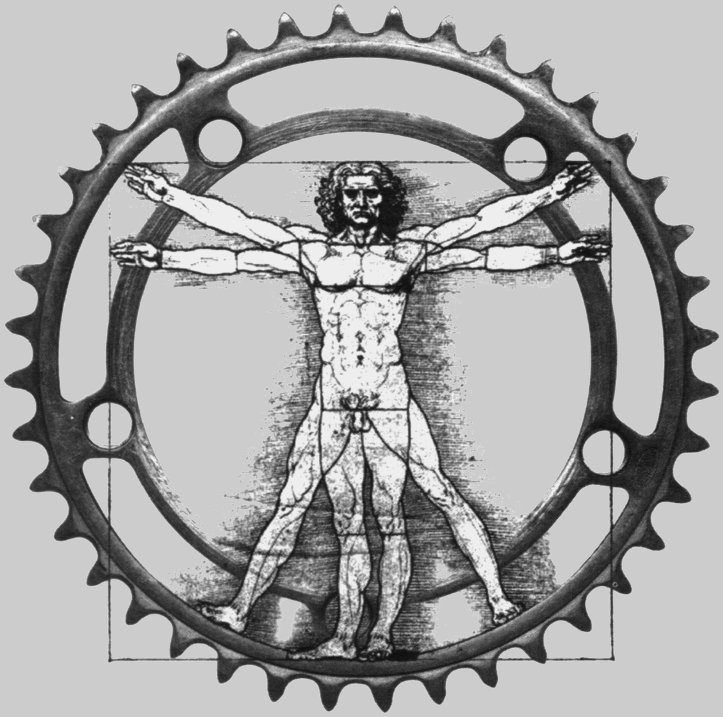


HUMAN POWER

TECHNICAL JOURNAL OF THE IHPVA

NUMBER 55 WINTER 2003-2004

Number 55
Winter 2003-2004 **$5.50**

HUMAN POWER

is the technical journal of the International* Human Powered Vehicle Association.

Human Power 55, Winter 2003/4

Editor and production (this issue):
Theodor Schmidt
Ortbühlweg 44
CH-3612 Steffisburg
tschmidt@ihpva.org

Associate Editor
David Gordon Wilson
21 Winthrop Street
Winchester, MA 01890-2851 USA
dgwilson@comcast.net

IHPVA
Richard Ballantine, UK, Chair
Paul Gracey, HPVA representative

Publisher
HPVA
PO Box 1307
San Luis Obispo, CA 93406-1307 USA
Fax (USA) (866) 643-7102

Human Power (ISSN 0898-6908) is published irregularly by the Human Powered Vehicle Association, a non-profit organization dedicated to promoting improvement, innovation and creativity in the use of human power generally, and especially in the design and development of human-powered vehicles.

***Special note for this issue**

This is the last issue of Human Power produced on behalf of the IHPVA. Future issues will be entirely by and for the HPVA. Please send contributions and comments for the next issue of Human Power to the HPVA. The IHPVA may endorse future electronic human power journals which become available on the internet and are mostly freely available for everybody. Any information about this will be available at www.ihpva.org.

FROM THE EDITOR

This issue of Human Power marks an end, a beginning, and transitions. Jean Anderson, who had produced Human Power for many years, retired from most HPV duties with the last issue. Jean had worked in almost every position the IHPVA and the HPVA had to offer and we are grateful for her great amount of work for so long. Jean remains physically active and rides her recumbent trike in San Luis Obispo and elsewhere.

This is my last issue of Human Power. I want to help start an electronic publication called Human Power International Journal, which will be freely available to everybody with internet access. The publisher will be the newly formed Human Power Institute (HuPI) described in this issue. The HPVA will however also continue to print Human Power and has several offers from potential new editors to choose from.

Why this new development? The traditional human-powered vehicle movement has matured but is in danger of stagnation. With so many splendid achievements having been accomplished on land, water and in the air, the pioneering spirit has diminished. With so many excellent HPVs and HPBs available commercially, the interest in home construction has all but vanished, and with regular HPV races being organized in many countries, the original spirit of rebellion against established sports bodies is past, indeed we have become established sports bodies ourselves.

What about the environmental side of the human power movement? There are some positive developments in western countries which have by now at least realized the problems of over-motorization, even if most aren't doing anything about it. Improvements are the increase in pedestrian areas and the introduction of pedicabs or velotaxis in some western cities. In Switzerland so-called "Low-Speed Vehicles" (LV, i.e. walking, cycling, and HPVs) have been officially recognized as a traffic category worthy of promotion by government funds. In contrast, the mayors of many large Chinese cities regard HPVs of all types as backward and are promoting motor vehicles and beginning to ban cyclists from major streets. The developing countries seem intent on repeating northern-western mistakes and regarding money and consumerism more highly than a high quality of life or indeed life itself.

With the globalization of information it is now possible - and necessary - to shorten the dark "oil age" for the developing countries. To this end, the information we have to offer must be available worldwide on the internet.

It must also be available freely, especially as differences in earning power vary enormously worldwide. Freedom of information promotes its propagation, as is shown by the presently largest common project of global human society: the GNU/Linux computer operating system with free and open source software.

Also of interest to readers will be former editor Dave Wilson's project to make available the archive of all past issues of Human Power on a CD-ROM and on the internet, complete with a searchable index. Dave and the founding members of the Human Power Institute have been hard at work at assembling and indexing the issues. The CD-ROMs should be available presently from the HPVA and other IHPVA members.

Several readers wrote regarding the editorials in HP54 which commented the connection between HPVs, oil and the Iraqi wars. One reader supported the war and three the reasoning in the editorials. All stressed the importance of the upcoming elections later this year in the USA, which have far-reaching consequences for the USA and for the whole world.

Besides choices like between right and might, freedom and deception, education and armaments, I feel it is of utmost importance to ratify the Kyoto Protocol for limiting greenhouse gases, which still requires the signature of Russia or the USA. The good news is: most readers of Human Power can do a lot about it: ride HPVs more and drive less, telecommunicate more and fly less, and above all: *go and vote in November*: it has never been so important!

Also in this issue...

The main theme is the efficiency of human and mechanical drive trains. We have a continuation of Danny Too's work with his and Gerald Landwer´s article "Biomechanics of HPV" Part 2.

Iain Crouch's article on the optimization of gearing for an automatic bicycle reaches a similar conclusion: you have to get the operating points of both the human engine and the mechanical gearing system right.

Rohloff are also concerned about the right gear in their article on hub gear efficiency. This is however mainly a criticism on Kyle's and Berto's previous efficiency measurements, including lots of new measurement data of the 14 speed Rohloff hub gear.

Vernon Forbes rounds up this subject with his "Elegy for Sturmey Archer", a fascinating combination of intimate hub gear details, recent industrial history, and moral indignation of the present harsh industrial climate which shifts jobs overseas just as easily as goods.

Finally, we have Bill Patterson's proposal for a new scoring system for human-powered helicopters and a success story describing a fleet of Philip Thiel's "Escargot" human-powered houseboats.

The Biomechanics of Force and Power Production in Human Powered Vehicles

by Danny Too and Gerald E. Landwer

Abstract

There are a large number of factors affecting performance in human powered vehicles (HPV). Designers of HPV's often focus on how resistive forces (friction, drag) can be minimized, as opposed to how propulsive forces can be maximized. How to maximize propulsive forces through vehicle design is not often understood because of a complex interaction between internal biomechanical factors (muscle force/torque/power production) and external mechanical factors (e.g., seat-to-pedal distance, crank arm length, seat-tube angle, backrest angle, chain wheel size). The purpose of this paper is two-fold: (1) to provide information, from a biomechanical and physiological perspective, how muscle force is produced and modified; and (2) to examine how the muscle force produced interacts with external mechanical factors to produce power.

Introduction

Speed and performance in land based HPVs are a function of the amount of propulsive forces produced versus the amount of resistive forces that need to be overcome. Designers of HPVs often focus on minimizing resistive forces (drag and rolling resistance) in the construction of a vehicle. This would include reducing vehicle cross-sectional area, the surface area, and drag coefficient to decrease aerodynamic drag. To decrease rolling resistance, vehicle and rider weight would be reduced, and the wheel and tire properties modified (e.g., using a larger wheel diameter, greater tire pressure, etc.). Since aerodynamic drag forces have a greater effect on speed than rolling resistance, the design and construction of HPVs have focused predominantly on how to minimize drag forces. A vehicle is often constructed first, with the objective to minimize drag, and then a rider is selected to fit in the vehicle - without consideration as to whether the rider is in the most effective seating position to maximize force and power production.

In attempts to increase propulsive force, designers will modify or manipulate external mechanical factors such as crank arm length, seat-to-pedal distance, seat-tube angle, backrest angle, chain wheel size, and gear ratio (and/or select bigger and more powerful riders, such as competitive cyclists or world class athletes), without really understanding how muscle force is generated, modified and might interact with these external mechanical factors. Modifications of these mechanical factors are often done intuitively or randomly, without empirical data to support the variable(s) that should be manipulated, the extent of these manipulations, and whether some variables might interact with other variables to affect power production. Therefore, depending on the design of the vehicle, the rider could be seated in any number of cycling positions, with different body orientations and joint configurations, pedaling with any combination of crank arm length, seat-to-pedal distance, seat-tube angle, backrest angle, and chain wheel size - without scientific evidence as to what factors and/or combination of factors will maximize propulsive forces. This is thus the reason for such diversity in HPVs. It should be noted that the optimum parameter (e.g., crank arm length and/or seat-to-pedal distance) to maximize power for one cyclist (determined from trial and error) might not be optimum for another, especially when cyclists have different anthropometrical characteristics (in height, leg length, thigh/leg length ratio, etc). To provide information to designers of HPVs about how and why seating position may affect propulsive forces, a review of how muscle force and power are produced and modified, will be provided.

Force-Length Relationship

Based on the force-length relationship, a muscle can produce it's greatest force at it's resting length. At resting length, an optimal overlap occurs between the muscle contractile elements (actin and myosin filaments) resulting in a maximum number of cross bridges that can be formed. With increasing or decreasing muscle lengths from resting length (such as when a muscle is lengthening or shortening during a pedal cycle), the force a muscle can produce will decrease. Therefore, an inverted U-shape curve best describes the force a muscle can produce with increasing length from it's minimum length to resting length, and then from resting length to it's maximum length (see Figure 1).

For single joint muscles, the joint angle corresponding to this resting length can be determined experimentally using an isokinetic dynanometer or using maximal isometric contractions at different joint angles throughout the joint range of motion. However, for multi-joint muscles, it is much more difficult and complex to determine the joint angle(s) at which resting length and maximum force production occur at. For example, the rectus femoris is a two-joint muscle that crosses the hip and knee and is involved in flexion of the hip and extension of the knee. If maximal isometric knee extension strength is measured when the hip and knee are both at 90 degrees of flexion (such as the starting position for

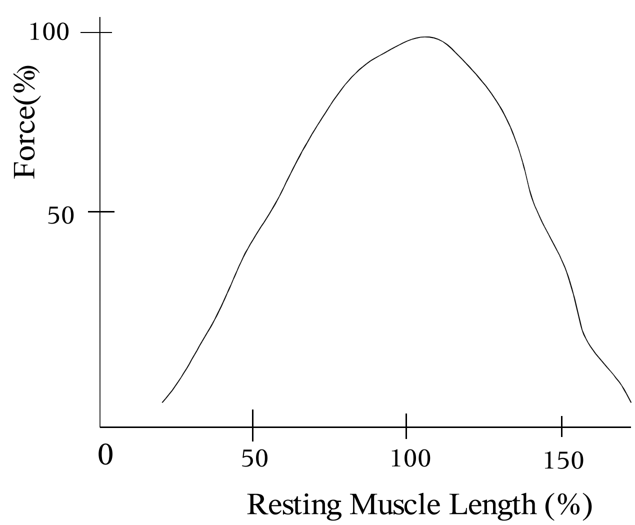


Figure 1: Force-Length Relationship

performing a leg extension when seated in an upright position), the force produced by the rectus femoris will change if the hip angle is changed (such as when leaning forward or backwards during the isometric contraction). Changes in hip angle (with hip flexion or extension) will change the length of the rectus femoris (shortening or lengthening it) and alter it's maximum force produced at the knee. Conversely, if the hip angle is fixed and the knee angle is free to vary, different maximum isometric forces will be observed with different knee angles (due to different muscle lengths of the rectus femoris). Complexity is further increased when, both the hip and knee angles change simultaneously during a dynamic contraction, such as in a squat or leg press. During a squat or leg press, when both the knees and hips are extending during the extension (pushing) phase of the squat or leg press, the rectus femoris would be shortening at the knee while lengthening at the hip. During this phase, the muscle length (and force produced) could remain the same or change, depending on whether the hip and knee are extending simultaneously, synchronously, asynchronously, and/or have the same change in angles. A similar analogy can be made to cycling.

In cycling, there are multi-joint muscles (hamstrings, rectus femoris, sartorius, gracilis) acting at the hip and knee, and knee and ankle (gastrocnemius, plantaris) to produce force during a pedal cycle. The hip, knee, and ankle joint angles (resulting in resting muscle lengths) that maximize force production during a pedal cycle are unknown. During the propulsive phase in cycling, both the hip and knee are extending. The hip and knee angles that might maximize hamstring force production (during hip extension when cycling) may not be the same angles to maximize rectus femoris force production at the knee (during knee extension). Knowing (or not knowing) the specific joint angles that would maximize force production during a pedal cycle is probably not that important if cyclists were constrained to pedal in the same seating position. For example, if a selected seating position (e.g., standard upright cycling position) results in joint angles that are fairly efficient (or inefficient) for one individual, it would probably result in joint angles that are similarly efficient (or inefficient) for others. But if two dissimilar cycling positions are used (e.g., a high upright sitting position versus a low recumbent sitting position), one cycling position may result in greater production of power due to more effective joint angles (from more optimal muscle lengths) than the other. In this case, information about the specific joint angles that would maximize force production during a pedal cycle is important if cycling performance is to be maximized.

Seat-to-Pedal Distance

If some seating position (e.g., standard upright) is selected regardless of whether it results in effective or ineffective muscle lengths and joint angles, and a standard crank arm length is used, the only manipulation to change hip, knee, and ankle angles, would be changes in seat-to-pedal distance (seat height). Of course the cyclist could shift the saddle-seat location a bit, or lean forward to rest on the handlebars, or sit more upright, to manipulate the hip angle. But this change in hip angle would be minimal compared to the change that would occur with changes in seat height. If the seat height is changed, the minimum and maximum angle of the hip and knee will change, although the range of motion at the hip and knee will remain the same. This would mean that with changes in seat height, contraction of the muscles would occur in different regions of the force/tension-length curve during a pedal cycle (although the amount of muscle shortening/lengthening would remain the same). Maximum force production would then occur with a seat height where muscle contraction corresponds to the portion of the force/tension-length curve closest to resting length (or at resting length). This is supported by studies that reveal an optimum seat height to maximize cycling performance in aerobic and anaerobic tests (Gregor & Rugg, 1986; Nordeen-Snyder, 1977; Shennum & deVries, 1976; Thomas, 1967; Too, 1993).

However, this traditional upright cycling position with specified joint angles (minimum, maximum and range of motion) for the hip, knee, and ankle (dictated by the seat height and standard crank arm length) during a pedal cycle might not be the most effective position to produce force. The most effective position may be a non-traditional cycling position (i.e., recumbent) that utilizes joint angles and muscle lengths (for both single and multi-joint muscles) that correspond to the resting length portion of the force/tension-length curve (Too, 1996). This is supported by studies where hip angles (minimum and maximum) were systematically manipulated (through changes in seat-tube-angle, using 5 positions ranging from a high sitting upright position with the hips above the pedals, to a low sitting position with the hips below the pedals) while the knee angles (minimum, maximum, range of motion) were controlled (Too, 1991, 1990).

Joint Angles, Muscle Length, and Crank Arm Length

Unlike changes in seat-to-pedal distance with a fixed crank arm length, a change in crank arm length with a fixed seat-to-pedal distance will result in a change in the range of motion during a pedal cycle at the hip and knee (Too & Landwer, 1999, 2000; Too & Williams, 2000). In addition, the minimum and maximum hip and knee angle will also change unless the seat-to-pedal distance is determined from maximal extension of the hip and knee during one pedal cycle. In this case, the maximum hip and knee angle will not change with changes in crank arm length whereas the minimum and range of motion will change. This presents greater complexity in determining the joint angles and range of angles at the hip and knee that would maximize force production because: (1) with changes in crank arm length, the amount of muscle shortening and lengthening would change, and depending on whether the crank arm length was increased or decreased, contraction of the muscles would occur over greater or lesser portions of the force/tension-length curve during a pedal cycle; and (2) with an increased crank arm length, a greater torque can be produced at the crank spindle with the same force (or the same torque can be produced with a smaller force). The interaction between the force produced at different muscle lengths - during a pedal cycle when different crank arm lengths are used - with the length of the crank arm, will ultimately determine the torque which can be produced at the crank spindle. Of course, the resulting interactions to produce force and torque would be even more complex if different combinations of seat-to-pedal distances, crank arm lengths, and seat-tube-angles were used, resulting in an extremely large number of combinations of joint angles (minimum, maximum, range of motion) and muscle lengths at the hip, knee, and ankle. It should be noted that it is not the actual seat-to-pedal distances, crank arm lengths, and seat-tube-angles that are important in maximizing force and torque. Instead, it is the resulting hip, knee, and ankle angles from the combined interactions of these external mechanical variables that correspond to the portion of the force-length curve closest to resting length to produce the force that will maximize torque and power production.

Force-Velocity-Power Relationship

Based on the force-velocity relationship, the force a muscle can produce will be affected by it's velocity of contraction. With a high velocity of contraction (and no load), minimum muscle force (and power) can be produced because the actin and myosin filaments would be sliding by each other faster than the cross bridges that can be formed and activated. As the load increases, the velocity of contraction decreases, and with a maximum load, the force of contraction becomes a maximal isometric one (resulting in zero power) (see Figure 2). Since power is a function of force and velocity, based on the force-velocity-power relationship, maximum power appears to be obtained with a load and velocity that is one third to two thirds of the maximum muscle force and velocity of contraction that can be produced.

From the force-velocity-power relationship, maximum power (or a desired power output) in cycling can be obtained with numerous combinations of load (chain wheel size, gear ratio) and velocity (pedaling frequency). However, it should be noted that there is not only an interaction between force (load), velocity (pedaling rate), and power, but also with muscle length. Depending on the muscle length with different cycling positions (i.e., upright or recumbent), the optimum combination of load and velocity to maximize power output is unknown and may vary with different cycling positions. This complexity is further increased if the crank arm length is manipulated.

Power Output, Load, and Pedaling Frequency

A change in crank arm length will not only affect force production by the hip and knee, by changing joint angles (minimum, maximum, range of motion) affecting muscle length, but it will also affect the torque produced at the crank spindle, the load that can be applied, the maximal pedaling frequency, and the resulting interactions in the production of power. For example, when compared to a long crank arm, a shorter crank arm will not only reduce the minimum, maximum, and joint range of motion at the hip and knee over one pedal cycle affecting muscle force production, but it will also result in a reduced torque (if the same force is applied) at the pedals. However, because of the shorter crank arm, there is a potential for a greater maximal pedaling frequency. Whether this greater maximal pedaling frequency can be obtained, will then be dependent on the load (gear ratio, chain wheel size) and resistance that needs to be overcome.

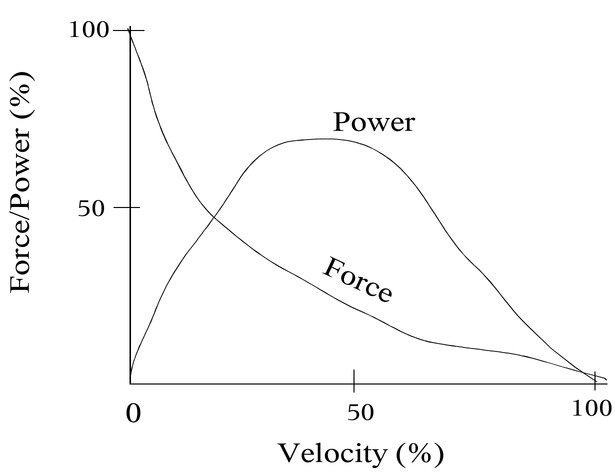


Figure 2. Force-Velocity-Power Relationship

According to Seabury, Adams, and Ramey (1977), (1) there is a most efficient pedaling rate for each power output; (2) the most efficient pedaling rate increases with power output; (3) the increase in energy expenditure when pedaling slower than optimal is greater at high power outputs than at low power outputs; and (4) the increase in energy expenditure when pedaling faster than optimal is greater at low power outputs than at high power outputs. This would suggest that if a given sustained power output is required to set a new distance record in some human powered vehicle event (such as the hour record or 24 hour record), it becomes important to know not just what is the optimal pedaling rate, but also the interaction of pedaling rate with crank arm length and load, in order to maximize power output, yet minimize energy expenditure and muscle fatigue.

On the other hand, to maximize performance of human powered vehicles for short distances (200 meter sprint) and set new speed records, a great deal of power would be required but only for a short period of time. To maximize this power, it is desirable to maximize both, force (i.e., load, gear ratio) and velocity (pedaling frequency). However, according to the force-velocity-power relationship, increasing force (load) to a maximum value will result in a decreasing contraction velocity (pedaling rate) to a minimum value. Therefore, with a fixed crank arm length, the maximum power appears to be obtained with a load and velocity that is 1/3-2/3 the maximum muscle force and velocity of contraction that can be produced. If the crank arm length is free to vary, the interaction between force (in this case, it would be torque) and velocity to produce maximum power, would be more complex. With a given force, the torque applied to the crank spindle would be less for a shorter crank arm, but the maximum pedaling rate would be greater. Conversely, with a given force, the torque applied to the crank spindle would be greater for a longer crank arm - and a greater maximum load can be used - but the maximum pedaling rate would be lower when compared to a shorter crank arm. To maximize power with increasing load, force and torque would also have to increase, assuming pedaling rate is already at a maximum. However, according to the force-velocity-power relationship, as load continually increases, there will be a critical load beyond which will result a decrement in velocity (pedaling rate), and this would be especially true for shorter cranks. With longer crank arms, greater loads can be used because greater torques can be produced, and due to the decreased maximal pedaling rate for longer cranks, the critical load beyond which will result in a decrement in velocity (pedaling rate) will be much greater than that expected for shorter crank arms. What is the critical load for different crank arm lengths (short and long), beyond which there will be a decrement in pedaling rate and/or power, is unknown. What is the optimal combination(s) of load and pedaling cadence for different crank arm lengths to maximize power production or to minimize the energy requirement for a given power output are also unknown. Of course this complexity is increased with the interaction of other factors (such as changes in seat-to-pedal distances, seating positions, etc.)

Other Considerations

Body orientation (trunk angle) with respect to the ground, and location of the lower extremities relative to the crank spindle are additional factors that need to be considered because of their possible effect on force production and total force contribution to the pedals in cycling. Changes in body orientation (trunk angle) will affect muscle force/tension-length relationships and force production if it results in hip angle changes. Changes in body orientation (trunk angle) without changes in hip angle may affect the body weight contribution to the force on the pedals (depending on the location of the lower extremities to the crank spindle). For example, a cyclist in a standard upright bicycle would have the leg weight contributing to the total force on the pedals during the power stroke. However, if a cyclist was in a reclining/recumbent position where the lower extremities were below the crank spindle (e.g., cycling in an inverted position), work would have to be done in not just overcoming the cycle resistance/load, but also in overcoming the weight of the lower limbs when pedaling-working against gravity, resulting in less total force applied to the pedals during the power/pushing stroke. Too (1989, 1994) determined that changing the body orientation (trunk angle) with respect to the ground does affect peak power production and power output. In fact, if cycling in a completely inverted position, it would probably be easier and more effective to pull against the pedals during the recovery phase (using the leg weight when it is aided by gravity) than during the power phase (where work would have to be done against gravity to overcome the lower limb weight). This would explain why recumbent bicycles are less effective in climbing hills when compared to the standard upright bicycle. Low sitting position recumbent vehicles that have pedals located above the cyclist's hip, require the cyclist to pedal upwards against gravity (to overcome some portion of their leg weight) during the power stroke. When climbing hills (and depending on the angle of the hill), the cyclist would need to overcome an even greater proportion of the lower limb weight during the power stroke, and thus requires an even greater expenditure of energy.

Summary and Concluding Remarks

As the limits of engineering design in HPVs to minimize resistive forces are reached, it becomes essential to focus on maximizing the propulsive forces. This requires an examination of the human engine powering the vehicle and how to maximize it's efficiency. This necessitates not just an understanding of how muscle force is produced (based on force/tension-length and force-velocity-power relationships), but also how they interact with external mechanical variables such as seat-to-pedal distance, seat-tube angle, and crank arm length to alter lower extremity joint angles (hip, knee, ankle), affecting force and power production. It should be noted that it is not the manipulation of the external mechanical variables that is important, but rather how the manipulation affects joint angles of the hip, knee, and ankle during the pedaling action. The question should *not* be "what is the optimal crank arm length or seat-to-pedal distance to maximize force and power production?" but rather "what are the joint angles that would maximize force and power production, and what manipulations in HPV design should be done to obtain these joint angles?" It should also be noted that the optimal crank arm length for a very tall individual will probably not be optimal for a very short individual, whereas the joint angles to maximize force and power will probably be similar for both the tall and short individual. It is beyond the scope of this paper to review the existing literature involving manipulations in external mechanical variables and the resulting effects on joint angles and cycling performance. However, that would be a topic for a future paper.

About the authors

Danny Too is an associate professor in the Department of Physical Education and Sport at the State University of New York at Brockport, and has been involved in human powered vehicle research since 1985. He can be reached at the
State University New York at Brockport
350 New Campus Drive
Brockport, New York 14420

Email: dtoo@brockport.edu

Gerald E. Landwer is a professor in the Department of Educational Leadership at the University of Nevada, Las Vegas.

References

Gregor, R. J. and S. G. Rugg. Effects of saddle height and pedaling cadence on power output and efficiency. In: Science of Cycling, E. R. Burke (ed.). Champaign, IL: Human Kinetics Publishers, 1986, pp. 69-90.

Nordeen-Snyder, K.S. (1977). The effect of bicycle seat height variation upon oxygen consumption and lower limb kinematics. Medicine and Science in Sports, 9, 113-117.

Seabury, J.J., Adams, W.C. & Ramsey, M.R. (1977). Influence of pedaling rate and power output on energy expenditure during bicycle ergometry. Ergonomics, 20, 491-498.

Shennum, P.L. & deVries, H.A (1976). The effect of saddle height on oxygen consumption during bicycle ergometer work. Medicine and Science in Sports, 8, 119-121.

Thomas, V. (1967). Scientific setting of saddle position. American Cycling, 6(4), 12-13.

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

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

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

Too, D.(1993). The effect of seat-to-pedal distance on anaerobic power and capacity in recumbent cycling. Medicine and Science in Sports and Exercise, 25(5), S68. (Abstract)

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

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

Too, D., & Landwer, G.E. (1999). The effect of pedal crankarm length on joint angle and cycling duration in upright cycle ergometry. XVIIth International Society of Biomechanics, Book of Abstracts, 311.

Too, D., & Landwer, G.E. (2000). The effect of pedal crankarm length on joint angle and power production in upright cycle ergometry. Journal of Sport Sciences, 18, 153-161.

Too, D., & Williams, C. (2000). Determination of the crank-arm length to maximize power production in recumbent-cycle ergometry. Human Power: Technical Journal of the International Human Powered Vehicle Association, 51, 3-6.

Optimal gear selection on an automatic bicycle

by Iain Crouch

Abstract

This paper describes an electronically controlled automatic transmission which was designed and built for use on a bicycle as a final year university project. The particular focus of the project was on the algorithm which determines the optimal gear to select at any time for maximised performance; i.e. acceleration and top speed, given the amount of effort the cyclist is putting in. This optimal gear selection is shown to be possible with the correct use of a prior strategy based on a fixed target cadence (pedalling rate). However, this cadence is unknown and varies over time and between different cyclists. It is shown that a cyclist's optimal cadence can be continuously estimated during normal cycling by a controller which fits recorded data to an assumed model. Microprocessor-based hardware was constructed and fitted to a bicycle to allow such a controller to be implemented and tested, and examples of the results are given and discussed.

Motivation

Automatic gearboxes have been fitted to motor vehicles for decades, yet rarely feature on human powered vehicles. The technical problems are obvious: the extra weight, cost and complexity of an automatic governor would be significant when fitted to a simple, lightweight cycle. In addition, existing 'crash' transmissions are not designed to transmit power during gear changes; to modify them would further compromise weight, strength, cost and efficiency. Using an electronically controlled gear selector can help reduce the weight penalty, and such devices have been introduced as optional extras on continuous-power transmissions such as the Shimano Nexus Auto-D hub gears and the Browning split chainring system (Kyle, 1995). Mechanical compromises are still present, however, so they are aimed in general at recreational cyclists - the Shimano system is specifically targeted at novice cyclists who find manual shifting complicated or distracting.

Figure 1, top: Modified transmission, showing automated derailleur, torque sensor and reed switches

Figure 2, below:Main unit

Existing automatic systems usually employ a simple gear selection strategy that attempts to maintain the cyclist's cadence at some fixed value, say 64 rpm. This is a comfortable rate for most people, however it is limiting in terms of performance – for example a racing cyclist would develop his peak power at a much higher rate. Furthermore, the selection of the best gear for maximum acceleration depends on characteristics that vary between cyclists and over time (due to fatigue for example). Some systems (including the Browning automatic) have the facility for training or adapting to the cyclist's preferences, which will improve performance but is still not optimal as it relies on the perception of the human rider.

The focus of this investigation was on improving the 'intelligence' of the automatic gear selection strategy, to assess whether more experienced cyclists could then find performance gains in using automatic rather than manual selection. To accomplish this, the extra intelligence must not only make up for the additional weight and complexity, but also the disadvantages of taking control away for the rider – for example a human cyclist has the advantage of being able to see changing conditions ahead. However (s)he also takes the effort of performing the gear change into account when considering changing, and will often only change gear when (s)he is uncomfortable with the gear (s)he is in, whereas an automatic selector may always choose the optimal gear given the information that it has.

It will be shown that a cadence-based selection strategy can be optimal if it uses the correct target cadence.

Hardware Description

As a university project rather than a commercial development, there was the opportunity to disregard some mechanical difficulties and practical issues and concentrate on investigating the control aspects. The prototype used to develop and test the control algorithms was therefore based on a conventional 'crash' gearchange, with modifications built for durability rather than light weight. This means that, although the drivetrain components were new and therefore as smooth-shifting as they could be, any cyclist using it must be vaguely aware that unexpected gearchanges would occur. The practicalities of commercial implementations on other drivetrains are considered in the 'Discussion' section.

The test bicycle was built around a Giant Terrago MTB frame, fitted with a Shimano Deore LX groupset and slick tyres. Figure 1 shows the modified transmission, which has a 9-speed cassette and a single front chainring. The existing derailleur was modified using a geared motor and a feedback potentiometer, which allows its position to be sampled by the onboard 8-bit PIC microcontroller and compared with the position for the desired gear. The motor was a surplus component whose specification far exceeded the estimated requirements; it rotated at 8rpm and produced 0.6 Nm (1.8 Nm peak) torque - greater than the torque from the derailleur's original return spring. Despite the motor's high inertia, a simple proportional feedback control routine, combined with the natural damping of the system, gave a satisfactory response for the derailleur, with a very slight overshoot to aid shifting.

Two magnets were mounted on opposing rear wheel spokes, allowing the microcontroller to time pulses from reed switches mounted on the bicycle's chainstays and calculate its speed. Similarly, magnets mounted on the cranks with corresponding reed switches on the chainstays send pulses to the microcontroller when the cranks are approximately horizontal. A chainring (actually the 'granny ring' from the bicycle's chainset) was mounted on a machined disk, which in turn was attached to a cantilever arm via a bearing and held against the tense upper part of the chain. The applied torque could then be measured using a pair of strain gauges fixed to the cantilever. The signal is amplified and sampled by the microcontroller's analog to digital converter. Data is regularly logged so it can be downloaded to a PC after each test run. The control system is completely self-contained, with batteries and a display built into a main unit (figure 2), and control routines, written in assembly language, running on the microcontroller.

The Model Used

To allow investigation, simulation and development of the control strategy, a general model of cyclists' output characteristics was required. A parabolic approximation to the relationship between maximum power and cadence was found to be satisfactory: derived power versus cadence curves [including Whitt and Wilson, 1974] were found to closely approximate parabolas, especially over the region of interest close to the cyclist's optimum peak power cadence. The relationship is shown as a graph in figure 3. As this parabola is known to pass through the origin, the model of available power P versus cadence ω of a cyclist may be completely defined by two values: his or her peak power P_{MAX} and corresponding optimum cadence ω_{OPT}:

$$P = P_{MAX} \left(1 - (\omega - \omega_{OPT})^2 / \omega_{OPT}^2 \right)$$

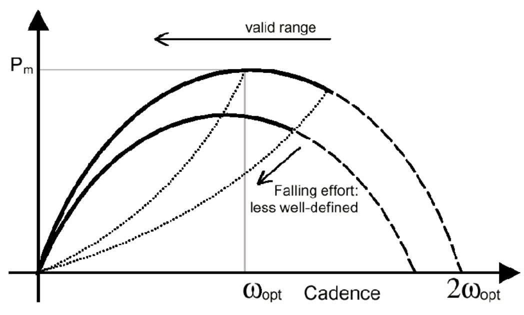


Figure 3: Diagram of the relationship between a cyclist's maximum power output and his pedalling rate

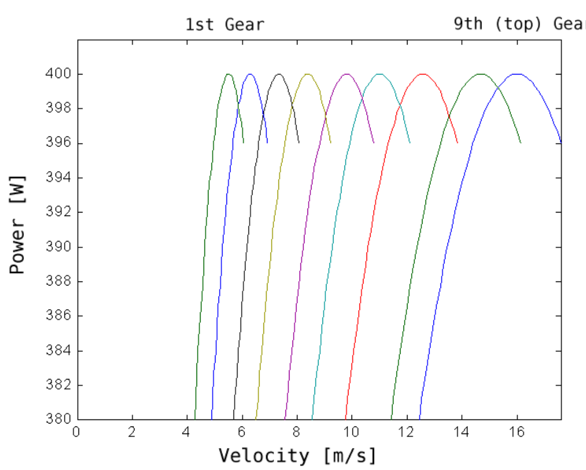


Figure 4: A Matlab graph of the corresponding available power versus road speed for different gears.

The cyclist's available power versus road speed, mapped through the available gears, is shown in figure 4. The graph

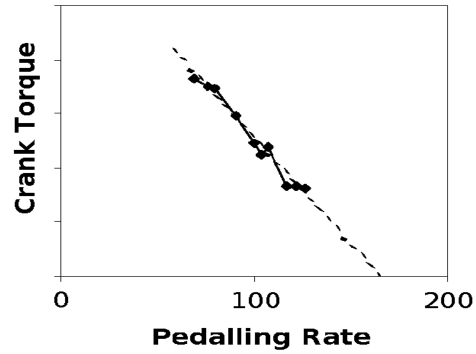


Figure 5: Torque vs. Pedalling Rate

shows that correct gear selection is essential to maximise performance – an automatic controller can calculate this correct gear, given the road speed, gear ratios and the cyclist's optimum cadence (his peak power need not be known, as this only scales the graph vertically). While road speed can be measured accurately, and the gear ratios are known, the optimum cadence remains unknown and variable. Running simulations (in Matlab) of maximum rate accelerations indicated that errors in the target cadence value used by a controller of just 10rpm would noticeably restrict performance. The variation in efficiency between gears is negligible and was disregarded.

Control – estimating the optimum cadence

This parabolic model not only allows a mathematical analysis of the effects of automatic gear selection, but also provides the key to allowing continuous online estimation of the optimal Cadence ω_{OPT} the one missing parameter that is required for theoretically optimal control.

A parabolic power versus cadence relationship implies a linear crank torque versus cadence relationship. This was verified using collected data such as the maximum rate acceleration in figure 3. The zero-torque intercept of the fitted line corresponds to the zero power intercept of figure 3; hence the cyclist's peak power is found at half the maximum cadence predicted by the fitted torque data. (NB this will not necessarily be the cyclist's actual maximum as it is outside the range of the data used to develop the model). This is the basis of the optimum cadence estimation algorithm - in figure 5 the intercept of 164rpm predicts, plausibly, an optimum cadence of 82rpm. Furthermore, data for other, less than maximum rate accelerations was also found to approximate straight lines, albeit less cleanly, with correspondingly lower estimated optimal cadences. This was an early indication that the controller would be capable of adapting to cyclists when they are not fully exerting themselves, despite the less well-defined nature of the corresponding data.

Test data sets of torque/cadence pairs over typical accelerations were taken and used to develop the estimation algorithm, which was then implemented and tested on the prototype bicycle in the following form:

A single torque sample is taken every right crank stroke (for simplicity, and consistency of data) and recorded along with the current cadence. This then updates an array which contains the last eight torque/cadence data pairs. This array is sorted to detect outlying points (bad data) and allow the gradient of the torque/cadence relationship to be approximated using a specially designed regression technique. The new optimal cadence estimation can then be found by extrapolating the line to find the cadence at zero torque, and dividing it by two. The previous target cadence is then updated by averaging it with the new estimation, which is weighted (or rejected) according to the conditioning of data (a better spread will give a better approximation) and how well the data points fit the linear approximation (best if the cyclist is fully and continually exerting himself). Other criteria were imposed to simplify computation, for example the cyclist must be accelerating (as data collected when cycling at constant speed is ill conditioned, while usable data from decelerations is rare).

The new target cadence is then used by a gear selection routine to calculate the range of cadences appropriate to the current gear. It can the shift up or down accordingly if the current cadence is outside the range. Only one gear change is permitted per crank revolution to avoid damaging the actuator.

Results

Although exhaustive testing is beyond the scope of the investigation and unnecessary at this stage of development, data collected over several tests provides convincing evidence that:

- the estimated optimum cadence coincides with the target cadence for best performance
- it is capable of tracking changes in optimal cadence
- the response is stable, robust, consistent and fast, surpassing the early aims of the project

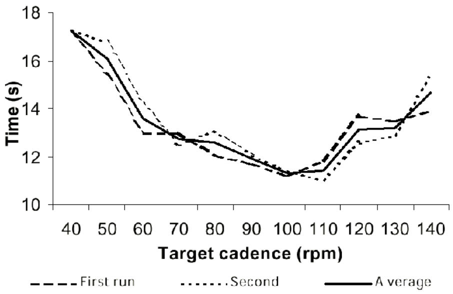


Figure 6: Timed sprints for fixed cadence controllers

Figure 6 was generated by timing sprints, with the controller using a fixed target cadence – 2 times were recorded at each multiple of 10rpm. The graph indicates that the cyclist's performance is maximised if his cadence is kept close to 100rpm. An early version of the estimation algorithm was also running; although the runs were short to avoid tiring the cyclist (<200m), it still regularly estimated optimum cadences close to the actual optimum of 100.

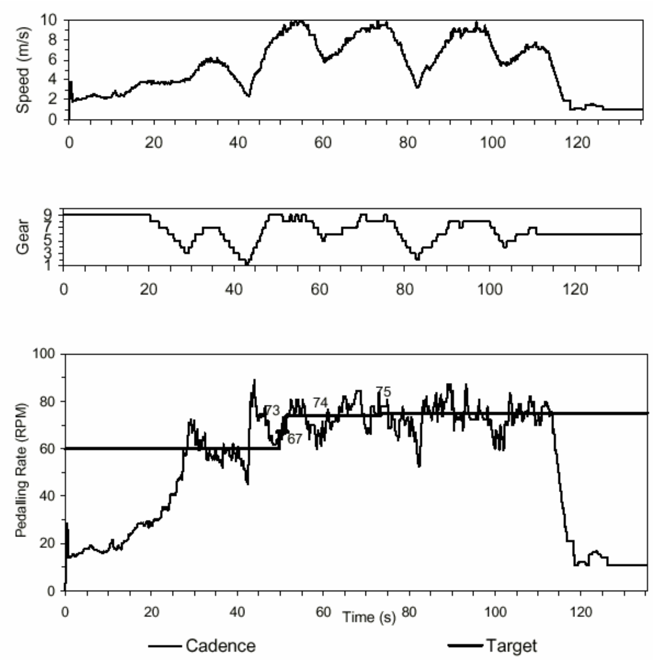


Figure 7. Evolution of estimated optimum cadence over a test run

Figures 7 and 8 are examples of logged data, independently taken for test runs using the estimation algorithm. The data was recorded over urban courses, providing rich information for the controller due to frequent accelerations after corners and junctions so that its behaviour could be observed. In figure 7, the cyclist is fully exerting himself for the duration of the run, and is therefore exhibiting a well defined and relatively constant power-cadence relationship. The target cadence starts at 60 by default, but converges smoothly and rapidly to 74 during the first hard acceleration. The single minor update to 75 during the next acceleration is further evidence of convergence. In addition it suggests not only that the controller is capable of extracting information from further accelerations but that the estimated optimum found is close to the previous estimate, which was calculated from an independent data set. This implies a degree of consistency in both controller and cyclist behaviour. The reasonably high estimated optimum of around 75rpm corresponds to the cyclist's effort level.

Figure 8 was recorded by a cyclist who was not fully exerting himself, yet the target cadence still converges rapidly over each acceleration, from the initial value of 80 (the flat data between 33 and 66s was caused by the chain coming off!). The final, maximal exertion sprint provides the clean torque data to allow the target cadence to jump rapidly from 64 to 94rpm. Although the investigation was only intended to demonstrate the ability of a controller to converge on an optimum slowly, assuming continuous exertion and slow rate of change of ω_{OPT} due to fatigue, even this simple incarnation backs up the earlier suggestion that the controller would be capable of tracking ω_{OPT} at sub-maximal effort.

Other sets of test data over different conditions show similar characteristics; no irregularities were ever observed in the target cadence found, but the fast response means that there is scope for further filtering to reduce sensitivity if desired.

In the opinion of the users, the gears selected by the controller felt natural and comfortable, especially as the controller shifted more often than they would normally bother with. By contrast, if the target cadence of the controller was fixed,

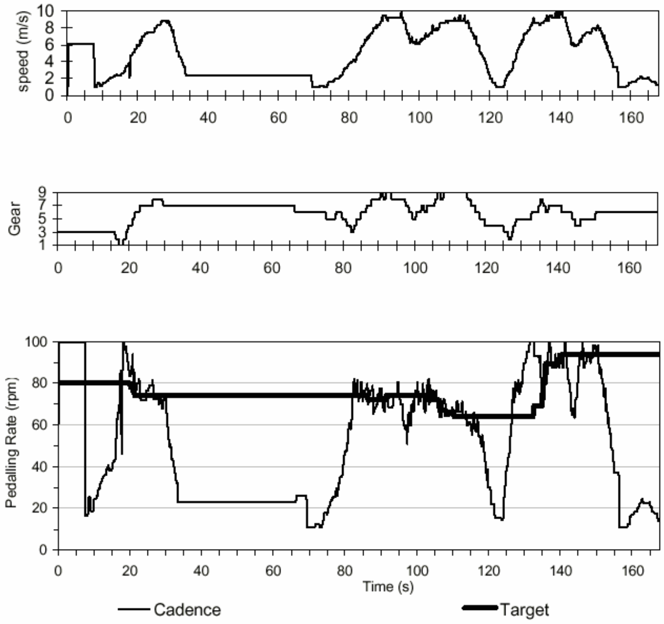


Figure 8. Evolution of estimated optimum cadence over another test run

users found the enforced cadences to be restrictive, forcing them to cycle more leisurely or aggressively than they desired.

Discussion

An analytic approach to optimising automatic gear control on HPVs has been investigated, developed and demonstrated. The resulting controller is capable of adapting to the changing rider characteristics despite requiring no prior training or setting up; the user does not have to intervene or even be aware of the controller's operation. Although a thorough evaluation is not possible at this stage due to the nature of the prototype transmission and the amount of variation in characteristics that would have to be accounted for, the investigation provides convincing support for the theory on which it is based. This suggests that the algorithm is indeed worthy of further development for more suitable transmissions.

The parabolic power / linear torque model used is surprisingly simple and effective, and lends itself well to the two stage optimum cadence estimation and gear selection algorithm. Other approaches based on the same theory are possible – for example a controller could aim to minimise steps in the torque applied at the wheel over gear changes. However the approach used has the advantage of also using the model to filter the input data, at the linear regression stage. Success with the analytic approach meant it is was not necessary to resort to common learning techniques (such as neural networks or fuzzy logic), however for example fuzzy rules could be used to add heuristics, such as a cost associated with changing gear.

Gear selection based on prior knowledge of theory and accurate measurements, rather than the cyclist's own preferences, habits and perception, should result in performance gains. In the case of this project that means faster acceleration, and although a direct comparison is not possible due to the crash transmission, the results obtained are a strong indication that the advantages of automatic gear selection can outweigh the disadvantages for cyclists at all levels. The system has great potential for further development to achieve this: the control software uses only a fraction of the simple 8-bit processor's time, so there is much scope for increasing the complexity and flexibility of the controller. The main unit and torque sensor can be made very light, especially if the torque sensor is moved to the crank itself and the actuator is made more efficient to reduce battery requirements (the average current at the moment is a few milliamps). Cadence and crank position can also be determined from the torque variation. Weight is also saved in other areas due to the removal of the mechanical derailleur cable, shifter and return spring. Having more precise control of the derailleur position and shift timing may allow the crash transmission to be adapted for faster, continuous-torque gear changes. Furthermore, the more frequent and correct gear selection may mean that fewer gears are required, possibly with a single front chainring to avoid extra automation in the case of the crash transmission.

The optimum cadence estimator could find other uses which do not depend on developing an appropriate transmission. For example it could be the basis of a training aid, which would resemble a cycle computer with a remote crank-mounted torque sensor, that could tell a cyclist which gear to be in at any point as well as recording his optimum cadence variation over time.

Conclusion

This investigation has resulted in the development of simple basis for electronic gear selection which merits further investigation and has the potential to benefit riders of even the highest standards. At the very least, it has enlightened the author on the nature of cycling in general and improved his own gear selection greatly.

References

Whitt, Frank R, Wilson, David G. 1974. Bicycling Science. MIT Press, Boston

Kyle, Chester R. 1995 The Browning Automatic Bicycle Transmission. Cycling Science Winter 1995

The author

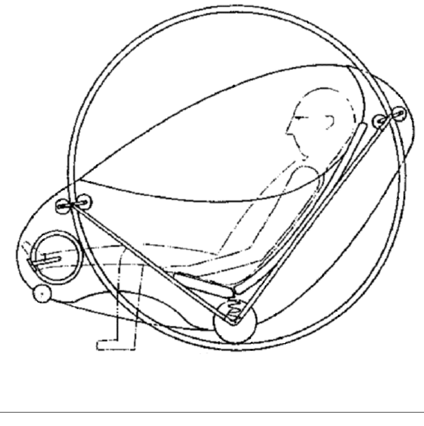
Iain Crouch is a mountain biker from Glasgow, Scotland. He has recently completed an MEng in Engineering Science at Oxford University and can be reached at: <iaincrouch@hotmail.com>

Letter: recumbent monocycle

I have been searching for pics and/or drawings of a pedal powered monocycle. (you ride inside the wheel...usually four to five feet in diameter.) I would like to know if you have any info/links or other news. My next project will be a pedal boat using a hydrobike drive unit. I recently raced a full size one for three miles and came in halfway in the pack amoungst kayakers and canoes. This is a resounding success given my previous last place finishes with a home built chain-drive unit! A picture of what I want to build can be seen at: <http://www.valedo.com/5370410.gif>

Brian Burgess
<pedalyurassoff@peoplepc.com>

[Any reader with more info please contact Brian Burgess.. His link is pictured below. Ed.]



Efficiency Measurements of Bicycle Transmissions - a neverending Story?

by Bernhard Rohloff and Peter Greb (translated by Thomas Siemann)

In *Human Power 52 (Summer 2001)* there was a report of an efficiency test of bicycle transmissions "The Mechanical Efficiency of Bicycle Derailleur and Hub-Gear Transmissions" by Chester. Kyle PhD, and Frank Berto. The test included three derailleur systems with from 4 to 27 gears as well as eight gear hub transmissions with from 3 to 14 gears. The results of the test are summarized in the table below:

Table 1: Derailleur and gearhub transmission efficiencies measured by Kyle and Berto.

Transmission Type:	Efficiency (%)
Derailleurs	87-97
Gear Hubs	86-95
Note: Test performed with 80W, 150W, 200W input.	

The motivated reader of the report will find contradictions behind their measurements. Our specific interest in giving a critique of the publication is based on differences of the results compared to our efficiency measurements. These can be summarized as follows:

Table 2: Derailleur and SPEEDHUB 500/14 efficiencies as measured by Rohloff.

Transmission Type:	Efficiency (%)
Derailleurs	95-98.5
SPEEDHUB 500/14	95-98.5
Note: Test performed with 400W input.	

The lower range of Kyle and Berto's measurements are up to eight percent lower than those made by Rohloff. The reasons for this are presented in this document.

1. Verifiability - The text does not say if only single measurements were performed or if the measurements were confirmed by repeated measurements. Furthermore, there is no information about the duration of break-in time the testing samples underwent. This is especially important for hubs with dragging seals which need a minimum run-in time in order to level off friction losses from the seals. Rohloff has determined this to be extremely important for tests under 200W; this will be discussed later in this document.

2. Precision of measurement – The results are shown as absolute values with no information about the tolerances of the measurements. Only the precision of the dynamometer and the tachometer were given without any information about the width of the measuring range and the related tolerance variations. The ergometer wheel produced variable losses of over 2% with different loads. The ergometer wheel losses at different speeds were not measured. However this is important when evaluating transmissions with a large range of gears such as the 27 speed derailleur system or the Rohloff SPEEDHUB 500/14, because the speed differences between the smallest and largest gear are more than 500%.

3. Plausibility - The report regarding the gearhubs states correctly that the efficiency of planetary gear systems drops as the number of active gears increases. This fact should reflect itself in the measurements of the efficiency of the gearhubs.

The speed-ratio of the Sachs three speed hub is reducing in gear one, increasing in gear three, and direct drive in gear two. Unlike gears one and three, there shouldn't be any gearing losses in gear two. At 80W the measured efficiency of gear two is much lower than those of gear one and gear three, which is not evidently plausible. At 200W the results are very similar in all three gears with efficiency values of 94.1%, 94.9%, and 94.1% for gears one, two, and three respectively.

The trend shown that the efficiency of the SPEEDHUB 500/14 drops in higher gears is also in contrast to the design of the transmission as well as the fact that gear four and gear nine are more efficient than the direct drive gear eleven. In gear eleven there cannot be any gearing losses since no planetary gears are rotating, unlike in all other gears. As can be seen in Table 3, in the first seven gears there is always one more gearset active than in the higher seven gears. Therefore, the losses in the higher seven gears must be smaller than the losses in the lower seven gears.

Table 3: Active gear sets in the Rohloff SPEEDHUB 500/14 for each gear.

Gear	1	2	3	4	5	6	7	8	9	10	11	12	13	14
No. of active gearsets	2	2	3	1	3	2	2	1	1	2	0	2	1	1

There are three planetary gear sets linked in series in the SPEEDHUB 500/14. The single gears are the result of different combinations of gears within those three gearsets.

The efficiency variations of the Shimano four speed, Shimano seven speed, and Sturmey-Archer 7 speed transmissions do reflect the construction of the transmission.

4. Validity – The validity of the testing method. The measurements made by Kyle and Berto were performed while applying constant torque with power input at 80W, 150W, and 200W. Those loads were meant to reflect a typical bicycling situation. Rohloff does not believe that the loads or the power applied sufficiently model a typical cyclist. The power produced by the cyclist consists of a relatively constant speed and widely variable torque due to the crank kinematics. Measurements show that while speed variations of about 5% are typical, the torque variations can be over 90% throughout a single crank revolution. Table 4 shows the results at different power inputs.

Table 4: Maximum and minimum torque measurements during a pedal stroke.

Power input (W), Speed (rev/min)	100 W, 75 rpm	300 W, 75 rpm	575 W, 50 rpm
Max. Torque (N·m)	21.6	68	200
Min. Torque (N·m)	3.8	8	20

The power characteristics are largely governed by the torque component.

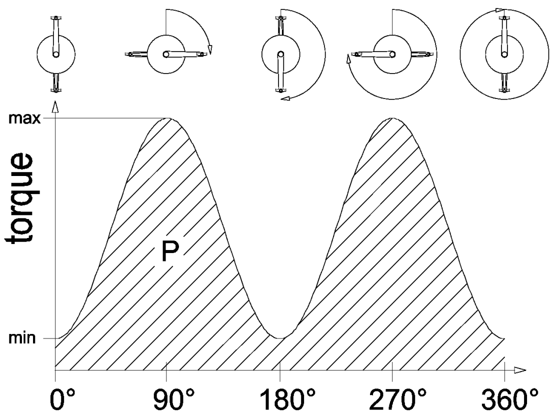


Figure 1: Torque vs. crank angle for one crank revolution.
Power is area under curve.

The cyclical torque of a cyclist produces an alternating load situation on all power transmitting parts, chainlinks, chainrings, bearings, gears, etc, which is very important to keep in mind when evaluating the mechanical losses which effect the efficiency.

A precise simulation of the cyclical torque is not easy to produce in the laboratory and from a measuring point of view, excessively costly. For this reason electric motors with a constant power input are used. This brings up the question of how to choose the appropriate power input when using a constant torque so that the efficiency measurement correlates to the efficiency that would be measured with the cyclical load actually applied in the real world.

We encountered a similar problem when designing our chain and chainring wear test, which is operated at constant torque. Extensive comparisons between components used in real world and components worn out on the test bench showed the following: If the field-tested components were used at an average of 150 W with an average cyclic torque between 5 Nm and 30 Nm, this correlated to a chain tested at a constant torque of 30 Nm in our laboratory.

It can be assumed that the reasons that cause the wear of components are the same ones that are responsible for the efficiency. Therefore you can deduce from the comparisons that a in a lab test, a constant power input using the maximum value of the cyclic load produces results that are closer to reality than choosing a constant power input using the average load.

For example, an average cycling power 80W in real life should be simulated by a test bench power of 160W at the same speed.

5. Interpretation of the measurements – In order to give a correct interpretation of the results it is important to establish what the losses are composed of.

Losses are created by friction. The value is determined by the type of friction (rolling or sliding), the size of surfaces in contact, type of surface finish, material hardness, lubrication, combination of the rubbing parts. Two separate types of losses exist in bicycle transmissions:

A) Power dependent losses. These are created by friction of parts that are moving under a driving load, i.e. chainlinks, gears, bearings, etc. The quantity of the loss grows proportionally to the transmitted power.

B) Power independent losses. These losses are created by friction of moving parts and are not changed by the driving load, in other words these losses are constant regardless of the load applied, e.g. Gaskets and shims. With lubricants, the quantity of loss depends on speed, temperature, and lubricant viscosity.

In the following example, two bicycle transmission systems are compared. Both have a 91% efficiency at 50W input. They have two different power dependent and power independent losses.

In system A, seven percent of the input power is lost due to power dependent friction plus one Watt of power independent friction for each value of input power. The values shown in Table 5 are input powers from 50 W to 500 W with their respective efficiency ranging from 91%-92.8%.

Table 5: System 'A' power loss components.

Input Power (W)	50	100	200	300	400	500
Power dependent losses (7%) (W)	3.5	7	14	21	28	35
Power independet losses (W)	1	1	1	1	1	1
Total loss (W)	4.5	8	15	22	29	36
Total efficiency, (%)	91	92	92.5	92.7	92.75	92.8

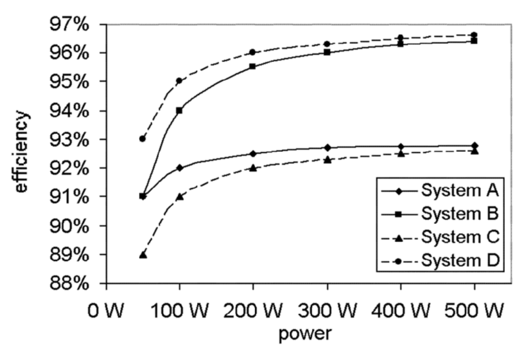
Table 6: System 'B' power loss components.

Input Power (W)	50	100	200	300	400	500
Power dependent losses (3%) (W)	1.5	3	6	9	12	15
Power independent losses (W)	3	3	3	3	3	3
Total loss (W)	4.5	6	9	12	15	18
Total efficieny (%)	91	94	96	96	96.3	96.4

In system B, only three percent of the input power is lost due to power dependent friction that exists in the chain, gears, etc. An additional 3 W of power is lost due to power independent friction that exists due to tight seals.

At 50 W power input the efficiency of system 'B' is at 91%, the same as system 'A.' At higher power inputs, the overall efficiency increases until it reaches 96.4% the efficiency is significantly higher than the efficiency of system 'A.' This is due to the fact that the power dependent losses become dominant over the power independent losses at higher power inputs.

Figure 2: Total efficiency of system A through D.



In addition to curves for systems A and B, Figure 2 also shows curves for systems C and D. The curve for system C describes how the power independent losses increase from one to two Watts due to temperature or lubricating film changes at the seal of system A. The curve for system D describes the efficiency changes of system B with a reduction from three to two Watts of the power independent loss for the same reason. The examples show that for power input of less than 200 W that even small changes of +/- 1W of power independent losses play a large role in the overall efficiency. Since power independent losses are the result of a complex relationship between speed changes, temperature changes (created by own friction heating), and lubrication. These variations can occur in the test situation. If power input is less than 200 W, it must be confirmed that the influence of those variations are verified by repeated tests. Over 200 W the influence of power independent losses can be neglected.

Knowing that, all measurement values shouldn't be absolute values, but rather represented as a range of values showing the corresponding upper and lower boundaries.

6. Reason for efficiency measurements – The reason for efficiency measurements is to find out which one of the different bicycle transmissions converts the most of the bicyclist's power into forward motion. To propel the rider forward in the most efficient manner, it is important that the rider be able to choose an appropriate gear for the given load or riding situation, a gear that is suitable to the rider's fitness level.

The development of power in the muscles is subject to a grade of efficiency. This efficiency is the ratio of metabolic capacity and the delivered mechanical power, i.e. the power at the crank. The efficiency depends on the muscle power combined with the speed of movement, if both variables reach their optimum, the muscle efficiency can increase by 25%. [See also article by Too and Landwer in this issue. Ed.]

The differences in muscle efficiency between positive and negative fatigue ratios (bodily stress/developed power) can easily vary by 10%. This is of much larger value than the variation of mechanical efficiencies of various bicycle transmissions systems.

Table 7: Comparison of muscle and mechanical efficiency of the bicycle-rider system.

	Rider A	Rider B
Muscle Efficiency (%)	24	22
Transmission Efficiency (%)	93	97
Overall Efficiency (%)	**22**	**21**

Rider A is using a perfect gear ratio for the situation and his muscle efficiency is 24%. His bicycle transmission is moving in a gear with relatively poor mechanical efficiency of 93%. Rider B is using an unfavorable gear with a high efficiency of 97%., however, because of the unfavorable speed, his muscles work at 22% efficiency. The overall efficiency shows taking into consideration muscle and transmission losses that rider A is riding more efficiently even though his transmission efficiency is lower than rider B's.

In order to use the rider as a "bicycle engine" most effectively, the ratio increments between the gears are as important as a good mechanical efficiency. The most efficient energy conversion is very limited using transmissions with only a few gears. A larger selection of gears with smaller increments make a favorable energy conversion possible in a wider range of riding situations, but only if the correct gear is used. Sport medical research shows that the increments between gears must be smaller than 15% to benefit the rider's efficiency.

Under this point of view it does not make sense to compare transmissions with only a few gears, large gaps, and small overall range, with transmissions with many gears, small increments, and a large range of gears. A comparison of different transmission systems should always take the application into consideration.

7. Conclusions

A) All measurements below 200 W need to be evaluated cautiously because the influences of the variations of the power independent losses are high.

B) From a practical point of view changes of efficiency play a major role only when riding above the recreational level i.e. greater than 100W.

C) When comparing transmission systems, gear range and number of gears should be taken into consideration in addition to the efficiency.

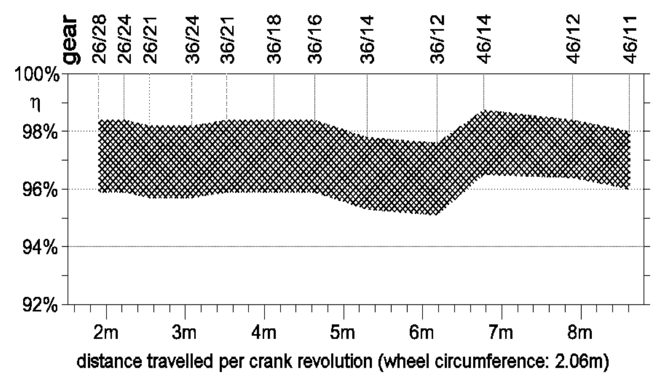
Rohloff measurement results – We would like to point out that the points represented here should be a stimulus for a discussion since there are so many open questions in the field of practical efficiency measurements of bicycle transmission systems. As a comparison to Kyle and Berto's results, our results in Figure 3 and Figure 4 show our efficiency measurements of a 24-speed derailleur system with 46-36-26 toothed chainrings and Shimano XT 11-28 toothed cassette, and the Rohloff SPEEDHUB 500/14 with a primary gearing of 46 tooth chainring and 16 tooth hub cog. Both systems had been run in for 100 km.

The measurements include the losses of the complete transmission, bottom bracket, chain, hubs, etc. In order to simulate a strong rider who applies about 160 W and produces a maximum torque of 50 Nm (285 N @ pedal), the measurements were taken at a power of 314 W with constant torque.

Table 8

crank speed (rev/min)	60
brake power, constant (W)	314
torque (Nm)	50

Figure 3: Efficiency of a 24-speed mountain bike drivetrain.



The reproducibility of the results and their precision was verified by repeated test runs. Figure 3 shows the efficiency of the derailleur system plotted vs. distance per crank revolution. Note the gear ratios are not consistently spaced as can be seen on the plot.

The derailleur system was tested first in clean and well-lubricated conditions. In order to achieve results closer to real-life use, the chain and the sprockets were replaced with components which had been subjected to 1000 km of field use and had not been cleaned. The average efficiency was measured to be 1% lower than the clean drivetrain. The plot in Figure 3. includes the data for a new and used drivetrain and a +/- 0.5% uncertainty.

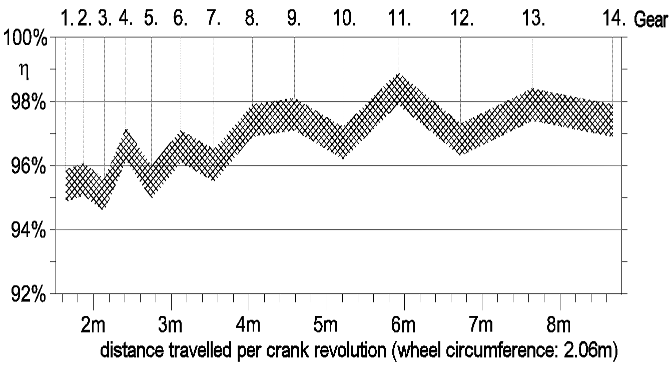


Figure 4: SPEEDHUB 500/14 efficiency

Figure 4 shows the efficiency range of the Rohloff SPEEDHUB 500/14 plotted vs. distance traveled per crank revolution. The increase between all gear ratios on the SPEEDHUB 500/14 is always the same percentage. Sprocket and chain were replaced by components ridden 1000 km. Efficiency differences were not measureable. The range of efficiency represents the used and unused drivetrain plus the +/- 0.5% uncertainty.

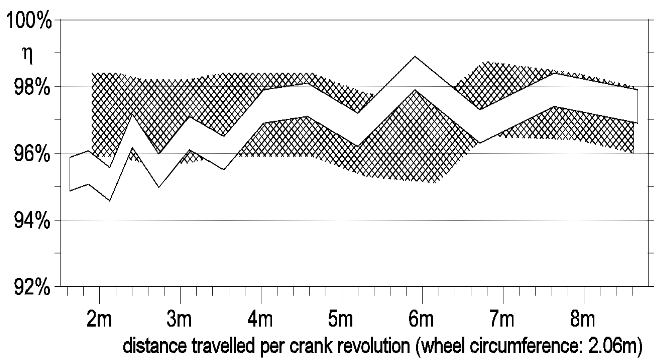


Figure 5: Efficiency of both derailleur drivetrain and Rohloff SPEEDHUB 500/14

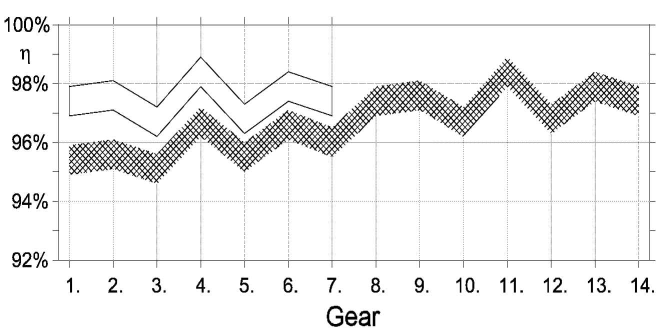


Figure 6: SPEEDHUB 500/14 efficiency comparison.

Gears 8-14 shifted to the left to compare with gears 1-7.

Figure 5 shows the efficiency ranges of Figures 3 and 4 on the same plot for comparison.

The efficiency of internally geared hubs drops when the number of working planetary sets increases. This fact must be shown in the efficiency results of the gear hubs tested. In the SPEEDHUB 500/14 there are three planetary gear sets that can be used in series. The unique gear ratios are created by engaging different combinations gears within these planetary sets. Table 3 shows the number of the active (working) planetary gear sets per gear. Figure 6 shows the range of efficiency of the SPEEDHUB 500/14 plotted vs. gear number. The efficiency plots confirms the number of the active planetary sets as represented in Table 3. Gear 11 has the highest efficiency because it is the direct drive gear, no planetary gearsets are activated. The curve between gear 1 and 7 corresponds with the curve between gear 8 and 14. This is due to the fact that the first two planetary gear sets are shifted between gears 1 and 7 in the same way as they are between gears 8-14, however gears 1-7 have an extra planetary gearset activated providing a compound low gear. The efficiency between gear 1-7 is about 2% lower due to the use of the third planetary gear set. In order to show this fact more clearly the curve between gear 8-14 has been copied and shifted to the left so that it can be compared with the curve representing the efficiencies of gears 1-7. The results correspond to the gear combination or respectively to the number of active planetary gears inside the hub.

Conclusion – The explanations show that efficiency of bicycle transmissions depends on many factors of which exact measurements may involve prohibitive costs. In order to measure real-life values, factors such as contamination, lubrication, wear, and production tolerances should be included as well as sports medical research. We think that there is still a lot of room for tests and discussions.

About the Authors

Bernhard Rohloff and Peter Greb can be reached at:
Rohloff, Moenchebergstrasse 30,
D-34125 Kassel, Germany.
Translator Thomas Siemann is at:
Rohloff USA in Berkeley, CA 94707

Reply from Chester Kyle

Dear Editor,

I have read Rohloff's remarks on our transmission efficiency tests and have several comments on their discussion.

Our tests were run over a two day period. It would have been better to test repeatedly over longer periods, but this was not possible due to limited time and funds. However, we feel that the results are valid under the conditions we tested.

We understand the position of Rohloff, whose transmission did well in our tests when compared to other hub gears, but whose efficiency was about 2% lower than the derailleur transmissions. It's natural for researchers to question the test methods of others when results don't agree with their own. However, the principle reason for the Rohloff's disagreement is the difference in applied power input between the two test methods. We will comment more on this later. Rohloff's laboratory efficiencies were about 2% higher than ours, but this is understandable given their methods.

All of our transmissions were tested and compared under the same conditions. Our test efficiencies were repeatable to within less than one percent over two separate test sessions several months apart. For the conditions we tested under, our methods were sufficiently accurate to discriminate between transmissions and gears and to rank order the efficiency of the transmissions. All of the hub gear transmissions were tested using light oil as a lubricant. However, the Rohloff was new and not worn in before testing. This could have affected the efficiency under low loads, but probably not under loads of 200 watts or more.

We chose to compare all of our transmissions at 200 watts average load or less and at a constant cadence of 75 RPM. Ordinary hub gears are never used in bicycle racing and are seldom even in recreational cycling. They are, however, commonly used on European city commuter bikes where speeds are almost always below 25 km/h. Power requirements for low speed commuting are normally less than 150 watts. 200 watts average power is sufficient to propel a bicycle at over 32 km/h on level ground with no wind. Therefore except in laboratory experiments, hub gears are almost never subjected to the high loads that derailleur transmissions are. Rohloff is correct in saying that efficiency improves as the load increases. They tested at 400 watts, double what we did and found efficiencies approaching 98%. We tested only one transmission at more than 200 watts and found the Shimano derailleur transmission in 25th gear, under loads from 307 to 370 watts input, was about 98% efficient (our Figure 14).

Because of the high inertia of the bicycle rider system, the speed variation due to variable torque (pedal force) at the crank is very small. At racing speeds a computer simulation shows speed variation is less than plus or minus 0.13% due to the variable torque of the crank. We therefore felt that testing at a constant speed of 75 RPM was realistic. Racers pedal at a higher cadence, but the purpose of our tests was to approximate more normal riding conditions.

Simulating variable crank torque is not practical with an electric motor dynamometer and as far as I know, no current or past transmission test apparatus has successfully used this technique. Rolhoff applied a much higher constant torque than our average to simulate maximum chain tension and gear and chain wear, but this also is not realistic. Transmission efficiency varies continuously around the crank cycle - it is high under high torque and lower under low torque. The average efficiency is somewhere in between. Testing only at high torque as Rohloff did, does not give an accurate comparison. Unless transmissions are tested on the road or in the laboratory using a precision research crank dynamometer with an actual cyclist, there is really no certainty which of the laboratory test methods is more valid. Unfortunately highly accurate laboratory crank dynamometer tests have not yet been developed.

To summarize, we are reasonably confident that the rank order between transmission efficiencies that we found would not change appreciably as load is varied within a normal range. In other words, transmissions should rank about the same at either low or high loads. We feel that the loads we tested under are typical of the actual conditions under which hub gears are used and represent a reasonable average efficiency. In our article we therefore concluded that hub gears are about 2% less efficient that derailleur transmissions under typical field conditions. We see no reason to change that conclusion.

The Rohloff is an excellent transmission - in fact it is quite elegant in its function - it shifts sequentially from gear 1 through gear 14 easily and logically - unlike triple chainring derailleur transmissions. The Rohloff would probably serve well for HPV racing since it would much simplify the chain line.

– *Chester Kyle*

[Ed. Comment, also applying to the article by Vernon Forbes on the next page: I never cease to be amazed at the extremely high torques standard hub gears will stand without failing even when used in very heavy and sometimes powered vehicles, such as the 550-1400kg Thuner Trampelwurm (described in HP54), or my Velocity Dolphin electric bicycle with a normal hub gear taking up both the torque from a 250 W electric motor and from a 24 speed derailleur drive, or various other electric vehicles.]

ERRATA FOR HUMAN POWER NUMBER 54, SPRING 2003

Page 6, Eq. 11:

„ρ“ should be „**p**“

Page 6, Fig. 3:

Re = pV/(R m T) should be:
Re = **L**pV/(R m T)

Page 23: first column, lines 16-17:

...seen in figure **5** (not 7), the combination I=1.06 and G=**2.0** (not 3.8) is optimal.

Page 23: second column, lines 27-29:

..G at **2.0** (not 3.8) and I at 1.06 kgm^2, however lowering G to **1.5** (not 2.85) or even **1.14** (not 2.17) may result in a reasonable compromise...

ANNOUNCEMENT

A new association, tentatively known as the Human Power Institute (HuPI), has been formed in order to promote the development and use of human power for an environmentally sustainable and socially just society. Launched in January, 2004, HuPI seeks to establish a website information database and foster the international exchange of information amoung all parties interested in the technologies and benefits of human power. HuPI has primarily a virtual presence on the internet as the most economic means of making information and resources available worldwide.

HuPI is to be a locus for research and development in all areas of human power in a scientific and engineering context. Much of this work is technological in nature and has to do with specific tasks, such as the design of machines for transport. As well, HuPI is devoted to exploring and understanding how human power technology benefits society across a wide range of areas, including economics, agriculture, social rubric, psychology, and general well being.

HuPI´s first project is initiated and sponsored by Dave Wilson, editor of Human Power for 18 years. He wishes to make the wealth of information in previous issues of Human Power more easily accessible, and to this end commissioned a compilation of all issues in the PDF format, complete with searchable index. This archive is to be made available on the IHPVA website and on a CD-ROM, which will be available for sale at nominal prices from some IHPVA member associations, in particular the HPVA.

In mid-2004, HuPI plans to start the Human Power International Journal, a web-based open electronic journal. Initially editted by Theo Schmidt, HPIJ will be available for free via the HuPI website: **`http://www.hupi.org`** which is also the primary contact to HuPI.

Why was HuPI formed? The IHPVA and its members are concentrating on HPV racing, records and events. HuPI wishes to complement that worthy endeavor with readily available internet-based information to help foster a greater application of human power in daily life.

Founders of HuPI are: Richard Ballantine, Theo Schmidt, John Snyder, Elrey John Stephens, Brian Wilson, David Gordon Wilson.

Human Powered Helicopter, a retrospective

by William B. Patterson

Introduction

The Igor Sikorsky prize was offered by the American Helicopter Society in 1979. Its purpose was to encourage student interest in helicopters. The prize has been wildly successful. Numerous flight attempts have been made. Each one is the center of intense thought and effort by many students.

The two recognized hovers will be discussed. Also, we can observe the dead-end design that has left most attempts ground-bound over the last quarter-century. Finally, an additional set of rules will be proposed. These rules will allow easy comparison between the next generation of HPH flights.

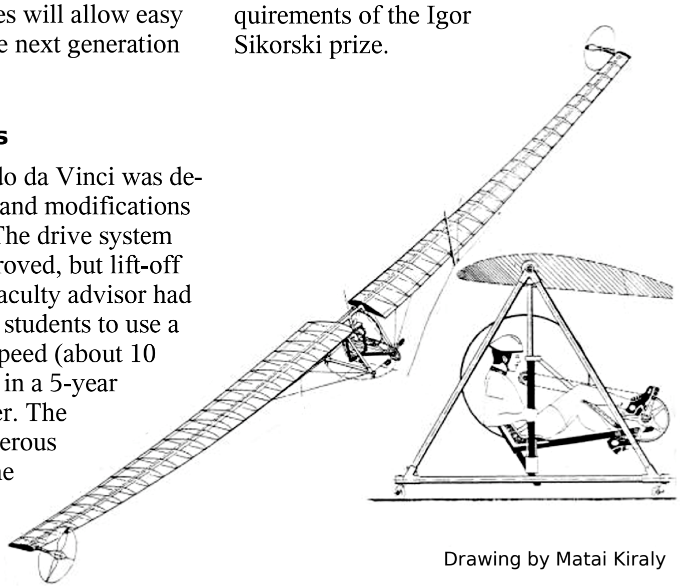
Recognized Hovers

In 1981 the Leonardo da Vinci was designed. Flight attempts and modifications continued for 4 years. The drive system and structure were improved, but lift-off was not possible. The faculty advisor had mistakenly directed the students to use a 30 feet per second tip speed (about 10 m/s). His error resulted in a 5-year delay in achieving hover. The students contacted numerous engineers throughout the country to optimize tip speed and make other improvements to the design.

The Leonardo da Vinci II was designed in 1986 with a more efficient tip speed and various improvements. It achieved a partial lift off but immediately crashed. The Leonardo da Vince III was designed in 1988 with more attention to rotor in-plane buckling and flew for a short time. It was the first human-powered hover in the history of the human race. Future plans were to reduce weight and increase stability. (4).

A Japanese team at Nihon University hovered the Yuri III in 1994. Picture and data see (1). Dr Naito had tried various configurations over a 10-year period before trying a multiple rotor system. These are the only recognized hovers.

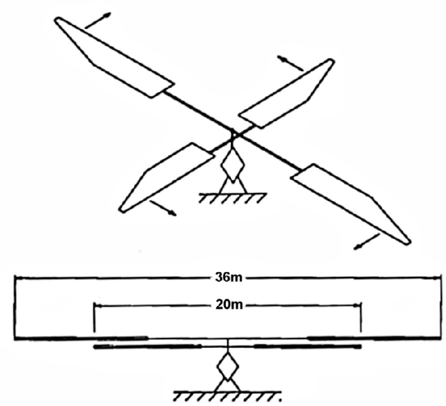
The da Vinci flew because it had a very large rotor and was light for it's size. The Yuri flew because the multiple rotors reduced the load on each system and the system was very light. The flight of the Yuri also pointed out the flight profile of a successful prizewinner. The winning design team will need to have a machine that can act as a hovercraft for say 50 seconds, and as a helicopter for say 10 seconds, in order to achieve the 1 minute hover duration (any height) and 3 meter height (any duration) requirements of the Igor Sikorski prize.



Drawing by Matai Kiraly

Figures 1 and 2 **Helicopter "Leonardo da Vinci II"**

Figures 3 and 4 from internet sites



After the Yuri: the Dead End

The first ever HPH was a coaxial system built in Oregon before the Sikorsky prize was offered. The first HPH that the author observed was a coaxial system at Westland Helicopters in 1981. Since those helicopters were constructed, most attempts have been configured as coaxial systems. Two teams are currently building such systems in Canada.

The great majority of HPH flight attempts have been made with the coaxial configuration. They have all been doomed to failure. See (2) for early Japanese configurations of a HPH which did not fly.

The design consideration for a HPH is different from high power machines. The rotor blades are the heaviest part of the craft. The number of blades should be reduced to the bare minimum. The CALPOLY team seriously tried to design a single rotor helicopter, similar to windmills on the Baltic coast.

The blades are also moving very slowly, so we get no centripetal stiffening. They are light and flexible; the airflow near the rotors will impinge on the others causing unacceptable dynamic reactions.

Efficiency is lost when force is concentrated rather than dispersed. A powered hub will lose power and will add to the weight. Both the Yuri and the da Vinci used thread to transfer power from the pilot to the rotor system. Future design teams should be warned of the pitfalls in the coaxial system.

We see that the coaxial systems have failed because they have too many rotor blades for too little disk area. The rotor wash causes unacceptable dynamic problems and the drive system is too heavy and inefficient.

Suggested competition rules.

The current rules are set up as an either/or situation, but need not be changed. A new set of rules may foster more competition by providing a method of scoring various HPH flights.

The induced flow field around a helicopter is based on Z/R, where Z is the altitude of the aerodynamic surfaces and R is the rotor radius.

For the purposes of a human powered helicopter, Z should be defined as the lowest portion of any aerodynamic surface, including the rotor, fins or skirt if used.

Current rules specify that the HPH must stay within a 10-meter box. This rule supposes that the machine is controlled. All human powered flights have been with uncontrolled machines. Therefore, time should stop when the machine touches the ground or when it has translated 5 meters from the take-off position.

Score should be given as follows:

- 1 point is given for each second off the ground while the machine remains within 5 meters of the lift off position.
- 1 point is given for each 5 cm of Z.
- Human powered helicopter recognition must not be given for a hovercraft. A hovercraft in this case, is defined as a machine which generates negligible induced flow. A true helicopter should reach a minimum Z of perhaps 0.5 meters for a small period of time.

Induced power OGE is:

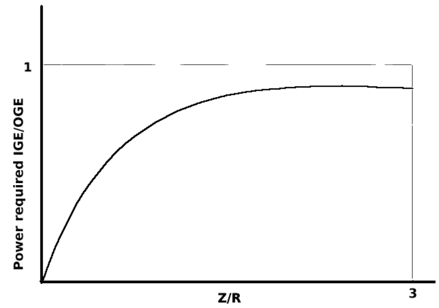
$$\text{Power OGE} = W^{3/2} / (2\rho\pi R^2)^{1/2}$$

W weight
ρ air density
π pi
R rotor radius

See Ref. 3.

The induced power is reduced to zero as Z is reduced to zero.

Future judges must take this information into account when observing flight attempts.



References.

1. D.G. Wilson. Editorial Human Power Spring-summer 1994
2. Akira Naito. “Review of developments in Human Powered Helicopters” Human Power summer 1991.
3. Gessow & Meyers “Aerodynamic of the Helicopter” Ungar 1952
4. Totah & Patterson “Control of a Human Powered Helicopter in a hover” NASA Tech Memo 101029 Nov 1988

About the author

Bill Patterson is a Mechanical Engineering Prof. Emeritus at CALPOLY, San Luis Obispo, California 93407 USA

Letter from Chris Roper

Bill Patterson remains forever respected by me as the designer of the world`s first human powered helicopter.

However, I cannot extend this respect to his rules in their current proposed form.

I don`t like the definition of Z as the lowest point of any aerodynamic surface. You could argue forever about what is and what isn`t an aerodynamic surface.

There is another point regarding this Z. His own Da Vinci had the rotors not at the lowest of the craft, whereas the Yuri did. So by using his Z rather than absolute altitude gained, which of them is shown the more favourably?

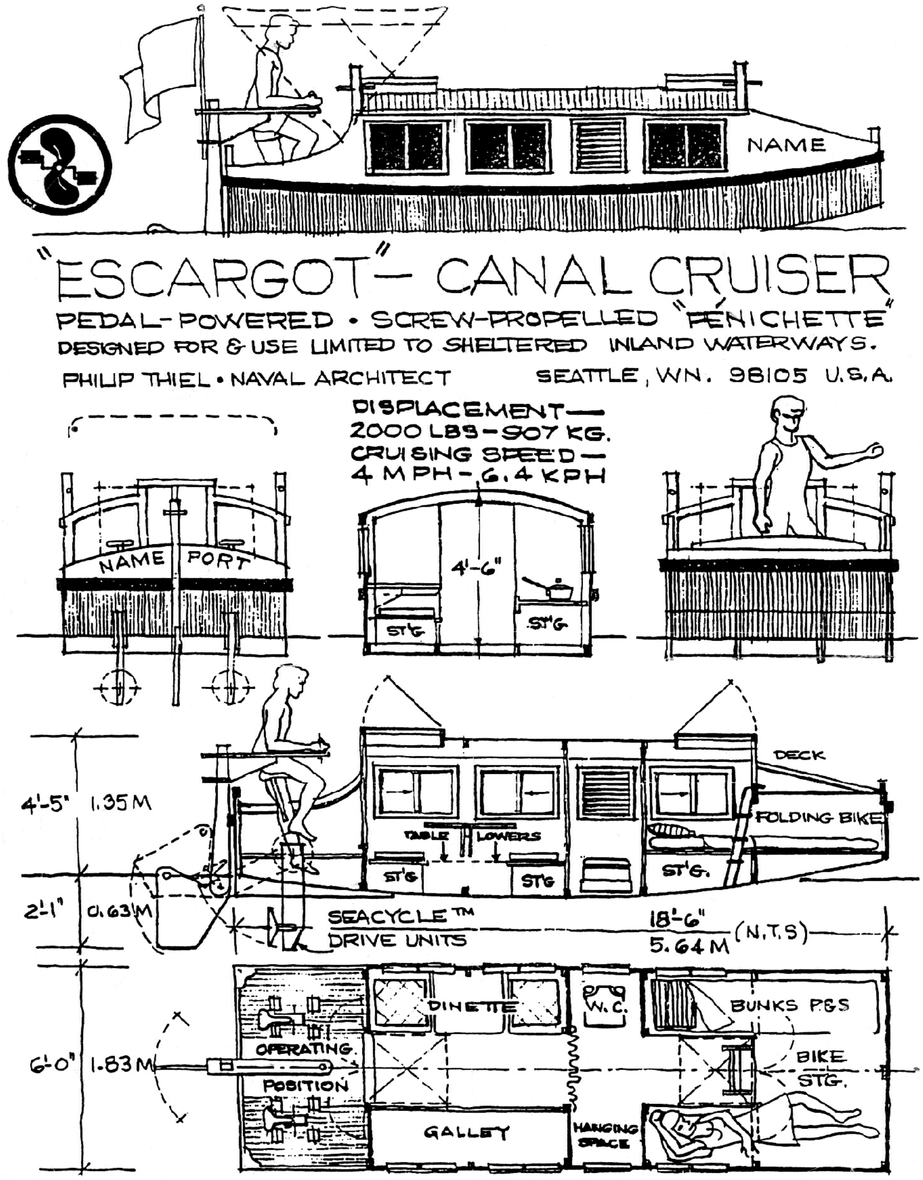
Surprise, surprise, the Da Vinci .

Maybe, with this, and possibly other slight modifications, his rules could become recognised.

I would also like to comment on the proposed exclusion of "hovercraft", but feel this to be inappropriate as I am a hovercraft owner myself.

Chris Roper, HPVA Vice President for Air

Green Fleet - human-powered houseboats on the Ruhr



Some years ago, Human Power associate editor *Philip Thiel* from Seattle designed the "Escargot", a tiny pedal-powered houseboat to sleep two or three. I was fortunate enough to be able to try out the first one built on the Norfolk Broads in England. A loveable houseboat, powered by two "Sea-Cycle" screw-propeller drives. These drives, which are configured for rather light boats, weren't ideal for the heavier Escargot and made for heavy pedalling at low revolutions. In the meantime, more Escargots have been built and used in the German Ruhr area. There should be a total of 5 craft for the 2004 season!

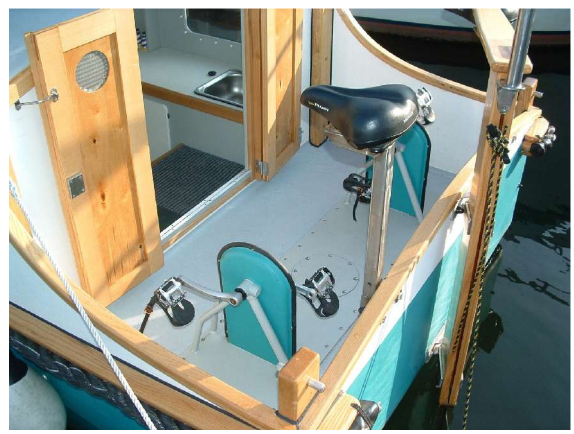
This new fleet is built to Phil Thiel's orginal plans, but uses custom built pedal drive units. The houseboats are constructed and operated by the Hesse Boatyard in Mühlheim, who let them out for charter and also operate a floating boat-café on the river Ruhr.

The Escargots are also fitted with a small electric drive which can be charged up by a small solar panel, although this is too small to provide direct solar propulsion. The unconventional craft are licenced to carry 6 adults and have up to 3 bunks. They are available from April to October, a three-day charter costing EURO 230.-. When it is cool, the small cabin can be heated, using an arrangement of 8 tea-candles!

A conversation with Ms. Hesse supplied interesting facts about the reaction of the public to the boats, e.g. it seems that the Escargots are especially popular with women, children and families, a complete contrast to the usual huge motor boats usually chartered by groups of men! The Hesse's success-story is also partially due to support by the local government, which is promoting low-impact tourism in this area formerly more known for coal and steel industries. [ts]

For further information contact:

Bootswerft Hesse
Hafenstrasse 15
D - 45478 Mühlheim/Ruhr, Germany
email: `boote-muehlheim@web.de`
web: `www.gruene-flotte.de`



More pictures of the Escargot, showing the interior with sitting headroom, the twin pedal-drive units and the propeller they are coupled to, and also the electric outboard drive incorporated into the rudder.

Continued from page 20:

Annotated Bibliography

Berto, F. and Kyle, C. 2001.
HUMAN POWER, #52, pp. 3-11. Though other articles have appeared measuring the efficiency of epicyclic gearing, this is possibly the most carefully done.

Berto, F., Shepherd, R., Henry, R. 2000.
THE DANCING CHAIN, Self-pub., San Francisco. The definitive text on derailleur design and history. Imprimatur.

Hadland, T. 1987.
THE STURMEY-ARCHER STORY, Self pub., s.l. The definitive text on Sturmey-Archer hub gear design and history. Nihil Obstat. Imprim.

Hadland, T. 2000.
"Raleigh in the last quarter of the 20th century", 11th International Cycle History Conference Proceedings, A. Ritchie & R. van der Plas, Eds. Van der Plas publications, San Francisco. This chronicles the confusing business dealings that led to Raleigh's leaving England and Sturmey-Archer's closing. An unabridged version is at: http://www.users.globalnet.co.uk/~hadland/#raleigh

van der Plas, R. 1991.
BICYCLE TECHNOLOGY. Bicycle Books. San Francis, CA

Read, P. M.
STURMEY-ARCHER: "THE END" AND "NEW BEGINNINGS"; (1902-2000 onwards) hub gear drawings, diagrams and parts lists, Fourth Ed., (1997), Self pub., Milton-Keynes. Peter Read's massive tome is the definitive guide to repair each S-A design with many small changes carefully documented. It includes the many aftermarket improvements that Sturmey-Archer cognoscenti have developed over the years.

International Human Powered Vehicle Association

IHPVA

PO Box 1307

San Luis Obispo, CA 93406 USA

http://www.ihpva.org