A Comparative Aerodynamic Study of Commercial Bicycle Wheels Using CFD

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Abstract
A CFD methodology is used to study the performance of several commercial bicycle wheels over a range of speeds and yaw angles. The wheels studied in this work include the Rolf Sestriere, HED H3 TriSpoke, the Zipp 404, 808 and 1080 deep rim wheels and the Zipp Sub9 disc wheel. Wheels are modeled at speeds of 20mph and 30mph, in contact with the ground, using Reynolds-Averaged Navier Stokes (RANS). Drag, vertical and side (or lift) forces are reported for each wheel. Turning moments are also calculated using the resolved side forces to examine aspects of stability and maneuverability. Drag and side forces over the range of yaw angles studied compare favorably to experimental wind tunnel results. The previously reported unique transition from downward to upward acting vertical force on the Zipp 404 wheel for increasing yaw angles is observed for all deep rim wheels and the disc wheel studied here. Wheels were also modeled at a critical yaw angle of 10 degrees using Delayed Detached Eddy Simulation (DDES) to examine the transient aspects of flows around moving bicycle wheels. It is hoped that a more complete comprehension of these results will lead to improvements in performance, safety and control of bicycle racing wheels used by amateur and professional cyclists and triathletes.

Nomenclature
\[
\begin{align*}
C_d &= \text{drag coefficient} (= \frac{F_d}{0.5 \rho V^2 S}) \\
C_s &= \text{side force coefficient} (= \frac{F_s}{0.5 \rho V^2 S}) \\
C_v &= \text{vertical force coefficient} (= \frac{F_v}{0.5 \rho V^2 S}) \\
D &= \text{nominal bicycle wheel diameter} \\
F_d &= \text{axial drag force} \\
F_s &= \text{side (or lift force)} \\
F_v &= \text{vertical force} \\
f &= \text{frequency} \\
\rho &= \text{pressure} \\
S &= \text{reference area, } \pi D^2/4 \\
St &= \text{Strouhal No. } (= fD/V) \\
t^* &= \text{dimensionless time } (= tV/D) \\
u &= \text{velocity vector} \\
V &= \text{bicycle speed (in direction of travel)} \\
\beta &= \text{yaw angle} \\
\mu &= \text{viscosity} \\
\end{align*}
\]

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I. Introduction

Once again, margins of victory separating the top three professional riders in the 2008 installation of the Tour de France and the 2009 Giro D’Italia were less than 2 minutes after nearly 90 hours of racing. Small time differences between the top finishers, on the order of 2% in the individual and team time trial stages, were critical in determining final finish positions. High profile events such as these serve to focus on the relevance of seemingly small performance gains associated with technological advances in cycling equipment. As a result, Tour Pro cyclists, Professional Triathletes, and Age Group triathletes are becoming increasingly aware of the importance of aerodynamics and the role it plays in the selection of their equipment. Recently, it has been suggested that wind tunnels have become the greatest catalyst for technological progress in the cycling industry in the last two decades. And, Manufacturers such as Zipp make significant annual investments in wind tunnel testing. Their design methodology is based on three steps. First, computer modeling is used to reduce their typical design space from 100 prototypes down to 10. In the next step, they use the wind tunnel test results from their initial prototypes to guide their final designs. Wind tunnel validation of their final designs is the last step. They, along with several other cycling companies, continue to regard wind tunnel testing as the definitive standard in evaluating the success of their new designs. However, it should be noted that there is no standard for wind tunnel testing of bike wheels and components. In addition, the elevated platforms designed to move the bicycle frame and wheel above the boundary layer on the floor of the wind tunnel will vary, and in many cases, results are normalized to match tunnel wind speeds with measured axial forces. Consequently, direct comparisons of wind tunnel studies between the wheel manufacturers are extremely difficult to make.

An additional constraint for bicycle and wheel manufacturers is to maintain compliance with the standards set forth by the International Cycling Union, specifically in the area concerning non-standard wheels in conformity with article 1.3.018. Recently, the International Cycling Union made the decision to increase equipment inspections, imposing its guidelines for restricting fuselage form on tubes and crossbars to have ratios on width and diameters not exceeding three to one. Although the intent of this restriction is to level the technical advantages for teams competing in major events like the Tour de France, what this will do is decrease the development cycles for equipment manufacturers. It is likely that effectively applied CFD methods, offering the benefit of reducing wind tunnel testing, may be the most effective way for manufacturers to be responsive to changes.

It is commonly recognized that main contributors to overall drag are the rider, the frame including fork and aerobars, and the wheels. Comprehensive reviews by Burke and Lukes et.al. cite many efforts to identify the most relevant contributions to overall performance improvements in bicycle racing. 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 flow around a rolling wheel in contact with the ground. Wray et al. \textsuperscript{18} used RANS with standard k-\( \varepsilon \) and RNG k-\( \varepsilon \) 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. \textsuperscript{19} 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 \textsuperscript{20} (S-A) model and a two equation realizable k-\( \varepsilon \) turbulence model \textsuperscript{21} (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 were 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.

To date, little work has been done to compare CFD results for rolling bicycle wheels in contact with the ground against wind tunnel data. Godo et al. \textsuperscript{22} used a novel methodology to validate computed drag results against wind tunnel data \textsuperscript{7,11} for an existing commercially produced bicycle wheel (Zipp 404). Steady and unsteady computations were performed using the commercial solver, AcuSolve\textsuperscript{TM}, \textsuperscript{23} over a range of speeds and yaw angles. They 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 wheel, hub and spokes, something not possible with wind tunnel testing, they observed this unexpected transition to be limited to just the wheel. The hub was seen to act as expected in terms of the direction of vertical force, following the well known Kutta-Joukowski theorem.

In this work, we extend our efforts to model the flow around several commercial bicycle racing wheels. Our objectives will be three-fold. First we hope to be able to validate trends and drag measurements based on newly obtained wind tunnel data for these commercial wheels. Second, we will attempt to use computations such as the resolution of forces on components (tire, rim, hub and spokes), turning moments, and streamline and streakline visualizations, which can only be obtained from CFD analyses, to offer detailed insights into the complex flow structures around rotating bicycle wheels. And finally, by presenting the automation methodologies necessary for completing a CFD study of this scope, it is hoped that commercial wheel and bicycle manufacturers may be motivated to increase the use of CFD in their workflow and production efforts.

**II. Methodology**

**A. Wheel, Hub and Spoke Geometry**

A profile gauge was used to measure the cross sectional profile for the 700c production versions of the Zipp 404, 808 and 1080 deep rim wheels (Zipp Speed Weaponry, Indiana, USA), the Rolf Sestriere standard rim wheel (Rolf Prima Wheel systems), and the HED H3 TriSpoke wheel (HED Cycling Products). A notional model of the Zipp Sub9 (Zipp Speed Weaponry, Indiana, USA) was also constructed. Each wheel was modeled with a Continental Tubular tire attached and inflated to 120psi. A standard hub profile, based on the Zipp 404 wheel, was used for all wheels except the HED H3 TriSpoke and Zipp Sub9. Spoke cross sections for all wheels were based on the commercially available CX-Ray spoke (Sapim Race Spokes, Belgium). Spoke counts and spacing matched the production wheels. An illustration of the wheel cross sectional profiles obtained from the profile gauge, and the final CFD mesh geometries used for each wheel are illustrated in Figure 1. The frame 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. Computational Grid Development
Highly parameterized journals were developed for GAMBIT and TGrid to create the geometry and computational grids used in this work. This approach makes it possible in future studies to easily change other elements of the bicycle wheel geometry, such as the tire, rim profile or spoke count. For each wheel, the model domain was divided into two sub-volumes, with one containing the spokes (or disc in the case of the Zipp Sub9), hub and inner edge of the wheel rim, and the other containing the remaining toroidal rim surface, tire and the ground contact, and the surrounding volume. This division provides for application of a rotational frame of reference to the inner wheel section, permitting realistic movement of the spokes in transient modeling efforts. A non-conformal interface was used between the inner wheel sub-volume and the surrounding fluid volume. An illustration of this general methodology is shown in Figure 2.
A yaw angle was applied to correctly orient the complete wheel assembly, including the inner wheel volume, within the surrounding volume. Thus, the mesh created for each wheel was unique for each yaw angle. This approach permitted the use of uniform upstream boundary conditions. The dimensions of the surrounding domain volume were standardized using the outer wheel diameter, D. The wheel center was located 0.75D from the domain inlet. The domain width, length and height were set to be wheel diameter factors of 2D, 4D and 1.5D, respectively. Subsequent checks on converged RANS solutions were performed to ensure that the domain extents were sufficient to resolve pressures at the domain boundaries.

Figure 2: Methodology for Meshing the Wheel Geometry

Figure 3: Mesh Domain Around Wheel (left) and Prism Layers (right) on Zipp 1080 Rim and Tire
Three prismatic boundary layers of constant thickness (0.025cm per layer) were generated for the wheel rim and tire, the hub and the spokes for all wheels. In addition, a rectangular zone was created around each wheel in which the volume mesh was refined relative to the rest of the domain. An illustration of the mesh used to study the Zipp 1080 is shown in Figure 3. Grid convergence for each mesh was verified by first calculating the resolved forces on each of the different wheel designs for a converged RANS solution on a starting grid density at the condition of zero degrees of yaw and a speed of 20mph. Next, the mesh surface density and volume density were both increased, increasing the total mesh size by approximately 10%, and the resolved forces were compared to those obtained for the preceding mesh. The mesh density was considered to be sufficient when the resolved forces were within 5% of the preceding case. For most wheels, this process led to a mesh of sufficient density after 2 to 4 process iterations. Care was taken to ensure that the surface grid cell size, based on area, particularly on the rim and tire, was similar for each wheel studied. For this work, computational grids ranging in size from approximately 6 million tetrahedral and wedge elements (for the HED H3 TriSpoke) to 11 million tetrahedral and wedge elements (for the Zipp Sub9 disc) were used. Once the final mesh density was achieved, the meshes needed for the remaining yaw angles were automatically generated. A total of 60 meshes, one for each of the six wheels examined at each of the 10 different yaw angles were produced for this work.

C. Boundary Conditions

A uniform velocity profile was applied to the upstream boundary of the computational domain. This velocity was chosen to match the forward speed of the bicycle. For this work speeds of 20mph and 30mph were simulated. A constant eddy viscosity inlet condition was specified to be 0.001 m²/s. The ground plane was modeled as a no-slip surface, with a constant translational velocity matching the specified upstream speed. The ground plane velocity components were adjusted to be parallel to the bicycle wheel axis of travel at the yaw angle specified. Yaw angles spanning the range from 0 to 20 degrees were examined. For the RANS computations, 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 upstream velocity being studied. For the DDES computations, 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 speed matching the outer wheel reference frame. A pressure outlet condition was applied to the downstream boundary of the model domain. Slip conditions were applied to the remaining outer boundaries of the surrounding volume. An illustration of these boundary conditions and the computational domain extents are illustrated in Figure 4.
D. Numerical Methodology

In this work, the Navier-Stokes equations were solved using AcuSolve™, a commercially available flow solver based on the Galerkin/Least-Squares (GLS) finite element method. AcuSolve™ 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.

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™ 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 is used to integrate the equations in time for transient simulations. This approach has recently been verified as being second-order accurate in time. 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™ to simulate the flow around the bike wheel:

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

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

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 \( \sim 0.04 \)) 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 steady RANS simulations, the single equation Spalart-Allmaras (SA) turbulence model 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:
\begin{equation}
\frac{\partial \vec{v}}{\partial t} + \mathbf{u} \cdot \nabla \vec{v} = c_b \vec{S} + c_{w_1} \left[ \frac{\vec{v}^2}{d} \right] + \frac{1}{\sigma} \left[ \nabla \cdot \left( \vec{v} + \vec{v} \right) \nabla \vec{v} \right] + c_{w_2} \left( \nabla \vec{v} \right)^2 \right) \end{equation}

\begin{align*}
\vec{S} &= |\vec{S}| + \frac{\vec{v}}{c_b^2 d^2} f_{v_2} \\
g &= r + c_{w_2} (r^6 - r) \\
\chi &= \frac{\vec{v}}{\nu} \\
S_y &= \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_{w_3}}{g^6 + c_{w_3}} \right]^{1/6}
\end{align*}

Where: \( \vec{v} \) = Spalart-Allmaras auxiliary variable, \( d \) = length scale, \( c_{b_1} = 0.1355 \), \( \sigma = 2/3 \), \( c_{b_2} = 0.622 \), \( \kappa = 0.41 \), \( c_{w_1} = c_{b_1}/\kappa^2 + (1+c_{b_2})/\sigma \), \( c_{w_2} = 0.3 \), \( c_{w_3} = 2.0 \), \( c_{\nu_1} = 7.1 \).

The eddy viscosity is then defined by:

\[ \nu_t = \vec{v} f_{v_1} \]

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 50 time steps for most simulations. Yaw angles above 12 degrees require small amounts of relaxation to obtain numerical convergence.

For the transient simulations, the Delayed Detached Eddy Simulation (DDES) model was used. 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, \( d \), is replaced by \( \tilde{d} \) in Eq. (3). This modified length scale is obtained using the following relations:

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

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.

Transient simulations in this work were carried out for a 10° yaw angle for all wheels at a speed of 20mph. Calculations were run for two full rotations of the bicycle wheel, with simulation results saved at every 2° of wheel rotation. In subsequent frequency analyses of the resolved forces, only the last 256 time steps for each transient case were used since it was noted that the numerical solution required some time steps in order to be stabilized.

All postprocessing of the simulation results was automated through the use of the FVX™ programming language feature available with FieldView. Resolved forces, computed from FieldView were verified directly with AcuSolve output. Flow structures were identified using the vortex core detection algorithms implemented within FieldView.

**E. Scope of Work and Force Resolution**

The scope of this work encompassed the study of six different bicycle wheels at 10 different yaw angles (0°, 2°, 5°, 8°, 10°, 12°, 14°, 16°, 18° and 20°) at the two speeds of 20mph and 30mph, resulting in 120 individual design points run at steady state. Based on these results, transient studies were then run for one critical yaw angle of 10 at 20mph, generating solution results for 256 time steps for each wheel – this produced a total of 1536 solutions to be analyzed. Because of the repetitive and quantitative nature of the calculations required, postprocessing of the simulation results to resolve drag, side and vertical forces and moments, and generate reports, was automated through the use of the FVX™ programming language feature available with FieldView.

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.

Resolved side forces were used to calculate turning moments for all wheels. A further calculation was carried out on the tire, rim and hub (and including the spokes for the HED H3 TriSpoke, and disc for the Zipp Sub9) to examine the circumferential force variations. This was accomplished by computing the resolved force components on a circular segment spanning 1°, repeated to encompass the entire 360° circumference of the wheel.

III. Current Results
A. Drag Coefficients for RANS Calculations and Comparison to Wind Tunnel Data
Drag coefficients, CD, calculated from the CFD simulations are compared with previously published experimental data (see Greenwell et al., Kyle et al., Tew et al., Prasuhn and Kuhnen) and unpublished data recently released for this work from Zipp Speed Weaponry, Indiana, USA in Figures 6a thru 6c.

![Figure 6a: Comparisons Between Wind Tunnel Results and CFD for Zipp404 and 808](image)

![Figure 6b: Comparisons Between Wind Tunnel Results and CFD for Zipp1080 and Sub9](image)
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For the Zipp 404, \( C_D \) at 0 degrees yaw was well within the range of published drag coefficients (see floating bar at zero degrees in each figure above). In addition, a comparison to the wind tunnel data provided by Zipp shows very strong agreement in terms of the behavior of the drag force with respect to yaw angle. Both CFD and experimental results show a minimum drag value at 10 degrees yaw. The Zipp 808 data is very similar to the Zipp 404; the drag coefficient at zero degrees yaw again is within the range of published drag coefficients. And, again agreement with the trend of a minimum drag at 10 degrees yaw is well captured. Comparison against the drag data from Kuhnen\textsuperscript{36} shows reasonably good trend agreement, with their minimum drag occurring at a slightly higher yaw angle of 12 degrees. For the Zipp 1080, the Zipp wind tunnel results show a minimum drag at 15 degrees, while the CFD results show this minimum to occur earlier at 10 degrees. General wind tunnel trend behavior for this wheel however is in good agreement with the CFD predictions. The Zipp Sub9 drag coefficients lie well within the range of other published data covering several different disc wheel designs for the case of zero degrees yaw. Although the CFD results for total drag do not predict a ‘negative drag’, as reported by Zipp for their wind tunnel testing, the trends are again in very strong agreement. A local minimum in drag is seen for the CFD results at 8 degrees yaw, as compared to the ‘negative drag’ result for 12 degrees. At the higher yaw angles, an oscillation in drag force is seen both experimentally and computationally. The authors were unable to find wind tunnel data specifically for the Rolf Sestriere. However, data for other wheels having a similar box rim geometry and spoke count is available. The data for the Campagnolo Shamal, (a wheel having a deeper rim than the Rolf Sestriere, but not as deep as the Zipp 404, with a 16-spoke count), from Greenwell et.al.\textsuperscript{7} and Tew et.al.\textsuperscript{11} is in fairly reasonable agreement with the CFD results despite the differences in geometry. Data for the HED TriSpoke is seen to agree well with wind tunnel data for several competitive trispoke wheels from other manufacturers at zero degrees yaw. In addition the trend agreement with the drag coefficient versus yaw, reported by Zipp, and to a lesser degree Greenwell et.al.\textsuperscript{7}, is quite strong, with the drag showing a maximum value at around 5 to 10 degrees, and exhibiting a weaker dependence on yaw angle when compared with the other wheels studied here.

It is notable that the CFD predictions appear to be lower than the wind tunnel results provided by Zipp for all wheels, with the only slight exception being the Zipp Sub9 wheel at a yaw angle of 12.5 degrees. This may be due in part to differences between the boundary conditions applied for the CFD simulations and those present in the wind tunnel. The most critical difference is the absence of a moving ground plane in the wind tunnel. Another significant source of difference may be attributable to often unmatched experimental speed of the rotating wheel and the upstream air flow speed within the tunnel itself. In considering the wide range of differences between wind tunnel geometry and balance
hardware, the mounting fixtures to hold (and possibly turn) the wheels, and the differences in testing protocols that the experimental data has been generated from, the agreement with CFD predictions is seen to be highly encouraging.

B. Resolved Forces and Moments for RANS calculations

Forces for each of the wheels have been resolved into separate components for drag, side and vertical forces. For the case of drag, forces have been further broken down in Figure 7 to show the separate contributions from pressure and viscous forces.
In general, a considerable variation is observed between wheels. As the depth of the rim increases, the extent of the total drag minimum is also seen to increase. This drag minimum is seen to occur for the Zipp 404, Zipp 808, Zipp 1080 and Zipp Sub9 at a range of 8 to 10 degrees yaw – in real world riding conditions, this range of yaw is considered to be predominantly experienced by cyclists, and is regarded as a performance target by wheel manufacturers. In examining the drag contribution solely from pressure, we see that the total drag is largely dominated by this source. For the case of the Zipp Sub9 wheel, the pressure drag was seen to fall below zero for some yaw angles. Viscous drag contributions, also seen to vary widely between wheels, are small but are much higher than those previously reported by Godo et al.22. For the case of the Rolf Sestriere, the viscous drag appears to be nearly constant over the range of yaw angles studied. However, as the rim depth increases the viscous contribution is seen to be higher at low angles where the flow is likely to be attached to a higher percentage of the total rim surface. As the flow begins to separate off of the wheel rim (as is expected with increasing yaw), the viscous drag is seen to drop

Experimental results for the side force were observed to generally increase in a nearly linear fashion with respect to increasing yaw as noted by Greenwell et.al.7 and Tew et.al.11. In Figure 8, we also observed this trend for all of the wheels studied. Not surprisingly, the side force at a given yaw angle is generally proportional to the rim depth of the wheel. The rim depth on the HED TriSpoke is actually less compared to the Zipp 404, however the added surface area of the three large spokes produces higher side forces than the Zipp 404.

![Figure 8: Total Side Forces for all Wheels as a Function of Yaw Angle](image)

![Figure 9: Total Vertical Forces for all Wheels as a Function of Yaw Angle](image)
Vertical force components shown in Figure 9 have not been reported for wind tunnel studies. In our calculations we noted an interesting transition where these forces switched from acting downward to acting upward at a yaw angle between 5 to 8 degrees for the deep rim wheels and the disc wheel. The yaw angle at which this occurs does not seem to be directly related to rim depth. The Zipp 808 is seen to exhibit the transition at the lowest yaw angle, followed by the Zipp 404. The Zipp 1080 and the Zipp Sub9 appear to make the transition at about the same yaw angle. The Rolf also exhibits this transitional behavior which was unexpected since this transition was previously considered to be associated only with the rim of deeper rimmed wheels. The HED H3 TriSpoke wheel was seen to exhibit a very different functional dependence on yaw – this is most likely due to the fact that for the RANS calculation, the large spokes are not rotating. It is reasonable to expect that the position of these spokes will have a significant impact on the vertical force component.

There has been some question as to whether the vertical force predicted in the RANS simulations is relevant. In Figure 10, the ratio of vertical force to the drag force has been plotted as a function of yaw angle. The results for the Zipp Sub9 are not shown; the very low drag forces at higher yaw angles makes interpretation of this ratio difficult; under certain conditions, the vertical force can be several times larger than the drag force. For the deeper rim wheels, specifically the Zipp 808 and Zipp 1080, the vertical force is quite relevant at angles above 10 degrees.

To further explore the stability of each of the wheels, side forces acting on the tire, rim, spokes and hub were integrated from the leading edge to the trailing edge. Profiles of the integrated side forces at 20mph are illustrated in Figure 11 for the Zipp 404 and Zipp Sub9 wheels.
The plot illustrating the side forces on the Zipp 404 was considered to be representative for all of the other spoked wheels. In general side forces are seen to be high at the leading and trailing edges of the wheel. The local increase in side forces at the center of the wheel is due to the forces acting on the hub. The forces on the Zipp Sub9, and to a lesser degree on the Zipp 1080 show a different trend. A large increase in the side force was noted on the leading edge of the wheel, with the maximum side force occurring midway between the hub and the front of the wheel.

The turning moments computed from the integrated side forces are shown for each wheel as a function of yaw angle in Figure 12. For the wheels with shallower rim depths (Rolf Sestriere, Zipp 404, Zipp808 and HED TriSpoke), the turning moments were observed to exhibit a maximum at a yaw angle of approximately 8 to 10 degrees. For the Zipp 1080 and the Zipp Sub9, forces acting on the leading edge created turning moments which were seen to increase in a negative direction, forcing the rider to ‘pull into the wind’ as yaw angles increased. For the Zipp Sub9, the observation of this strong moment is not a concern for stability since disc wheels are not ridden on the front of a racing bike. However, the Zipp 1080 is used as a front wheel for racing, and as a result strong turning moments will have relevance to stability and handling.

It is notable that there is little difference between 20mph and 30mph in trends for the resolved forces and moments for all wheels. This should make the task of wheel production by manufacturers and wheel selection easier since the competitive rider can expect the wheels simulated here to offer performance benefits at either speed.

C. Circumferential Resolved Forces for RANS calculations

To better understand the resolved forces acting on the tire and rim, circumferential plots were generated by performing integrations on a series of 1° arc segments, plotted along the wheel circumference. Results are presented for all wheels at 20mph only; the results for 30mph are qualitatively similar. In these illustrations, the effective motion of the wheel is from left to right, with the wheel rotating in a clockwise direction. The leading and trailing edges of the wheel are at circumferential angles of 0 and 180 degrees; the ground contact and top of the wheel are at 90 and 270 degrees, respectively.
Total drag forces are plotted for each wheel in Figure 13. All of the spoked wheels (with the exception of the HED TriSpoke) exhibit the same characteristic behavior. The drag force reaches a local maximum at the front of the wheel for all yaw angles. On the trailing edge, a significant degree of disruption, caused by the upstream flow from the leading edge and the spokes can be seen.

This disruption is most pronounced for the Zipp 1080. There is the least variation over the range of yaw angles for the Rolf Sestriere. As the rim depth of the wheel increases, the variation with respect to yaw angle increases. The 10 degree yaw condition produced circumferential drag curves which were among the lowest of all yaw angles for the spoked wheels. The lowest circumferential drag at the leading edge of the wheel for the Zipp 1080 did occur at the 10 degree yaw condition. It is believed that flow remains attached at the front of the spoked wheels up to 10 degrees in most cases, and beyond this yaw angle, the flow becomes detached, causing the sudden increase in drag. The circumferential results for the HED H3 TriSpoke are somewhat unrepresentative since they only apply to the point in the rotation of the wheel matching the modeled spoke positions. However, it is clear that the presence of the spokes strongly dominates the local drag values around the circumference of the wheel. The Zipp Sub9 results show that there is a region of positive drag in the leading section of the wheel for yaw angles up to 8 degrees. At 10 degrees and above (with the exception of 12 degrees), the drag over the entire front half of the wheel is negative. The drag on the upper and lower trailing quadrants of the disc wheel becomes positive again, indicating that flow may be reattaching after becoming initially separated.

*Figure 13: Circumferential Total Drag Force for all Wheels for all Yaw Angles*
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Figure 14: Circumferential Viscous Drag for all Wheel for all Yaw Angles

Figure 15: Circumferential Side Force for all Wheel for all Yaw Angles
Similar to Figure 7, the pressure drag contribution to the total drag dominates, and circumferential plots (not shown) show this to be the case. In Figure 14, the viscous drag contributions are shown for all wheels. Again, there is the least variation with respect to yaw for the Rolf Sestriere wheel. Circumferential plots show that viscous forces are highest on the top section of the wheel, where the greatest speed difference between the forward rotation of the wheel and oncoming air flow due to the forward motion of the bike occurs. In the trailing upper quadrant of the wheel, the viscous forces are disrupted, again likely due to flow separation. As the yaw angle increase, the viscous force is seen to get smaller. The extent of the viscous force reduction increases as the depth of the rim increases. This observation is true for all spoked wheels, and also extends to the Zipp Sub9 disc as well. The circumferential viscous forces on the HED H3 TriSpoke are once again dominated by the position of the spokes. There is only a very slight variation with respect to yaw angle.

Circumferential side forces are presented in Figure 15. For all wheels, side forces are seen to increase as the yaw angle increases. The variation with yaw angle is the smallest for the Rolf Sestriere and is the greatest for the Zipp Sub9. The circumferential side force plots also reveal that as the rim depth increases, the side forces on the leading edge of the wheel become larger than the side forces on the trailing edge of the wheel. This shift with respect to increasing yaw is responsible for the change in sign of the turning moment, previously illustrated in Figure 12. For the case of the HED H3 TriSpoke, the highest side force occurs on the spoke which is closest to the ground contact. The side force on this spoke is greatly in excess of the force on the other two spokes. We speculate that there may be an interaction between the spoke descending towards the ground which is responsible for the characteristic sound that this wheel is known to make.
Circumferential vertical forces are presented in Figure 16. A positive vertical force is observed on the top section of all wheels. A positive vertical force is also seen at a circumferential angle of approximately 30 degrees, on all spoked wheels. As the rim depth increases, and as the yaw increases, a region of positive vertical force is seen to grow in the lower trailing quadrant of each spoke wheel. So, at higher yaw angles, we see that the deeper rim wheels are more effective at producing positive vertical force around the full wheel circumference. This observation does not hold true for the Zipp Sub9, where the circumferential zone of positive vertical force is limited to just the upper half of the wheel. For the HED TriSpoke, the large spokes again dominate the location of maximum positive vertical force.

D. Resolved Forces for DDES Calculations

The resolved forces and moment, including the tire, wheel rim, hub and spokes (or disc for the case of the Zipp Sub9), calculated over the last 256 solution time steps for each wheel, run for the case of 10 degrees yaw at 20mph, are illustrated in Figures 17 thru 20.

In all the time history plots, the axis on the bottom abscissa represented seconds. The axis on the top of the plot represented the rotational angle of the wheel for the last 256 time steps covering 512 degrees of wheel rotation. The amplitude and frequency of the resolved forces and moments were seen to be different for each of the wheels studied. For the HED H3 TriSpoke only 4 periodic cycles were observed. These peaks on the time history plots corresponded with the passage of the wide spokes. Drag, side force and moment time history plots for the other wheels showed several more cycles, with the greatest number of cycles being recorded for the Rolf Sestriere. As the wheel rim depth increased, the number of cycles was seen to decrease and the amplitude of oscillation was generally seen to increase, exhibiting a maximum for the Zipp 1080. Time history results for the Zipp Sub9 were observed to have many more small fluctuations relative to the results for the other wheels. This observation suggested that the flow field for the Zipp Sub9 was much more computationally unstable relative to the other wheels examined.
Figure 17: Drag Force versus Time and Power Density Spectra for all Wheels at 10° Yaw, 20mph
Figure 18: Side Force versus Time and Power Density Spectra for all Wheels at 10° Yaw, 20mph
Figure 19: Vertical Force versus Time and Power Density Spectra for all Wheels at 10° Yaw, 20mph
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Figure 20: Moments versus Time and Power Density Spectra for all Wheels at 10° Yaw, 20mph
Power spectral densities computed using Fourier transforms for each resolved force and the moments, for each wheel, are shown, plotted against the Strouhal number, $St, St = \frac{fd}{V}$ in Figures 17 thru 20. The dominant Strouhal number peak for each wheel, based on the resolved forces and moments are summarized in Table 1.

<table>
<thead>
<tr>
<th></th>
<th>Drag</th>
<th>Side</th>
<th>Vertical</th>
<th>Moment</th>
</tr>
</thead>
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<tr>
<td>Rolf Sestriere</td>
<td>6.043</td>
<td>6.043</td>
<td>4.700</td>
<td>6.043</td>
</tr>
<tr>
<td>Zipp 404</td>
<td>4.924</td>
<td>4.700</td>
<td>4.700</td>
<td>4.924</td>
</tr>
<tr>
<td>Zipp 808</td>
<td>4.924</td>
<td>4.924</td>
<td>4.700</td>
<td>4.924</td>
</tr>
<tr>
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<td>2.910</td>
<td>2.910</td>
<td>-</td>
<td>1.790</td>
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<td>HED H3 TriSpoke</td>
<td>0.895</td>
<td>0.895</td>
<td>0.895</td>
<td>0.895</td>
</tr>
<tr>
<td>Zipp Sub9</td>
<td>0.895</td>
<td>1.567</td>
<td>-</td>
<td>0.895</td>
</tr>
</tbody>
</table>

*Table 1: Strouhal Numbers for Resolved Forces and Moments*

For the HED H3 TriSpoke we observed a dominant peak at a Strouhal number which matched the passage of the wide spokes. None 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. For all wheels (not including the HED H3 TriSpoke), the Strouhal number was generally seen to decrease as the as the rim depth of the wheel increased. This observation suggests that flow is remaining attached to the deeper rim wheels longer relative to the shallower rims. Also, the variation in Strouhal number for each of the wheels suggests that this measure may be a unique characteristic of each wheel.

In Table 2, the resolved forces calculated using DDES and averaged over the last 256 time steps, are compared against the RANS results, for the basic components of each wheel. The smallest difference between the averaged DDES and RANS results were seen for the Rolf Sestriere wheel. Significant differences however were observed for the total drag resolved for the wheel component (tire plus rim) for both the Zipp 1080 and Zipp Sub9 wheels. The same hub geometry was used for the Rolf Sestriere, Zipp 404, Zipp 808 and Zipp 1080 wheels. The drag on the hub for these wheels was observed to be nearly the same, and further, the averaged DDES and RANS results were also in very strong agreement. Agreement between averaged DDES and RANS for the drag on the spokes for each of the spoked wheels was also observed to be good. Averaged DDES and RANS side forces for all components of all wheels were also in very good agreement. The averaged DDES and RANS vertical forces were in near agreement with most of the components on each of the wheels. This is a somewhat surprising result since vertical forces at the 10 degree yaw angle tend to be near the transitional point where the direction switches from downward to upward.
The Zipp 1080 and Zipp Sub9 have the highest surface area of the wheels studied. For the case of the resolved drag on these wheels, the significant difference between averaged DDES and RANS results suggests that further work will be needed with the DDES cases to resolve these differences. It is encouraging to note that the resolved side and vertical forces and the drag on components such as the hub, do seem to be more consistent between the two solution methodologies. This in itself is somewhat unexpected, as the DDES results are expected to exhibit some differences from the RANS results.

<table>
<thead>
<tr>
<th>wheel</th>
<th>hub</th>
<th>spokes</th>
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</thead>
<tbody>
<tr>
<td>Rolf Sestriere</td>
<td>DDES (averaged)</td>
<td>0.352 0.018 0.922</td>
</tr>
<tr>
<td></td>
<td>RANS</td>
<td>0.352 0.013 0.953</td>
</tr>
<tr>
<td>Zipp 404</td>
<td>DDES (averaged)</td>
<td>0.320 0.035 1.867</td>
</tr>
<tr>
<td></td>
<td>RANS</td>
<td>0.299 0.045 2.067</td>
</tr>
<tr>
<td>Zipp 808</td>
<td>DDES (averaged)</td>
<td>0.334 0.046 2.696</td>
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<tr>
<td></td>
<td>RANS</td>
<td>0.258 0.064 2.965</td>
</tr>
<tr>
<td>Zipp 1080</td>
<td>DDES (averaged)</td>
<td>0.338 -0.049 4.887</td>
</tr>
<tr>
<td></td>
<td>RANS</td>
<td>0.080 0.033 6.002</td>
</tr>
<tr>
<td>HED H3 TriSpoke</td>
<td>DDES (averaged)</td>
<td>0.294 0.016 0.533</td>
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<tr>
<td></td>
<td>RANS</td>
<td>0.312 0.038 0.683</td>
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<tr>
<td>Zipp Sub9</td>
<td>DDES (averaged)</td>
<td>0.080 0.056 5.581</td>
</tr>
<tr>
<td></td>
<td>RANS</td>
<td>-0.008 0.055 5.418</td>
</tr>
</tbody>
</table>

Table 2: Resolved Forces, by Component, Comparing Averaged DDES vs RANS

E. Flow Features

Streamlines were used to visualize flow structures for the RANS calculations in order to investigate the nature of transitions and local drag minima at the critical yaw angle of 10 degrees. Streamline seeds were placed on the suction side of the each of the wheels near the leading edge. The resulting streamline trajectories, shown in Figure 21, were colored using the lateral velocity component perpendicular to the axial flow. For each wheel, a recirculation zone, starting near the top of the wheel and extending forward behind the rim and tire was observed. In general, as the rim depth was increased, the size of the recirculation zone increased. The extent of the recirculation zone was also seen to grow from the top of the wheel, following the leading edge of the wheel, in the direction of wheel rotation, as the rim depth increased.
For the case of the Zipp 1080 and the Zipp Sub9, this recirculation zone reached nearly to the ground plane. The HED H3 TriSpoke recirculation zone was not as clearly defined. Once again, the set position of the wide spokes had a significant impact on streamline trajectories. In order to fully resolve flow structures for this wheel, it is strongly recommended that analysis of the transient results is needed.

The transient DDES results were also used to visualize flow features. A seeding pattern containing 720 to 900 individual seed locations (depending on the wheel) was created on the suction side of the wheel. The seeds were located within 5mm of the wheel surface, and massless particles were released constantly from these locations for every time step. To provide visual clarity, the visibility of the particles coming from the seed locations was controlled to follow the rotation of the moving wheel. The calculated streaklines for each wheel at four separate time steps are shown in Figure 22. The massless particles were colored by the lateral velocity component perpendicular to the axial flow. For the case of the Rolf Sestriere, most of the seed particles moved away from the tire and rim. Recirculation zones were seen to develop at the top and bottom of the wheel. Some periodic shedding of the flow, indicated by the row-like structures located immediately behind the leading edge of the wheel was also observed. For the Zipp 404, an inner recirculation behind the leading edge of the wheel was clearly visible in the first frame (Step: 314). A tight recirculation zone being driven off of the top of the wheel was also observed.

A second recirculation zone, coming from the inner part of the rim near the top of the wheel was also present. Similar to the Rolf Sestriere, periodic shedding, evidenced by the row-like structures behind the leading edge of the rim was again seen. Also a larger recirculation zone at the bottom of the wheel (relative to the two at the top of the wheel) was also present. The Zipp 808 was also seen to have an inner recirculation behind the leading edge of the wheel, again clearly visible in the first frame (Step: 314). The two recirculation zones coming off of the top of the wheel, similar to the Zipp 404 were also present. The periodic shedding was still present, however, the massless particles were observed to be carried along with the wheel rim. The recirculation zone at the ground plane was seen to be disrupted and diffuse relative to...
the same structures present for both the Rolf Sestriere and the Zipp 404. Flow structures coming off of the Zipp 1080 were more difficult to discern since the massless particles were being strongly carried along the wheel rim and were only getting separated away from the wheel, predominantly at either the top or bottom of the wheel. The recirculation zones coming from the top of the wheel were observed to undulate in the vertical direction, suggesting that the fluid particles contained in this zone possessed a wider range of velocities relative to the shallower rimmed wheels. The recirculation zones at the top and bottom of the Zipp Sub9 were much larger in size relative to other wheels and the periodic release of large groups of massless particles was clearly evident. The HED H3 TriSpoke exhibited strong recirculation zones at the top and bottom which were significantly disrupted by the passage of the spokes.

Figure 22: Streaklines Showing Recirculation on the Suction Side of all Wheels at 10° Yaw
IV. Discussion

In this work CFD was used to explore the complex and unsteady nature of air flow around several commercially available bicycle wheels: The Rolf Sestriere, Zipp 404, Zipp 808, Zipp1080, HED H3 TriSpoke and Zipp Sub9 disc. A methodology, previously presented, was used to model each wheel with realistic spoke (or disc) rotation, using a realistic ground contact. Steady RANS computations were performed using the commercial solver, AcuSolve™, over a range of two speeds and 10 yaw angles, producing a total of 120 cases. Unsteady calculations, run using the DDES functionality of AcuSolve™ were carried out for each of the six wheels at the condition of 20mph and 10 degrees yaw. To resolve and understand these results, automated methodologies were developed using FieldView FVX™ scripts to handle the repetitive tasks of i) resolving the pressure and viscous forces on the wheel into axial drag, side (or lift) and vertical components and moments, ii) breaking the resolved forces into component contributions, iii) calculating the resolved forces along the wheel circumference, iv) preparing and reformatting the computed forces for seamless transfer into other software packages for further analysis, v) computation of streaklines and vi) visualization and animation of streaklines. A newly conceived workflow methodology involving both the sequential and simultaneous execution of GAMBIT and TGrid journals, AcuSolve™ and FieldView FVX™ scripts was introduced to substantially increase the throughput of data analysis for this work.

The CFD predictions of drag forces showed significant differences between wheels. For yaw angles in the range of 5 to 15 degrees, (those most commonly experienced by cyclists in practice), the deeper rim wheels offered a clear advantage. Agreement between our CFD results and the experimental drag force data, kindly provided by Zipp Speed Weaponry (Indiana, USA) for this work, along with several other published sources, was seen to be very good. Drag coefficients at zero degrees yaw were well within ranges published for all wheels studied. Also, trends of drag force as a function of yaw angle were well reproduced. In most cases, the drag coefficients predicted by CFD were lower than the wind tunnel data from Zipp. We speculate that this arises from differences between boundary conditions used in our simulations and the wind tunnel conditions. The most notable difference is the non-moving ground plane in the wind tunnel.

Plotting resolved forces around the wheel circumference shows significant differences between wheels. Drag forces on each of the spoked wheels shows considerable variation on the trailing half of the wheel which is likely due to flow separation coming from the leading edge of the wheels. It appears that up to a critical yaw angle of 10 degrees, for most wheels, the flow remains attached and as the yaw angle increases past this, we see an increase in the drag. This effect is the most pronounced for the Zipp 1080, which is the deepest rim studied here. Circumferential drag forces on the HED H3 TriSpoke are dominated by the position of the spokes. In order to obtain a representative understanding of how these large spokes affect the drag, all simulations for these wheels should be run as transient cases using DDES.

Zipp has previously reported the observation of a negative drag force on their Zipp Sub9 wheel. Although we were not able to reproduce that result for the total resolved drag, we did see the pressure drag component drop below zero on this wheel for some yaw angles. And, circumferential total drag forces on the Zipp Sub9 were seen to be negative for most of the leading half of the wheel, above the 10 degree yaw condition. A potential reason for why we were not able to fully match this particular experimental result is that the CFD model for this wheel was based on a notional geometry. Another critical variable was the choice of tire. In our study, a different tire (Continental Tubular tire) than that used to obtain the negative drag results was modeled. We recommend that simulations for this Zipp Sub9 wheel be re-run, using a verified production geometry and the correct tire shape, based on the tire model and inflation used to obtain the wind tunnel results. We further speculate that it may be possible to optimize toroidal rim profile between the tire and inner flat disc portion of the wheel to control the degree of flow attachment for this wheel, thereby improving its overall performance.
For the deeper rimmed wheels, specifically the Zipp 1080 and the Zipp Sub9, we observed a greater number iterations and higher relaxation values required to obtain convergence, a greater sensitivity of resolved forces to the grid (and a need for finer grids), and significant differences in the resolved drag on the tire and rim when comparing RANS and averaged DDES results. It is important to note that differences between RANS and averaged DDES results were observed to be relatively small, when comparing all other wheel components for each of the resolved forces. Refinement to the grid, particularly on the rim and the disc for the case of the Zipp Sub9, and a review of the solver methodology should be considered.

Side forces were observed to be in good qualitative agreement with previously published wind tunnel studies, and were well within expectation. Essentially as the area of the rim and spokes (or disc) was increased, the side force also increased. Resolved side forces were used to calculate turning moments. Several wheels (Rolf Sestriere, Zipp 404, Zipp 808 and the HED H3 TriSpoke) demonstrated a maximum moment at a yaw angle between 8 and 10 degrees. Turning moments for the Zipp 1080 and Zipp Sub9 acted in the opposite direction relative to the other wheels. A circumferential plot of the resolved side force on the Rolf Sestriere shows a relatively even distribution over all yaw angles. However, as the rim depth increases, we see the balance of side forces shift from the trailing half of the wheel to the leading half. It is this front loading of the wheel which forces the rider to ‘pull into the wind’ at higher yaw angles. This effect is particularly pronounced for the Zipp 1080 and the Zipp Sub9. Although the Zipp Sub9 is not used as a front wheel, thereby making turning moment data irrelevent, the Zipp 1080 is used as a front wheel. There is clearly a concern that the effort being saved by a rider due to the reduced drag may be compromised by the need for the rider to apply a turning force to the aerobars (or handlebars) to maintain the direction of travel of the bicycle.

The presence of resolved vertical forces, acting either up or down, depending on the yaw angle was again observed. Previously, the values presented for the Zipp 404 were small relative to the drag forces. In this work we have shown that the vertical forces are relatively more important for the deeper rim wheels, again raising the question of whether this force could contribute to instability. Based on a ratio of the vertical force to the drag force, it appears that vertical force effect is effectively neutralized over yaw angles from 6 to 12 degrees. At higher yaw angles, this ratio can become quite large. Consequently, we conclude that under normal riding conditions, the vertical force will have little effect on stability. At higher yaw angles, which could be experienced on windy days, we speculate that control of the deeper rims will be more important.

Transient studies in this work focused on a critical yaw angle of 10 degrees. We first note that there are very significant differences between all of the wheels, and further, that the resolved forces and moments exhibit strong periodic behavior. Streakline animations, showing very strong periodic shedding coming from the leading inner rim and outer tire on the suction side of all wheels, support this statement. Of all the wheels studied, only the HED H3 TriSpoke demonstrated a correlation, for all resolved forces, between the Strouhal Number (=0.895) and the rotational motion of the spokes. This is not surprising given that the surface area of the spokes on the triSpoke makes up 85% of the total area of the spokes and rim. By comparison, the spokes for the Zipp 1080 make up less than 5% of the total area of the spokes and rim. We propose that the dominant periodic behavior observed for each of the wheels comes from the periodic shedding of fluid off of the rim (and tire) of the rotating wheel. As the rim depth increases, the flow tends to stay attached to the wheel longer, and the Strouhal number is seen to go down. For the Rolf Sestriere, the wheel with the most limited ability to maintain attached flow, the Strouhal number is St=6.043; for the Zipp 1080, the wheel with the greatest ability to maintain attached flow (aside from the Zipp Sub9) the Strouhal number is St=2.910. We believe that the periodic nature of shedding from deep rim wheels can be fundamentally described with a unique Strouhal number (or numbers) for that wheel.
In this study, transient runs were limited to study one yaw angle. Since it is known that the yaw angle can have an effect on the Strouhal number for the same wheel\textsuperscript{22}, it is strongly recommended that future studies be carried out on these wheels over a range of yaw angles.

An objective of this work was to apply a working CFD methodology to study the performance of commercial bicycle wheels. The very strong agreement with experimental wind tunnel studies suggests that the approach we outline holds considerable promise. Owing to the flexibility of this methodology, it is now possible to use CFD to provide more definitive answers on some of the open questions within the competitive cycling and triathlon communities. Issues such as the optimum wheel rim depth and cross-sectional profile, spoke count and shape, wheel diameter (650c or 700c) and tire size would benefit significantly from an independent critical analysis. Finally, we feel that inclusion and optimization of the front fork and frame and brake calipers in conjunction with a specific wheel is a straightforward extension of this work, and that such optimization studies would lead to significant performance improvements.

**Acknowledgments**

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The authors would also like to express their thanks to Josh Poertner, Category Manager – Technical Director of Zipp Speed Weaponry, for providing us with the production wind tunnel data that was used in this work.

**References**

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