A Practical Analysis of Unsteady Flow Around a Bicycle Wheel, Fork and Partial Frame Using CFD

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Abstract

CFD is used to study air flow around a rotating bicycle wheel in contact with the ground, extending previous ‘wheel-only’ work on this problem by including the fork, head tube, top tube, down tube, caliper and brake pads. Unsteady simulations, using a Delayed Detached Eddy Simulation (DDES) turbulence model, were run for 9 different wheel and front fork configurations, over 10 different operating conditions (5 yaw angles, repeated for two different speeds, commonly encountered by cyclists), resulting in 90 transient design points. A novel co-processing approach and a new automated concurrent postprocessing methodology is introduced here to make the task of studying many transient cases practical. These developments led to a greater than 30-fold reduction in disk storage requirements, and up to a 20-fold reduction in the postprocessing time on a per time step basis. Concurrent postprocessing, implemented with a job scheduling system, permitted as many as 40 simultaneous postprocessing jobs to be run, further reducing the time to complete the data analysis. The contribution of aerodynamic torque to the overall power requirements for rotating wheels in contact with the ground is presented here for the first time. Fundamental performance characteristics for the drag force on the wheel and fork, the turning moment and overall power requirements are reported here. Differences noted for the design configurations studied suggest that the selection of not only the wheel but wheel and fork combination is a critical consideration for amateur and professional cyclists and triathletes. The co-processing and postprocessing methodologies introduced here are expected to have practical relevance to all transient CFD analyses.

Nomenclature

\[
\begin{align*}
C_D &= \text{drag coefficient} = \frac{F_D}{0.5\rho V^2 S} \\
C_s &= \text{side force coefficient} = \frac{F_s}{0.5\rho V^2 S} \\
C_v &= \text{vertical force coefficient} = \frac{F_v}{0.5\rho V^2 S} \\
D &= \text{nominal bicycle wheel diameter} \\
F_D &= \text{axial drag force} \\
F_s &= \text{side (or lift force)} \\
F_v &= \text{vertical force} \\
f &= \text{frequency} \\
M &= \text{torque (aerodynamic)} \\
P &= \text{power} \\
p &= \text{pressure} \\
R &= \text{nominal bicycle wheel radius} \\
S &= \text{reference area}, \pi D^2/4 \\
St &= \text{Strouhal No.} = f D/V \\
T_{\theta x} &= \text{\(\theta\)x shear stress component, used in torque calculation} \\
t^* &= \text{dimensionless time} = tV/D \\
u &= \text{velocity vector} \\
V &= \text{bicycle speed (in direction of travel)} \\
\beta &= \text{yaw angle} \\
\mu &= \text{viscosity} \\
\rho &= \text{density} \\
\omega &= \text{rotational speed}
\end{align*}
\]

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I. Introduction

By closely examining the finishing times of professional cyclists and triathletes over the past few years in several events such as the Tour de France, the Giro D’Italia and the Ironman World Championship, it has been recognized that small performance gains, on the order of roughly 3%, have highly significant relevance. For the professional cyclists, such time gains can mean the difference between standing on the podium or not after many days of competition in a grand Tour event. For professional and amateur triathletes who face strong competition to secure very limited slots at qualifying events, these small differences determine whether they will be able to compete at the World Championship event – a highly coveted goal in this athletic community.

In recent years, as manufacturers of racing bicycles and bicycle components have turned to wind tunnel testing to optimize component design, the athletes themselves are now able to purchase time in wind tunnels to refine and perfect their riding positions. Comprehensive reviews by Burke and Lukes et al. cite many efforts which validate the conventional wisdom that the main contributors to overall drag are the rider, the frame including fork and aerobars, and the wheels. Greenwell et al. have concluded that the drag contribution from the wheels is on the order of 10% to 15% of the total drag, and that with improvements in wheel design, an overall reduction in drag on the order of 2% to 3% is possible. The wind tunnel results of Zdravkovich demonstrated that the addition of simple splitter plates to extend the rim depth of standard wheels was seen to reduce drag by 5%. In the mid 1980’s, commercial bicycle racing wheel manufacturers started to build wheels with increasingly deeper rims having toroidal cross sections. The experimental studies published by Tew and Sayers showed that the newer aerodynamic wheels were able to reduce drag by up to 50% when compared to conventional wheels. Although many wind tunnel tests have now been performed on bicycle wheels, it has been difficult to make direct comparisons owing to variations in wind tunnel configurations and the testing apparatus used to support and rotate the bicycle wheels studied. In the works published by Kyle and Burke it was observed that a very significant reduction in drag was measured for rotating wheels when compared to stationary ones. Although the experimental work of Fackrell and Harvey was applied to study the aerodynamic forces on much wider automobile tires, it is worth noting that they also observed a reduction in drag and side forces when comparing rotating and stationary wheels.

To date, far less work has been done to apply CFD to study the general problem of flow around a rolling wheel in contact with the ground. Wray used RANS with standard k-ε and RNG k-ε turbulence models to explore the effect of increasing yaw angle for flow around a rotating car tire. He observed that the yaw angle was seen to have a strong association with the degree of flow separation occurring on the suction side of the wheel. A shortcoming of this study was the lack of experimental data available for comparison and the authors note this as a direction for future work. McManus et al. used an unsteady Reynolds Averaged Navier Stokes (URANS) approach to validate CFD against the experimental results of Fackrell and Harvey for an isolated rotating car tire in contact with the ground. The one-equation Spalart-Allmaras (S-A) model and a two equation realizable k-ε turbulence model (RKE) were used. The time averaged results from this study showed good qualitative and quantitative agreement with experimental data. The authors suggest that differences between their predictions on the rotating wheel and the wind tunnel results for surface pressures at the line of contact was due to errors in experimental method and not the result of shortcomings of the numerical approach that they employed. The S-A turbulence model was noted to provide results which were in better agreement with the experimental data than the RKE model.

With respect to the problem of using CFD to study the performance of bicycle wheels, we have previously used a novel methodology to validate computed drag results against wind tunnel data for an existing commercially produced bicycle wheel (Zipp 404). Steady and unsteady computations were performed using the commercial solver, AcuSolve, over a range of speeds and yaw angles. We observed a transition
from the expected downward acting force to the upward direction at a yaw angle between 5 and 8 degrees. By breaking the resolved forces down into individual contributions from the tire and rim, hub and spokes, something not possible with wind tunnel testing, we observed this unexpected transition to be limited to just the tire and rim. This work served to demonstrate that the tire and rim were responsible for nearly all of the aerodynamic performance effects observed. We subsequently extended our work to cover six different commercially produced wheels, namely the Rolf Sestriere, HED H3 TriSpoke, the Zipp 404, 808 and 1080 deep rim wheels and the Zipp Sub9 disc wheel. Drag coefficients, $C_D$, calculated from the CFD simulations compared very well with previously published experimental data (see Greenwell et al., Kyle et al., Tew et al., Prasuhn and Kuhnen) and unpublished data provided by Zipp Speed Weaponry, Indiana, USA. Unsteady Delayed Detached Eddy Simulation (DDES) calculations, run at a single critical yaw angle of 10°, supported the general conclusions that periodic shedding was observed, and, seen to be different for each wheel. Further, the calculated Strouhal number was seen to decrease as the rim depth of the wheel increased. The one exception to this observation was for the HED H3 TriSpoke – in this case, the Strouhal number matched the passage of the characteristic three wide, individual spokes.

II. Methodology

A. Wheel, Fork and Frame Geometry

A profile gauge was used to measure the cross sectional profile of the production Zipp 404 and Zipp 1080 wheel (Zipp Speed Weaponry, Indiana, USA) and the HED H3 TriSpoke wheel (HED Cycling Products). Each wheel was modeled with a Continental Tubular tire attached and inflated to 120psi. A hub profile, also measured from the Zipp 404 wheel using the profile gauge, was used for the Zipp1080 wheel as well. The HED H3 TriSpoke wheel hub was measured separately. Spoke cross sections for the Zipp404 and Zipp1080 were based on the commercially available CX-Ray spoke (Sapim Race Spokes, Belgium). The radius of curvature of the leading and trailing edges was estimated.
to be 0.5 mm. Spoke counts and spacing matched the production wheels. The profile gauge was again used to obtain cross sectional representations at several axial locations along two forks studied, namely the Reynolds Full Carbon aero fork (Reynolds Cycling, UT, USA, and the Blackwell Time Bandit (Blackwell Research, Austin TX, USA). Axial cross sections were then combined to create the three dimensional representation for each fork. The headset and head tube geometry were based on Chris King headset (NoThreadSet™, 2005). The top tube had a constant diameter, (30.5 mm) matching that of custom racing bicycle (2005 Elite Razor, Elite Bicycles Inc., Pennsylvania, USA). The down tube cross section was constant along its length and was measured using the profile gauge method. The 72° angle between the top tube and the front fork (and head tube), and the 49° angle between the top tube and down tube matched that of the previously mentioned custom racing bicycle. A notional caliper and set of brake pads were also included for the models containing forks. The final CFD mesh geometries used for each wheel are illustrated in Figure 1.

The geometry of time trial and triathlon bicycles positions much of the weight of the rider directly over the front wheel through the use of aerobars. To obtain an estimate of the contact area between the road and the front tire, a bicycle (2005 Elite Razor, Elite Bicycles Inc., Pennsylvania, USA) was mounted on a training device, a 170lb rider was positioned on the bicycle, and the contact area of the deflected tire was traced. The contact area was standardized as part of the model geometry used in this study. Under actual riding conditions, variations in the road surface would lead to vertical translation of the wheel, thus varying the contact area. It is felt that a ground contact consideration is important, and that a standardized ground contact shape is a reasonable approximation.

B. Grid Development

Geometry and meshing was fully automated through the development of parameterized journals for GAMBIT® and TGrid®. The model domain for the problem was divided into two sub-volumes, with one containing the spokes, hub and inner edge of the wheel rim, and the other containing the remaining toroidal wheel surface, the fork and frame, the ground contact and the surrounding volume. This division is needed to apply a rotational frame of reference to the inner wheel section, permitting realistic movement of the spokes in subsequent transient modeling efforts. A non-conformal interface was used between the inner wheel sub-volume and the surrounding fluid volume. An illustration of the methodology is shown in Figure 2.
The base surface mesh size was consistently maintained for the tire, rims and forks. On the hub, spokes, caliper and brake pads, head tube, top tube and down tube, the surface mesh was identical for all cases. Three prismatic boundary layers of constant thickness (0.025cm per layer) were generated for all geometry component surfaces. Subsequent $y'$ checks on converged RANS solutions were performed to ensure that the prism layer were within the recommended range\textsuperscript{21} for the turbulence models being used. In addition, two rectangular mesh refinement zones were created around each wheel to control the volume mesh near the wheel, fork and frame. The mesh used to study the Zipp 1080 with the Blackwell fork is shown in Figure 3.

![Figure 3: Volume mesh with 'near wheel' refinement zones](image)

A truncated ellipsoid shape was used to represent the far field boundary and is illustrated along with the boundary conditions described in the next section in Figure 4. Subsequent checks on converged RANS solutions were performed to ensure that the far field and downstream extents were sufficient to resolve pressures at these domain boundaries. For this work, baseline computational grids, for each of the nine cases illustrated in Figure 1, contained approximately 6 to 10 million tetrahedral elements.

![Figure 4: Boundary conditions and model position within the domain. Side view upper figure; top view lower figure.](image)
C. Boundary Conditions

The ground plane was modeled as a no-slip surface, with a constant translational velocity matching the forward speed of the bicycle. For this work a speed of either 20mph or 30mph was simulated. The fork, caliper & brake pads, head tube, top tube and down tube were modeled as no-slip surfaces with zero relative velocity. For the previous RANS analyses\textsuperscript{1,2}, a rotational frame of reference was applied to the outer wheel, inner wheel, spokes and hub using a rotational velocity which was consistent with the ground plane speed being studied. For the transient computations in this work, a rotational frame of reference was applied to the outer wheel as was done in the RANS case. On the inner wheel sub-volume, which contained the inner wheel rim, hub and spokes, a rotational mesh motion was applied to turn the wheel at a rotational speed matching the outer wheel reference frame.

A uniform velocity profile was applied to the far field boundary of the computational domain. The velocity vector direction was chosen to match one of five different yaw angles spanning the range from 0 to 20 degrees (e.g. 0°, 5°, 10°, 15° or 20°). A constant eddy viscosity far field condition was specified to be 0.001 m²/s. A pressure outlet condition was applied to the downstream boundary of the model domain. By using the truncated ellipsoid to represent the far field boundary, the same mesh was re-used for all yaw angles examined.

D. Numerical Methodology

In this work, the Navier-Stokes equations were solved using AcuSolve\textsuperscript{TM}, a commercially available flow solver based on the Galerkin/Least-Squares (GLS) finite element method\textsuperscript{21,26,27}. AcuSolve\textsuperscript{TM} is a general purpose CFD flow solver that is used in a wide variety of applications and industries. The flow solver is architected for parallel execution on shared and distributed memory computer systems and provides fast and efficient transient and steady state solutions for standard unstructured element topologies. Additional details of the numerical method are summarized by Johnson et.al\textsuperscript{28}.

The GLS formulation provides second order accuracy for spatial discretization of all variables and utilizes tightly controlled numerical diffusion operators to obtain stability and maintain accuracy. In addition to satisfying conservation laws globally, the formulation implemented in AcuSolve\textsuperscript{TM} ensures local conservation for individual elements. Equal-order nodal interpolation is used for all working variables, including pressure and turbulence equations. The semi-discrete generalized-alpha method\textsuperscript{29} is used to integrate the equations in time for transient simulations. This approach has recently been verified as being second-order accurate in time\textsuperscript{30}. The resultant system of equations is solved as a fully coupled pressure/velocity matrix system using a preconditioned iterative linear solver. The iterative solver yields robustness and rapid convergence on large unstructured meshes even when high aspect ratio and badly distorted elements are present.

The following form of the Navier-Stokes equations were solved by AcuSolve\textsuperscript{TM} to simulate the flow around the bike wheel:

\[
\frac{\partial \rho}{\partial t} + \nabla \cdot \rho \mathbf{u} = 0 \tag{1}
\]

\[
\rho \frac{\partial \mathbf{u}}{\partial t} + \mathbf{u} \cdot \nabla \mathbf{u} + \nabla p = \nabla \cdot \tau + \rho \mathbf{b} \tag{2}
\]

Where: \(\rho\) = density, \(\mathbf{u}\) = velocity vector, \(p\) = pressure, \(\tau\) = viscous stress tensor, \(\mathbf{b}\) = momentum source vector.
Due to the low Mach Number (Ma ~ 0.04) involved in these simulations, the flow was assumed to be incompressible, and the density time derivative in Eq. (1) was set to zero. For the preliminary steady RANS simulations, the single equation Spalart-Allmaras (SA) turbulence model\textsuperscript{19} was used. The turbulence equation is solved segregated from the flow equations using the GLS formulation. A stable linearization of the source terms is constructed to provide a robust implementation of the model. The model equation is as follows:

\[
\frac{\partial \tilde{v}}{\partial t} + u \cdot \nabla \tilde{v} = c_{b1} \tilde{S} \tilde{v} - c_{w1} f_w \left[ \frac{\tilde{v}}{d} \right]^2 + \frac{1}{\sigma} \left\{ \nabla \cdot \left[ (\nu + \tilde{v}) \nabla \tilde{v} \right] + c_{b2} \left( \nabla \tilde{v} \right)^2 \right\} \tag{3}
\]

Where:

\[
\begin{align*}
\tilde{S} &= |S| + \frac{\tilde{v}}{\kappa^2 d^2} f_{r2} \\
f_{r2} &= 1 - \frac{\chi}{1 + c_{w1} f_{v1}} \\
g &= r + c_{w2} (r^3 - r) \\
f_{v1} &= \frac{\chi^3}{\chi^3 + c_{v1}} \\
\chi &= \frac{\tilde{v}}{\nu} \\
S &= \frac{1}{2} \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \\
f_w &= g \left[ \frac{1 + c_{w3}^6}{g + c_{v3}^6} \right]^{1/6} \\
r &= \frac{\tilde{v}}{\tilde{S} \kappa^2 d^2} \\
|S| &= (2S_x S_y)^{1/2}
\end{align*}
\]

Where: $\tilde{v}$ = Spalart-Allmaras auxiliary variable, $d$ = length scale, $c_{b1} = 0.1355$, $\sigma = 2/3$, $c_{b2} = 0.622$, $\kappa = 0.41$, $c_{w1} = c_{b1}/\kappa^2 + (1+c_{b2})/\sigma$, $c_{w2} = 0.3$, $c_{v1} = 7.1$.

The eddy viscosity is then defined by:

\[
\nu_t = \tilde{v} f_{v1}
\]

For the steady state solutions presented in this work, a first order time integration approach with infinite time step size was used to iterate the solution to convergence. Steady state convergence was typically reached within 20 to 32 time steps for most simulations.

For the transient simulations, the Delayed Detached Eddy Simulation (DDES) model was used\textsuperscript{31}. This model differs from the (SA) RANS model only in the definition of the length scale. For the DDES model, the distance to the wall, $d_i$, is replaced by $\tilde{d}$ in Eq. (3). This modified length scale is obtained using the following relations:

\[
r_d = \frac{\nu_t + \nu}{\sqrt{u_i u_i \kappa^2 d^2}} \\
f_d = 1 - \tanh([8r_d]^3) \\
\tilde{d} = d - f_d \max(0, d - C_{DES} \Delta)
\]

Where: $\Delta$ = local element length scale, and $C_{DES}$ = the des constant.

This modification of the length scale causes the model to behave as RANS within boundary layers, and similar to the Smagorinsky LES subgrid model in separated flow regions. Note that the above definition of the length scale deviates from the original formulation of DES, and makes the RANS/LES transition criteria less sensitive to the mesh design.
E. Scope of Work and Force Resolution

The scope of this work encompassed the study of nine different bicycle wheel/fork configurations. Each configuration was run for five different yaw angles (e.g. 0°, 5°, 10°, 15° or 20°) chosen to match those that cyclists would normally encounter. Each yaw angle was run for two speeds: 20mph (competitive amateur cyclist/triathlete) and 30mph (professional cyclist/triathlete). This resulted in 90 individual design points run for two full wheel revolutions with time steps saved at every 2° of wheel rotation. Post-processing was run for all 90 design points on the last 256 time steps (covering 512 degrees of wheel rotation) only since it was noted that the numerical solution required some initial time steps in order to become stabilized. In total, 23,040 time steps were individually analyzed. For each time step, pressure and viscous forces were resolved to match: A) the axial drag force that a cyclist would experience in opposition to the direction of motion; B) the side (or lift) force, acting in a direction perpendicular to the direction of motion, and finally; C) the vertical force acting either towards or away from the ground plane, again acting perpendicular to the direction of motion. An illustration of the resolved forces with respect to the orientation of the bicycle wheel is shown in Figure 5. Also illustrated are the effective bicycle velocity and the effective wind velocity vectors which would represent the experimental conditions consistent with wind tunnel testing. The total (pressure+viscous) drag, side and vertical forces for each time step were further resolved on a component-by-component basis for the wheel, spokes, hub, fork and caliper & brake pads. In addition, the turning moment was computed for each of the wheels based on the side force acting on the wheel, spokes and hub.

It is of greatest interest to know the power that a rider needs to push a particular bicycle forward at a specific speed. This total power requirement is generally recognized as the sum of the following contributions:

1. Overall aerodynamic resistance
2. Wheel rotation
3. Rolling resistance
4. Friction losses
5. Gravity on inclines

![Figure 5: Force Resolution on the rotating bicycle wheel](image-url)
The power needed to overcome the drag, $F_D$, and the aerodynamic torque, $M$, acting on the front wheel is:

$$ P = F_D V + \sum_{\text{both sides}} M \omega $$

(4)

The torque acting on one side of the wheel is given by:

$$ M = \int_0^R r \cdot T_{\theta x} dA $$

(5)

A simple validation case to compute the torque acting on a spinning disk in a fluid at rest was carried out. A flat disc with a diameter and radius matching the nominal dimensions of the wheels being studied here was placed in a large surrounding volume. The base surface mesh, the number and thickness of the prism layers at the disc surface were all chosen to match the meshing characteristics for the wheel models. Disc rotational speeds matching the wheel speeds of 20mph and 30mph were examined. Comparisons between the torque computed from the CFD simulations and the experimental data, summarized from several sources by Schlichting et.al.32, were carried out.

Because of the repetitive and quantitative nature of the calculations required, all postprocessing of the simulation results needed to resolve forces, moments, torques and power were automated through the use of the FVXTM programming language feature available with FieldView33. Resolved forces, computed from FieldView were verified directly with AcuSolve™ output, on a subset of the total data produced for this study.

F. Co-processing and Concurrent Postprocessing Methodology

To postprocess all of the transient results generated for this study, a new co-processing methodology, and, a new concurrent postprocessing methodology were developed. With regard to the co-processing methodology, it was recognized that saving the full solver output at each of the 256 time steps for all of the 90 design points would add a significant amount of extra time and disk space to ultimately carry out the desired postprocessing calculations. To address this problem, co-processing between AcuSolve™ and FieldView was implemented to:

1. Reduce the disk space requirements needed to store the solver data during run-time
2. Keep the computational overhead low so as not to increase the solver run time, and
3. Remove the need for a (later) file conversion step requiring additional time and disk space

Since the primary objective of this study was to examine and compare the resolved forces, moments and power requirements for each of the design points, only the data near and at the bicycle component surfaces (tire, rim, hub, spokes, fork and so on) needed to be saved. Discarding the majority of the volume data, while keeping the solver data at the surfaces of interest, was done as follows. First, a python script was developed to create a ‘subset mesh’ which contained just the surface elements and a user-definable number of volume element layers starting from the relevant surfaces of interest. Provision for the unique specification of the surfaces of interest for the each of the nine bicycle wheel configurations (see Figure 1) permitted flexible creation of the subset mesh. In this work, a single volume mesh layer was used for each subset mesh.
To generate the intermediate co-processing files during the solver run, AcuSolve was set up to listen on a user-specified port for a specific client connection call. The client python script, when called, gathered the information from the subset mesh from memory, and wrote the results of the simulation for the current timestep onto disk using the standard FieldView unstructured file format. This basic co-processing methodology is illustrated in Figure 6.

To perform all of the postprocessing calculations of interest, three FieldView FVX programs were developed to automatically and sequentially run over all 256 timesteps, for the 10 operating conditions of speed and yaw angle, for each of the 9 designs (see Figure 1), returning data in spreadsheet ready form. The first of the three programs returned all of the resolved forces (drag, side and vertical), broken into pressure and viscous contributions for each of the major components (tire and rim, hub, spokes, fork and caliper & brake pads).

The second program computed the turning moment by integrating the product of the side force and the axial moment arm over a series of thresholded slices spanning the leading and trailing edge of the wheel. Side forces used in the moment calculations included the contributions from the tire, rim, hub and spokes.

The third FieldView FVX program returned the aerodynamic torque, as specified in equation (5). Calculations were repeated for both sides of the wheel by applying a threshold at the centerline in the direction normal to the wheel. The pressure side, suction side and total torque were tabulated in the spreadsheet output. Shear forces used in the torque calculations included contributions from the tire, rim, hub and spokes.

All postprocessing calculations were carried out remotely, in batch, on the co-processing output using a concurrent approach. Shell scripts were used to queue the jobs to run any one of the three FieldView FVX programs for all 90 design points at the same time. The queuing system (SLURM, Lawrence Livermore National Laboratory) on the available cluster computing resource ultimately controlled how many postprocessing jobs were being run simultaneously. However it was generally observed that more than 40 jobs were typically being run concurrently at any given time.
III. Current Results

A. Drag Coefficients for DDES Calculations and Comparison to Wind Tunnel Data

Drag coefficients, $C_D$, calculated from the DDES transient simulations are compared with previously published experimental data (see Greenwell et al.\textsuperscript{6}, Kyle et al.\textsuperscript{5,11-13}, Tew et al.\textsuperscript{10}, Prasuhn\textsuperscript{22} and Kuhnen\textsuperscript{23}) and unpublished data recently released for this work from Zipp Speed Weaponry, Indiana, USA in Figure 7. In each sub-figure, solid lines represent the experimental data for $CD$ as a function of yaw angle. The vertical bar, shown only at $0^\circ$ yaw, shows the range of experimental data reported from different sources for the wheel in question. The DDES data is shown by the whisker plots (20mph in blue, 30mph in red). Each whisker plot summarizes all 256 values of $CD$ for each operating condition (yaw angle and speed) for each wheel configuration. The caps at the end of each box indicate the extreme values (minimum and maximum), the box extents are defined by the lower and upper quartiles, and the line in the center of the box is the median $C_D$.

For the Zipp404, at $0^\circ$ yaw, the average drag coefficient, $C_D$, is calculated to be $0.0291$ for 20mph, and is slightly lower at $0.0272$ for 30mph. These results tend to agree reasonably well with previously published ‘wheel-only’ experimental data (see Greenwell et al.\textsuperscript{6}, Kyle et al.\textsuperscript{4,11-13}, Tew et al.\textsuperscript{10}, Prasuhn\textsuperscript{22} and Kuhnen\textsuperscript{23}). The drag coefficient trend with respect to yaw angle was also observed to be in good agreement with the published experimental data. The drag coefficient was nearly the same on the Zipp 404 with or without either fork present, and no significant differences between forks were observed. At higher yaw angles, the range between the minimum and maximum values was seen to increase. The presence or absence of the front fork appears to have no damping effect on this range.

On the deeper rimmed Zipp 1080 wheel, only limited drag coefficient data was available for comparison. At $0^\circ$ yaw, the average drag coefficient, $CD$, is calculated to be $0.0280$ for 20mph, and is slightly lower at $0.0256$ for 30mph. The drag coefficient was observed to be slightly lower at 30mph over the range of yaw angles studied. At 30mph only, drag forces exhibited a slight decrease at a $10^\circ$ yaw angle. In our previous study\textsuperscript{2}, RANS calculations showed that the Zipp 1080 exhibited a significant drag coefficient reduction at the $10^\circ$ yaw angle operating condition. Further, the overall trend for the drag coefficient with respect to yaw angle was seen to be in better agreement with the wind tunnel data for the RANS results compared to the DDES results. Similar to the Zipp 404, there was no significant difference whether either of the forks was present or not. The DDES results did show a dramatically increasing range of oscillation between the minimum and maximum calculated drag coefficient at both speeds for the higher yaw angles. The presence of either fork did not provide any damping effect on this oscillating behavior.

For the HED H3 TriSpoke, at $0^\circ$ yaw, the average drag coefficient, $C_D$, is calculated to be $0.0288$ for 20mph, and is slightly lower at $0.0271$ for 30mph. Data for the HED H3 TriSpoke is seen to agree well with wind tunnel data for several competitive trispoke wheels from other manufacturers at zero degrees yaw. The agreement with the drag coefficient versus yaw, reported by Zipp, and to a lesser degree Greenwell et al.\textsuperscript{6}, is well within the range of the experimental data. However, in contrast to the experimental results, the drag coefficient is seen to increase gradually with increasing yaw, and does not exhibit the characteristic maximum value at around 5 to 10 degrees.
In contrast with the other two wheels, the HED H3 TriSpoke exhibits a wider range between the minimum and maximum drag coefficients at lower yaw angles. This range stays relatively constant over the full range of yaw angles and does not increase at higher yaw angles in the same way that the Zipp 1080 is seen to do. Finally, there was no observable difference on the drag coefficient whether either fork was present.

**B. Resolved Drag Forces for DDES Calculations on all Wheels**

Time history plots, and the corresponding whisker plots, for the resolved drag force, on the tire, rim, hub and spokes, for all 9 design configurations (see Figure 1), for all yaw angles, at a speed of 20mph are illustrated in Figures 8a, 8b and 8c. In each of these three figures, the wheel only configuration is shown at far left; the Reynolds fork is shown in the middle, and the Blackwell fork is shown at far right. The overall trends and dependence on time with respect to yaw angle at 30mph are qualitatively similar to those observed at 20mph, and
consequently these results are not shown here. In Figure 8a, we observed a fairly uniform oscillation of drag on the Zipp 404 around a mean value at the higher yaw angles of 15° and 20°. As was noted with the drag coefficient, the absence or presence of the fork appeared to have no significant effect on the drag values. In Figure 8b, we see that the oscillations are less uniform about a mean value, and at the higher yaw angles, the amplitude of oscillation increases substantially. For the HED H3 TriSpoke wheel, we note that over 256 time steps, a spoke will make a passage through the fork gap four times. In Figure 8c, when either the Reynolds or Blackwell fork is present, we see that for
Low yaw angles, there are four maxima exhibited over the range of solution time steps shown. As the yaw angle increases, the behavior of the drag force with respect to time becomes more complex. At the highest yaw angle of 20°, there appears to be a dual peak on either side of the time at which the spokes pass through the fork gap. Notably, the double peak occurs even in the absence of the fork, for the wheel only case shown at left.

![Figure 8c: HED Trispoke Drag Force vs. Yaw Angle at 20mph](image)

### C. Resolved Drag Forces for DDES Calculations on all Forks

Time history plots, and the corresponding whisker plots, for the resolved drag force for both of the forks, configured for each wheel, for all yaw angles, at a speed of 20mph are illustrated in Figures 9a, 9b and 9c. No experimental data for drag forces on isolated forks was available for comparison. In each of these three figures, the Reynolds fork is shown at left, and the Blackwell fork is shown at right. The overall trends and dependence on time with respect to yaw angle at 30mph are very similar to those observed at 20mph, and consequently these results are not shown here. For all three wheels, we observed that there was a significant drag force difference between each fork, with the Blackwell fork consistently exhibiting nearly double the drag force computed for the Reynolds fork. With the Zipp 404 wheel, illustrated in Figure 9a, there is a difference with respect to the dependence of the fork drag force on yaw angle. For the Reynolds fork, the drag is seen reach a minimum at 5° yaw, and increase as the yaw angle increases. With the Blackwell fork, the drag force is highest at 0° yaw, and decreases as yaw increases.

Fork drag forces with the Zipp 1080 wheel configuration are illustrated in Figure 9b. With the Reynolds fork, we see almost no change in the drag force as the yaw angle increases. We do however see a general increase in the drag force amplitude as the yaw increases. With the Blackwell fork, a decrease in the drag with increasing yaw, similar to what was noted with the Zipp 404 wheel, was seen. And, again, the amplitude of the drag force was seen to increase with increasing yaw.
A Practical Analysis of Unsteady Flow Around a Bicycle Wheel, Fork and Partial Frame Using CFD

Figure 9a: Fork Drag Force vs. Yaw Angle (Zipp 404) at 20mph

Figure 9b: Fork Drag Force vs. Yaw Angle (Zipp 1080) at 20mph

Figure 9c: Fork Drag Force vs. Yaw Angle (Trispoke) at 20mph
The fork drag forces with the HED H3 TriSpoke wheel configuration is shown in Figure 9c. We observed a very strong dependence on the fork drag corresponding to the passage of the spokes. For the case of the Reynolds fork we noted a drag maximum/minimum pair at low yaw angles occurring just before and after the spoke passage. At higher yaw angles an interesting transition to a dual peak maximum (occurring just before and just after the spoke passage) was noted. The average drag force was seen to be relatively constant over all yaw angles. For the case of the Blackwell fork, a very dominant maximum drag was observed to occur at the moment of spoke passage between the forks. This maximum peak was seen to be highest at the 0° yaw case, and was seen to decrease slightly with increasing yaw.

The primary difference between the Blackwell fork and the Reynolds fork is the presence of several slots along each side of the fork. Streamlines, in Figure 10, illustrate how the slots in the fork are used to deflect air flow away from the wheel. Although this is a potentially desirable effect it accomplishes this at the expense of increasing the drag on the fork due to the action of redirecting the flow.

D. Resolved Turning Moments for DDES Calculations on all Wheels

Time history plots, and the corresponding whisker plots, for the turning moments computed from the integrated side forces, for all 9 design configurations, for all yaw angles, at a speed of 20mph are illustrated in Figures 11a, 11b and 11c. No experimental data for turning moments were available for comparison.

Figure 10: Streamlines depicting flow for the Zipp 1080 and Blackwell Fork

Figure 11a: Zipp 404 Turning Moment vs. Yaw Angle at 20mph
In each of these three figures, the wheel only configuration is shown at far left; the Reynolds fork is shown in the middle, and the Blackwell fork is shown at far right. The overall trends and dependence on time with respect to yaw angle at 30mph are very similar to those observed at 20mph, and consequently these results are not shown here.

For the Zipp 404, a maximum turning moment was observed at 10° yaw. This matched our previous RANS analysis for this wheel. A slight damping effect on the maximum turning moment was observed for both the Reynolds and Blackwell forks. At the highest yaw angle, 20°, the damping effect being provided by the forks was not significant. Additionally, the amplitude of the turning moment was seen to increase significantly at the higher yaw angles. However, the average turning moment was seen to decrease at these higher yaw angles.

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![Figure 11b: Zipp 1080 Turning Moment vs. Yaw Angle at 20mph](image)

![Figure 11c: HED H3 TriSpoke Turning Moment vs. Yaw Angle at 20mph](image)
Turning moments for the Zipp 1080 were seen to increase in terms of magnitude with increasing yaw angle. Turning moments for the Zipp 1080 were noted to act in a direction opposite to those observed for the Zipp 404.

The direction of the turning moment matched our previous RANS analysis\(^2\) for this wheel, however, the local maximum turning moment at 14° yaw was not seen with the DDES results. The amplitude of the turning moment fluctuations were seen to increase with increasing yaw angles.

Turning moments for the HED H3 TriSpoke were seen to reach a positive maximum at 5° to 10° yaw, and were then seen to become negative as the yaw angle increased. This trend matched our previous RANS analysis\(^2\) for this wheel very well. Time history plots showed that the fluctuations in turning moment matched the passage of the spokes between the fork gap. It is notable that at 15° yaw, the turning moments were evenly oscillating about a mean value near to zero.

**E. Resolved Torque - Validation Case**

Torque contours, obtained for a RANS simulation for the simple validation case (spinning disc in a (practically) infinite medium) are illustrated in Figure 12, for the nominal rotational speeds corresponding to 20mph and 30mph.

![Figure 12: Torque Contours for the Spinning Disc Validation Case](image)

The torque, calculated for the front and back of the spinning disc for the nominal speed of 20mph (\(\omega = 26.45\)) was 0.01318 N·m, essentially the same for both sides. This compared very with the range of 0.01264 to 0.01323 N·m, estimated from the experimental results\(^3\). At the higher nominal speed of 30mph (\(\omega = 39.68\)), the calculated torque was 0.02522 N·m (again, essentially the same for both sides) which also compared with the experimental range of 0.02552 to 0.02689 N·m.

**F. Resolved Aerodynamic Torque for DDES Calculations on all Wheels**

Time history plots, and the corresponding whisker plots, for the turning moments computed from the integrated side forces, for all 9 design configurations, for all yaw angles, at a speed of 20mph are illustrated in Figures 13a, 13b and 13c. No experimental data for aerodynamic torque were available for comparison. In each of these three figures, the wheel only configuration is shown at far left; the Reynolds fork is shown in the middle, and the Blackwell fork is shown at far right. The overall trends and dependence on time with respect to yaw angle at 30mph are qualitatively similar to those observed at 20mph, and consequently these results are not shown here.
Torque variations with respect to time tended to be small for all of the design points studied. For the Zipp 404, the torque was seen to reach a maximum at 5° yaw, and decrease with increasing yaw after that point. For yaw angles in the range of 10° to 15°, the presence of the Reynolds fork led to a reduction in torque, and a further reduction was seen with the Blackwell fork.

For the Zipp 1080, the torque dependence with respect to yaw was seen to be very similar to that observed for the Zipp 404, with a maximum value occurring at 5° yaw, and decreasing thereafter with increasing yaw angle. For this deeper rim, the present of either fork did not have any significant effect on torque.

Of all three wheels studied, the torque on the HED H3 TriSpoke was seen to be the lowest. Fluctuations in the torque were also seen to be quite small, and were nearly constant over the range of yaw angles studied.
G. Resolved Power Requirements for all Wheel/Fork Combinations

Time history plots, and the corresponding whisker plots, for the resolved power for both of the forks, configured for each wheel, for all yaw angles, at speeds of both 20mph and 30mph are illustrated in Figures 14a, 14b and 14c. The total power resolved for each design configuration was calculated by summing the power contribution for the wheel (including the tire, rim, hub and spokes) using equation (4) and the power needed to additionally overcome the fork drag. For all configurations, the power requirement at 30mph is substantially higher than that for 20mph. In general, the power needed to increase speed from 20mph to 30mph went up by a factor of 3X.
For the Zipp 404, illustrated in Figure 14a, it can be seen that the overall power requirement is highest with the Blackwell fork. For the case of the Reynolds fork, the power at both speeds is seen to increase with increasing yaw angle. The amplitude of oscillation about the mean power is seen to be fairly evenly distributed, and increases with increasing yaw angle. For the case of the Blackwell fork, a slight minimum power requirement is observed from 5° to 10° yaw.

The Zipp 1080 power requirements are shown in Figure 14b. The power requirements for the Reynolds fork are still lower than those needed for the Blackwell fork, however the difference between the two configurations is not as great as was observed for the Zipp 404 wheel.

Figure 14b: Power vs. Yaw Angle (Zipp1080) at 20mph (left) and at 30mph (right)

Figure 14c: Power vs. Yaw Angle (TriSpoke) at 20mph (left) and at 30mph (right)
The HED H3 TriSpoke power requirements are shown in Figure 14c. Again, the Blackwell fork requires greater power than the Reynolds fork. Peak power requirements occur when a spoke passes through the fork slot for both forks. In the case of the Blackwell fork, this peak value is far above the mean value, and reaches a maximum at a 15° yaw angle.

H. Frequency Resolution for all Wheels

Power spectral densities were computed using Fourier transforms for the resolved wheel drag force (see Figures 8a, 8b and 8c), the resolved fork drag force (see Figures 9a, 9b and 9c), the computed turning moment (see Figures 11a, 11b and 11c) and the fully resolved power (see Figures 14a, 14b and 14c). In most but not all cases, dominant peaks were observed for the different operating conditions of speed and yaw angle. The dominant Strouhal number, \( St \), where \( St = fD/V \) peak for each configuration, for each of the resolved properties noted are illustrated in Figures 15a thru 15d.
In Figure 15a, we observed a peak Strouhal number at 10° yaw, which then decreased with increasing yaw angle for the Zipp 404 wheel. There was little difference noted between any of the configurations. Strouhal numbers were also roughly the same for both speeds. A single peak Strouhal number was observed for the Zipp 1080 at 5° yaw. Similar to the Zipp 404, the Strouhal number was generally seen to decrease with increasing yaw angle. There was no consistent difference between the different configurations and speeds studied. In comparing the Strouhal numbers for the Zipp 404 and the Zipp 1080 with respect to yaw angle, the latter was seen to consistently exhibit the lower frequency. This observation suggests that flow is remaining attached to the Zipp 1080 wheel longer relative to the Zipp 404. For the HED H3 TriSpoke, the Strouhal number showed no dependence on yaw angle, configuration or speed. The frequency observed matched that of the passage of the spokes. Neither of the other wheels showed dominant peaks relating to the spoke passage; the periodic fluctuations were due entirely to the shedding of fluid from the wheels during their rotation.

Figure 15c: Strouhal Number (based on Turning Moment) vs. Yaw Angle

Figure 15d: Strouhal Number (based on Power) vs. Yaw Angle
In Figure 15b, we report the Strouhal number for drag on each of the forks for the various wheels. Notably, the results for the forks tend to match those observed for the drag coming from the wheels. This suggests that the periodic fluctuation of drag on the forks is dominated by the shedding of fluid coming from the wheels. With either the Zipp 404 or Zipp 1080 present, we note the same lack of dependence of Strouhal number on configuration or speed. With the HED H3 TriSpoke, we plotted several strong secondary peaks in addition to the dominant ones. These secondary peaks occur at harmonic frequencies, suggesting that there may be strong interactions between the fork and the characteristic wide spokes for the HED H3 TriSpoke wheel related to their passage between the fork gap.

Strouhal number calculations based on the turning moment (see Figure 15c) and the power (see Figure 15d) match those observed for the wheel drag force with respect to yaw angle, configuration and speed. This reinforces the concept that shedding from the wheels is a strongly periodic phenomenon which is dependent on the yaw angle and appears to be a characteristic of the wheel itself. In our previous work\(^2\), this same observation was noted for several ‘wheel-only’ studies carried out at a single yaw angle of 10\(^\circ\).

**IV. Discussion**

In this work CFD was used to explore the complex and unsteady nature of air flow around several commercially available front bicycle wheel configurations. Extending our previous work\(^1,2\), this study examined more realistic front wheel geometry, adding the front fork, top tube, head tube, down tube, caliper and brake pads to the modeling domain: Three wheels, namely the Zipp 404, Zipp1080 and HED H3 TriSpoke were considered. In addition, two commercially available front fork designs, the Reynolds Carbon fork and the Blackwell Bandit slotted fork, were also studied. This led us to consider a total of nine different design configurations. An approach, previously presented\(^1\), was used to model each wheel with accurate spoke rotation, using a realistic ground contact. Unsteady calculations were run with the commercial solver, AcuSolve\(^TM\) using the DDES turbulence model. Simulations were run for each of the nine design configurations over a combination of 2 speeds and 5 yaw angles, producing a total of 90 unique design points. This again extends our previous work\(^2\) where transient runs were carried out on only 6 unique design points (6 different ‘wheel-only’ configurations examined at one speed (20mph) and one yaw angle (10\(^\circ\))). For all 90 design points, 256 time steps were analyzed, resulting in the need to review 23,040 individual time steps. To make the analysis and management of this amount of data possible and practical, two new methodologies were introduced. First, a co-processing approach was developed between AcuSolve\(^TM\) and FieldView to generate substantially reduced, yet sufficient and accurate representations of the solver data for every time step. Second, several FieldView FVX\(^TM\) scripts were developed and subsequently run in a remote, concurrent, batch mode of operation to automate and significantly reduce the time needed to carry out the postprocessing presented in this study. These FieldView FVX\(^TM\) postprocessing scripts calculated i) the resolved forces (drag, side and vertical), broken into pressure and viscous contributions for each of the major components (tire and rim, hub, spokes, fork and caliper & brake pads), ii) the turning moments, iii) the aerodynamic torques acting on the rotating wheels, iv) the power requirements for each design configuration studied, and v) the power spectral densities for selected variables of interest. The workflow used here was further streamlined and automated by customizing FieldView FVX\(^TM\) scripts to generate spreadsheet ready data output for seamless transfer into other software packages.

The co-processing approach between AcuSolve\(^TM\) and FieldView delivered three significant benefits. First, disk space requirements needed to store the solver data were substantially reduced by writing out the solver results for each time step in a reduced form. The typical size of a time step result (for any of the 9 design configurations) was on the order of approximately 50MB. In our previous studies\(^1,2\), we needed to store the full volume data for each time step, and additionally, export the data to the FieldView format to complete the postprocessing. For each time step, disk space requirements were approximately 1.7GB. Co-processing reduced disk space requirements by more than a
factor of 30. Second, the time needed to postprocess each time step was significantly reduced. Elapsed times needed to calculate resolved forces and aerodynamic torques were recorded and compared when working with the full volume export versus the co-processing data. On average, a 15-fold to 20-fold reduction in elapsed postprocessing time was realized when working with the co-processing data. Numerical results were identical when using either the full volume data or the co-processing data. Third, the memory requirements needed for loading and postprocessing were reduced by roughly a factor of 2 when working with the co-processing data. The computational overhead introduced by having to write the co-processing files was observed to be roughly 3% of the total elapsed solver time. Normally, the native solver volume data would be written at each time step. The time spent by writing the co-processing files instead was observed to be offset by not having to write the native solver volume data. We believe that benefits of the co-processing methodology introduced here will be of considerable value, particularly for transient analyses.

The CFD predictions for the drag coefficient and trend behavior with respect to yaw angle were seen to be in relatively good agreement with the experimental drag force data, kindly provided by Zipp Speed Weaponry (Indiana, USA), along with several other published sources\textsuperscript{5,10,13,22,23} for the Zipp 404 and the HED H3 TriSpoke wheels. For these wheels, the predicted drag coefficients tended to be lower than those recorded from the wind tunnel. We speculate that the primary source of these differences arises from the use of different boundary conditions and domain extents used in our simulations relative to the actual wind tunnel conditions. Agreement between the predicted drag coefficients and the wind tunnel data for the Zipp 1080 was poor. Notably, the minimum drag at 15° yaw, which was observed experimentally, and correctly captured in our previous work using RANS modeling\textsuperscript{2}, was not seen. Very large oscillations in the drag force on the wheel at higher yaw angles lead us to speculate that the predicted flow may be prematurely detaching from, and quickly re-attaching to, the surface of this wheel in a physically unrealistic way. Although the surface area of the rim and tire for this wheel (0.487m\textsuperscript{2}) is higher than the Zipp 404 (0.319m\textsuperscript{2}) or the HED H3 TriSpoke (0.396m\textsuperscript{2}), the surface mesh density and y+ values were comparable between all three wheels studied. Grid refinement, particularly on the rim and the tire, and a review of the DDES turbulence modeling methodology should be considered.

It was expected that the presence of the front fork near the rotating bicycle wheel would have a significant impact on the resolved forces and turning moments for the wheels themselves, and preliminary unpublished RANS analyses on these design configurations reinforced this expectation. Surprisingly, the presence of either fork was seen to have no significant effect on the wheel drag for any of the wheels studied. A slight damping effect on the turning moment was seen on the Zipp 404 for intermediate yaw angles only; otherwise, no significant effects were observed. However, significant differences between forks were seen. The Blackwell fork exhibited much higher drag forces when compared to the Reynolds fork. Drag forces on the Reynolds fork were seen to increase slightly with increasing yaw angle when used with the Zipp 404, and remained relatively constant for all yaw angles when used with either the HED H3 TriSpoke or Zipp 1080 wheel. In contrast, the drag forces on the Blackwell fork were seen to generally decrease with increasing yaw angle. With the Blackwell fork/HED H3 TriSpoke combination, a very strong peak drag force, well in excess of the average fork drag, was observed as the wide spoke made the passage through the fork gap.

Turning moment trends with respect to yaw angle were in very good agreement with previous RANS results\textsuperscript{2} for each of the wheels studied here. The Zipp 404 exhibited a maximum turning moment at a 10° yaw angle for both speeds studied. Turning moments for the Zipp 1080 acted in the opposite direction relative to the Zipp 404. Previously\textsuperscript{2}, it was observed that as the rim depth increased, the balance of side forces shifted from the trailing half of the wheel to the leading half. This observation holds true for the DDES results as well. The HED H3 TriSpoke also demonstrated a maximum turning moment at a yaw angle of 10°, similar to that observed for the Zipp 404. Of greater relevance
to stability however is the amplitude of the turning moment. The Zipp 404 was observed to have a very low turning moment amplitude for all yaw angles except for 20°, suggesting that it should be a relatively stable wheel to ride. The Zipp 1080 however had a very large turning moment range for yaw angles above 10° suggesting that it might have control and stability issues in cross winds. Although the HED H3 TriSpoke was observed to exhibit a smaller turning moment range compared to the Zipp 1080, this range was relatively high and constant for yaw angles above 5°. While the average turning moment for the HED H3 TriSpoke was near zero at a 15° yaw angle, this may be misleading in terms of stability since the oscillations from the average value were significant.

The contributions to overall power requirements coming from the aerodynamic torque are calculated, and are believed to be presented for rotating wheels in contact with the ground for the first time in this work. We observed significant differences between wheels and significant variations with respect to yaw angle for both the Zipp 404 and Zipp 1080. As expected, the highest aerodynamic torque was predicted for the Zipp 1080 since it is the wheel with the greatest surface area. Although the HED H3 TriSpoke has a higher surface area relative to the Zipp 404, it exhibited lower aerodynamic torque values. Anecdotally, the contribution of aerodynamic torque to overall power is believed by the cycling community at large to be very small, on the order of a few percent at most. In this work, the contribution of aerodynamic torque to the total power requirement ranged from 9% to 10% for the HED H3 TriSpoke, 12% to 18% for the Zipp 404, and 17% to 24% for the Zipp 1080. Differences in the aerodynamic torque for each wheel were expected. Unpublished data from Zipp indicates that the ‘wattage-to-spin’ requirement, or the power needed to turn a test wheel during wind tunnel experiments, varies significantly between wheels.

To obtain a sense of the relative performance merits for the design configurations studied here, the overall power requirements were calculated for each combination of front wheel and fork; wheel only configurations were not included here. In general, it was observed for all cases that the power requirement needed to increase the speed from 20mph to 30mph went up by a factor slightly greater than three. This result is consistent with the convention that the increase in power goes up with the cube of the increase in speed. The highest power requirement was observed for the combination of the Zipp 1080 and the Blackwell fork, however, we re-iterate at this point that the drag predictions for the Zipp 1080 were not in good agreement with wind tunnel results. Notably the Blackwell fork had a higher power requirement than the Reynolds fork when used in combination with any of the wheels studied. The HED H3 TriSpoke exhibited the lowest power requirement relative to the other two wheels. But, the peak power requirement, which occurs during the transit of the spoke through the fork gap, is well in excess of the average value particularly for the case of the Blackwell fork configuration. Under practical riding conditions, if the rider is unable to generate the power needed to ‘push thru the peaks’, a lower average speed will be achieved. We suggest that power fluctuations, and not average power requirements, associated with the combination of a specific front wheel and fork should be of primary relevance to riders.

The wheel drag, fork drag, turning moment and power all exhibited strong periodic behavior with significant differences between wheels (but not configurations) and yaw angles. Of the three wheels studied here, only the HED H3 TriSpoke demonstrated a correlation, for all variables examined, between the Strouhal Number (=0.895) and the rotational motion of the three spokes. This observation was consistent for all yaw angles examined. A comparison of the Strouhal numbers based on the fork drag and either of the wheel drag, turning moment or power, suggest that the periodic fluctuations are being driven by the wheel for the case of the Zipp 404 and the Zipp 1080.

Previously we proposed that the dominant periodic behavior comes from the periodic shedding of fluid off of the rim and tire. Comparing the Zipp 1080 with the Zipp 404, we expect the flow to stay attached to the Zipp 1080 longer since it has a deeper rim, and this is consistent with
the Strouhal number predictions (for the same yaw angle). Starting at a 0° yaw angle, we observed the Strouhal number to increase for both the Zipp 404 and Zipp1080 (when periodic behavior was present) with increasing yaw angle. This suggests that the flow was initially attached and started to become detached as the yaw angle increased. Interestingly, for nearly all cases, a maximum Strouhal number was observed at 10° yaw. Beyond this 10° yaw angle, the Strouhal number was seen to decrease. In our previous work, we identified a vortex structure on the suction side of the wheel for the case of the Zipp 404. This vortex structure started forming in the upper quadrant of the wheel, and eventually extended along the leading edge of the wheel to a point just ahead of the ground contact. We speculate that as this structure grows with increasing yaw angles, the rate of shedding slows down, as evidenced by the observation of the reduced Strouhal number. Further analysis of the data should be conducted to confirm this speculation.

The primary objective of this work was to more realistically model the performance of a commercial bicycle wheel, first by including more of the components around the front wheel (such as the fork, frame, calipers and brake pads), and second by running transient instead of steady CFD simulations. The new co-processing and concurrent postprocessing methodologies developed for this work make the analysis of many wheel/fork/component combinations with AcuSolve™ and FieldView both possible and practical. Several critical issues such as the wheel rim depth and cross-sectional profile, wheel diameter (650c or 700c), and development of an integrated front fork/frame/caliper assembly, optimized to work with a specific wheel, still remain open. We believe that CFD is now in a position to fully explore wind tunnel discoveries, and advance the understanding of the critical design changes which ultimately lead to the performance improvements sought after by competitive and amateur cyclists and triathletes.

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References

A Practical Analysis of Unsteady Flow Around a Bicycle Wheel, Fork and Partial Frame Using CFD


