, and about half are tricycles and half bicycles. Each entry has useful information in a standard format, including the price. There's no doubt that the best place to try out HPVs is a symposium or a race-meet; having these standard data made the proceedings all the more useful.

The address given to order the volume is
studieveriging "Simon Stevin"
W-Hoog 1.07A, Postbus 513
5600 MB Eindhoven, The Netherlands
Ph. 31-40-473-313; fax 31-40-437-175

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Dave Wilson

THE ROTATOR COMPANION DOUBLE-RECUMBENT TANDEM

by
John Allen

I recently had an opportunity to ride the Rotator Companion tandem, a dual-recumbent tandem built by Steve Delaire. That's unusual enough in and of itself, but beyond this, the Rotator has many clever, fascinating and unusual mechanical details.

The most unusual aspect of the Rotator is the independent shifting system for the two riders. Essentially, the bike has two drive trains complete from crankset to freewheel. The two freewheels are mounted on a jackshaft which drives the rear wheel through an additional chain.

The Delaire Rotator dual-recumbent

The dual drive trains make for a very different teamwork experience than on a conventional tandem. Certainly, they allow better compatibility between riders with different pedaling styles, and have another significant advantage: starting and stopping does not have to be a synchronized ballet. This is probably an especial advantage on the Rotator, where startup is difficult enough as is. With the dual, independently freewheeling drive trains, one rider can steady the bike while the other applies the first pedal strokes.

There are other, mechanical advantages. On a conventional tandem, the final drive chain takes a double load, but it also does all of the shifting. Both riders must release force on the pedals when shifting; wear on the front derailleur, chain and freewheel are much greater than on a solo bike. But on the dual-drive-train Rotator, each of the two drive trains takes only the strain of its own rider, and does not wear any faster than a drive train on a conventional, solo bicycle. Only the short, final drive chain, with no derailleurs or multiple sprockets, takes the double load. Though the Rotator has small, 20-inch wheels, the final drive chain's sprockets have a step-up ratio which allows normal-sized chain wheels and sprockets to be used, where extra-large chainwheels or small sprockets would otherwise be necessary.

The final drive chain is tensioned with a two-pulley rear derailleur cage attached to the frame. The chain never drives in reverse, so there is no problem with losing tension as when this approach is used on the synchronizing chain of a conventionally geared tandem.

With the dual primary drive trains, it is not only possible for the two riders to choose different gears at any one time, but the two gearing systems can be set up differently to suit their preferences in cadence and shift pattern. This was in fact the case with the Rotator provided to us. Both drive trains were set up as 18-speed crossover, but the stoker's drive train had a somewhat lower bottom gear, consistent with a rider who preferred to spin faster when climbing hills. Delaire claims that the dual shifting makes it easier to climb hills, because the stoker can gear down for the hill and keep pushing when the captain shifts.

However, the dual shifting also has its disadvantages. One of the joys of tandeming for me is the ability to ride compatibly with a person who has little bicycling experience. On a conventional tandem, not only is the stoker relieved of the need to shift, but the stoker's and the captain's legs move as one. The stoker learns in the most direct possible way about shifting, cadence, coasting, cornering, starting and stopping. When quick decisions must be made about whether to stop or sprint, the stoker learns of them through his or her feet, and this is much quicker than a verbal command. On a conventional tandem, an inexperienced though strong stoker can efficiently contribute power; but the stoker on a dual-drive-train Rotator has to know how to use the gears. (The same issues occur with the Adams Porta-bike Rann trailer, and are worse because a child is less likely to know how to shift.)

TABLE 1
Rotator Companion tandem gearing

Wheel diameter	**Final-drive ratio**
19.2 inches	28/20

Captain's gearing (in inches)

Sprockets	Chainwheels 48	38	28
13	99.3	78.6	57.9
15	86.1	68.1	50.2
17	75.9	60.1	44.3
19	68.0	53.8	39.6
21	61.5	48.7	35.9
24	53.8	42.6	31.4

Stoker's gearing

Sprockets	Chainwheels 48	38	28
13	99.3	78.6	57.9
15	86.1	68.1	50.2
18	71.7	56.8	41.8
21	61.5	48.7	35.9
24	53.8	42.6	31.4
28	46.1	36.5	26.9

The Rotator's dual-cadence option requires a somewhat longer rear bottom bracket axle, to allow room on the jackshaft for the two freewheels. The example which I rode had an axle fully eight inches long, though Delaire states that this long an axle is not necessary: the particular one on this bike was obtained from another builder and was the most convenient one to install when this bike was built up. Even this unnecessarily long axle does not seem to affect pedaling efficiency significantly in the recumbent riding position, and it does not reduce cornering clearance.

The Rotator's dual-cadence option is different from that of the Counterpoint Opus recumbent/upright tandem. On the Opus, the stoker's drive train has its own shifting, but feeds into the captain's drive train. The stoker can choose a cadence differential, but the captain's shifting also affects the stoker's pedals.

One major decision in designing a recumbent is between short and long wheelbase. The front wheel must be either ahead of the pedals or behind them. A long-wheelbase solo recumbent is already as long as a conventional upright tandem. A short-wheelbase solo recumbent is likely to have somewhat skittish handling because the steering becomes very fast, and the rider's center of gravity is so close to the front wheel.

The Rotator has the front wheel behind the captain's pedals. This short-wheelbase option is a natural choice for a recumbent tandem, because the extra length for he second rider places the center of gravity farther toward the back, and slows the steering.

However, the Rotator also has the rear wheel tucked in under the stoker, and this may be a factor in its most serious difficulty, its slow-speed handling.

If you have ever ridden a bicycle with a very large, heavy package on your rear rack, you have some idea of how the Rotator feels to the captain. If you have ever ridden on the rear rack of a bicycle (I will assume no responsibility...), you have some idea how it feels to be the stoker.

The Rotator's wheelbase is only about 54 inches, shorter than that of a conventional tandem or solo long-wheelbase recumbent. The stoker's mass is almost directly over the rear wheel, and contributes almost nothing to the quick balance corrections achieved by steering motions of the front wheel. Larger steering motions are necessary to maintain balance at low speeds than with any other bicycle I have ridden. The Rotator develops a pronounced, and fast, weave -- about two cycles per second -- which can be very disconcerting to the stoker and hard to control for the captain.

The recumbent rider position exacerbates these handling difficulties for two reasons I can think of: the riders are lower, making the bike fall over faster (like the difference between trying to balance a yardstick and a foot ruler on the end of your finger); and the riders can not shift their upper bodies sideways to adjust balance as on an upright bike. Solo recumbents, however, remain predictable and easy to steer despite these differences.

When I asked Delaire about the Rotator's handling, he indicated that a recumbent tandem handles better with the front wheel farther forward. The first prototype of the Rotator tandem has its front wheel directly under the captain, with direct steering from under-the-seat handlebars. According to Delaire, this prototype is even more difficult to steer then the current Rotator: "like a unicycle," because the captain is right over the front wheel. He also indicates that he has experimented extensively with steering geometry, and changes in fork rake can make the Rotator's steering lighter or heavier but that the slow-speed weave is more-or-less "in the nature of the beast."

The problem is that the front wheel can not be directly under the crankset; if moved far enough forward to clear the crankset, the tandem has "tiller steering" and a 97-inch wheelbase. Delaire has actually built a tandem this way and finds it OK to ride, at least at high speed.

Controlling the Rotator does become less difficult with experience. The handling might also become more controllable with a rear wheel farther behind the stoker position. With a wheelbase only as long as that of a conventional tandem, the stoker's mass would shift farther sideways with steering motions, and they would more quickly correct balance.

I rode the Rotator in both the captain's and stoker's position, with my friend Sheldon Brown and with my wife.

Since a herniated spinal disc made it uncomfortable for her to ride an upright bicycle, my wife Elisse has put in over 30,000 miles on her Tour Easy solo recumbent. The recumbent riding position in and of itself holds no terrors for her. Nor does riding on the back of a tandem. My wife once tried riding on the back of my upright tandem. We kept going a half-mile, until her back pain stopped us. She did not feel uneasy or nervous.

The Rotator was another story. Both Sheldon and my wife felt insecure in the stoker's position, as though we were about to fall over. In the captain's position, I felt that we were not, and that control would improve if we just gained some speed; but that's not how my stokers felt.

Elisse also had a problem with the height of the seat. She is used to the Tour Easy, which lets her comfortably put both feet on the ground when starting and stopping. The Rotator's seats must be high for the riders to clear the wheels. It is necessary to slide back into the seat when starting and forward out of it when stopping. I was used to this, from my experience with a Counterpoint Presto solo recumbent, but my wife was not.

Delaire indicates that design modifications to compact the front fork as much as possible have lowered the stoker's saddle by two inches in newer versions.

The Rotator reminds me of a goose, very clumsy in takeoffs and landing but graceful once in flight. However, I know that Delaire is confident to ride hilly double centuries on the Rotator with his wife -- and the low bottom gears indicate that they expect to be climbing at low speeds. Clearly, experience plays a big part in feeling confident on this bike.

The Rotator is equipped with a Magura hydraulic caliper brake on the front wheel and an unidentified motorcycle-type disk brake on the rear wheel. Braking is powerful and easily controlled. The usual rule that a tandem needs three brakes can be broken here because the rear disk brake is powerful enough to serve also as a stopping brake.

The rear seat is attached with quick-release clamps which allow an almost instant adjustment for different stoker leg lengths. The captain's position is adjustable by a telescoping boom holding the front crankset, as on Counterpoint recumbents.

Certain details could use refinement. A kickstand is appropriate, and useful, as on any long recumbent. But with several feet of bottom tube, there is no need to position the kickstand right behind the crankset, as on a conventional bike. If you roll the bike backwards with the kickstand down, the crank will run into it and probably damage it. The stoker's thumbshifter levers are inside the seat rails where they are difficult to operate.

The bike uses a hodgepodge of imported metric and U.S. inch-size fittings, making it necessary to carry more tools. This is not the builder's fault but rather that of our political process which leaves us as one of two nations in the world (the other is Liberia) which has not adopted the metric system. One easily-remedied reason that the U.S. has trouble competing in world markets. But I digress.

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