SAE TECHNICAL PAPER SERIES

2004-01-3554

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Charles Hugh Ping
Auburn University

Reprinted From: Proceedings of the 2004 SAE Motorsports Engineering Conference and Exhibition

Motorsports Engineering Conference and Exhibition
Dearborn, Michigan
November 30-December 3, 2004

SAE International™

400 Commonwealth Drive, Warrendale, PA 15096-0001 U.S.A. Tel: (724) 776-4841 Fax: (724) 776-5760 Web: www.sae.org

ISSN 0148-7191

Printed in USA

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ABSTRACT

Even with relatively unrestrictive rules in the Formula SAE competition, established teams are fighting diminishing returns in vehicle mass and engine horsepower. The typical FSAE vehicle incorporates a six speed gearbox, yet reaches a (course-limited) top speed in competition of only about 110 kph. Selecting a final drive for this top speed would result in 5 gearshifts in less than 4 seconds. As a result, final drive ratio is very sensitive to shift delay time. Although vehicle mass, engine performance and traction still play a major role, a typical FSAE vehicle acceleration is significantly limited by the time it takes to complete a gearshift.

INTRODUCTION

Specifying the final drive ratio for a Formula SAE vehicle presents a unique problem. Typically, final drive ratio selection is a straightforward process, especially when gear ratios and primary reductions are fixed. In most forms of motorsport, the final drive is tailored so that the vehicle's top speed in high gear is equivalent to intended top speed on a given course. Some factor of safety must be provided to prevent exceeding engine RPM limits, and there may be some compromise due to other course attributes, but the process is primarily simple and straightforward. This method provides the greatest amount of thrust in each gear.

The typical FSAE vehicle incorporates a 5 or 6 speed transmission as part of its motorcycle-based engine. Top competition speed is approximately 110 kph. Applying a typical final drive ratio that utilizes all gears would result in 5 gear changes in less than 4 seconds. While such gearing would provide the best thrust in each gear, it is obvious that other factors, primarily shift delay time, become increasingly important. In any vehicle, a shorter duration shift delay time will result in an improved acceleration time. However, if vehicle final drive is optimized in simulation in accordance with shift delay time, it can be shown that the typical FSAE vehicle's acceleration can be more drastically improved.

SIMULATION

Because of the non-linearity of the problem, a simulation code was written to analyze the acceleration characteristics of a FSAE vehicle.

SIMULATION PARAMETERS

The 2003 Auburn FSAE vehicle was used as a basis for the simulation, utilizing its known parameters (CG location, wheelbase, mass, etc). Physical testing was performed on the vehicle to acquire simulation inputs. These inputs included torque numbers at the wheels, vehicle rolling resistance and a longitudinal tire model (wheel slip vs. acceleration).

The simulation is a bicycle model, with no lateral forces or slip angle (purely longitudinal). Weight transfer is simple, assuming a fixed suspension.

Inertia of the engine is included and multiplied by all gear reductions. The driveline inertia is neglected, assuming that the engine inertia value can be used as a compensating factor.

A modified Pacejka94 tire model of a FSAE-specification tire is used. The model proved to be inaccurate at FSAE normal forces (approximately 1700 N), and required adjustments to the horizontal shift variables to provide a reasonable curve.

The Auburn FSAE vehicle uses Hoosier tires of similar, but not identical, size and construction. Slip ratio versus thrust force data from acceleration data-logs was used to validate this compromise, and small changes to the peak value variables in the model provided improved approximations.

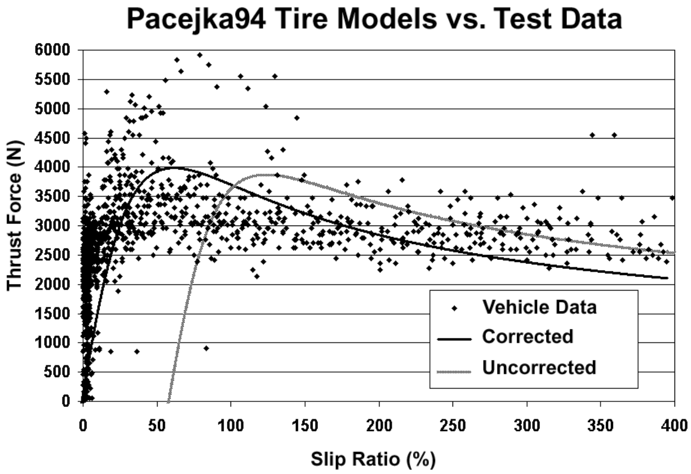

Figure 1: Comparison of vehicle tire data vs. Pacejka curves

SIMULATION VERIFICATION

To verify the simulation, acceleration testing was performed. Vehicle speed, driven wheel speed, longitudinal acceleration, distance, RPM, and throttle position were logged at 50 Hz to compare to simulation. Inertia values, rolling resistance, and drag coefficient variables were adjusted to match logged data to simulation outputs (See Figure 2). Special consideration was given to matching shift timing along the timed acceleration (shown by drops to zero acceleration). The tire model and slip ratio data was crucial for this correlation to work accurately.

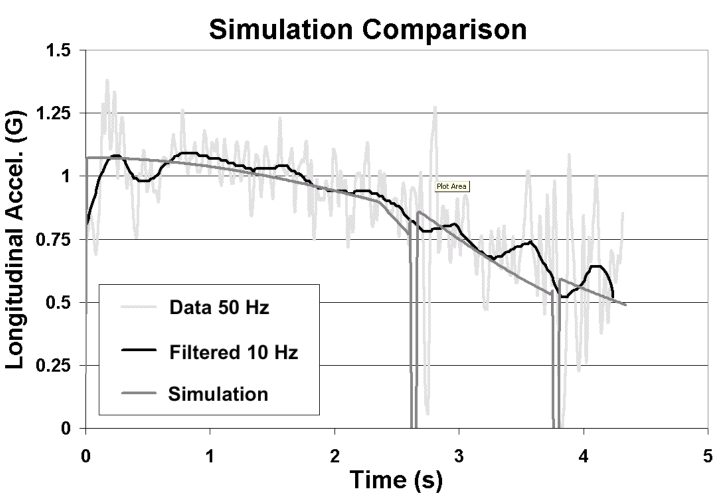

Figure 2: Comparison of vehicle acceleration data vs. simulation

Because shift delay time is one of the desired parameters from the simulation, it was important to define it so it could be easily quantified. It was decided that the shift delay time would be measured as the time the vehicle was no longer accelerating, based on longitudinal acceleration. This takes into account the delay of the entire vehicle as it reacts to a gear change, and will be a longer interval than the shift time itself. Shift delay was measured 'peak-to-peak' with an accelerometer, and the simulation shift delay was adjusted accordingly. As a result of this definition the simulation shift delay time is directly comparable to test data.

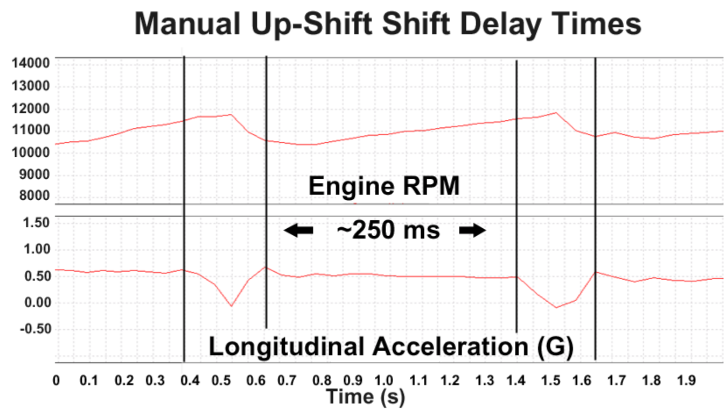

Figure 3: Definition and quantification of shift delay time

SIMULATION CONCLUSIONS

Once verified, shift delay time was varied from 1.0 to 0.1 seconds. For each shift delay time, final drive ratio is plotted in Figure 4:

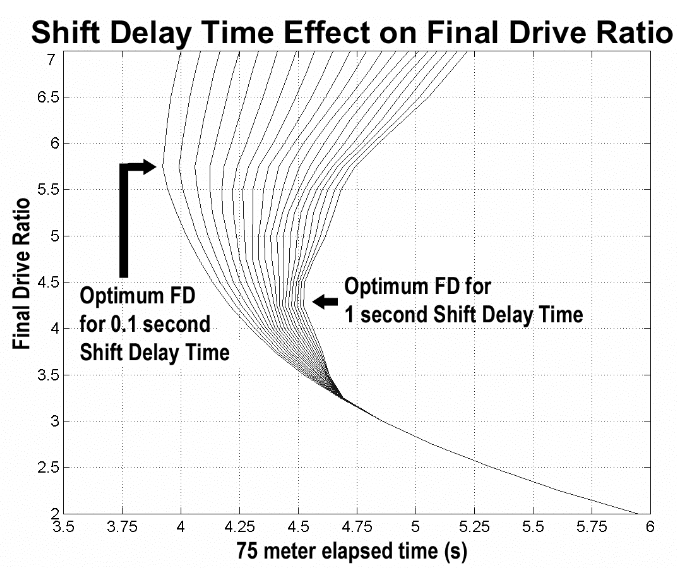

Figure 4: Simulation output for varying shift delay times and FD ratio.

Figure 4 shows that even with a shift delay time of nearly zero, there is still a limit to the final drive reduction. Acceleration time begins to drop off before a 5th gear change, even if shift delay time is zero. However, it is clear that with faster shift delay times the optimum final drive reduction increases.

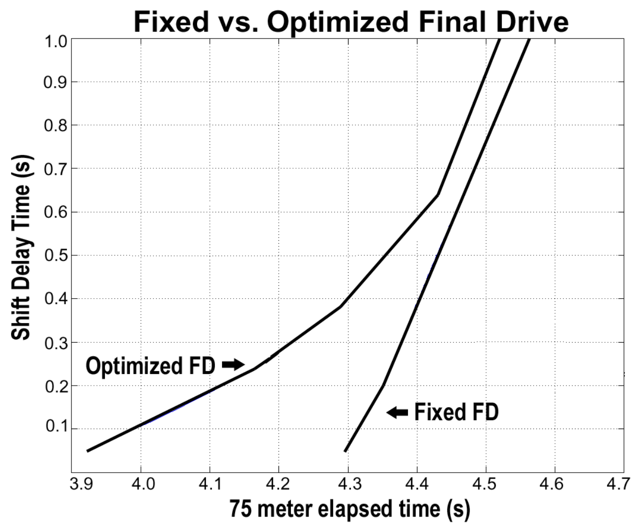

Figure 5: Increasing returns from shift delay time when FD ratio is changed

Figure 5 was created by running the simulation through the same shift delay time range, once with a fixed final drive ratio, and again with a final drive ratio optimized for the new shift delay time. As shown, if final drive ratio is held constant, shift delay time improvement is approximately linear. However, if final drive ratio optimization is included in the shift delay time benefit, it offers increasing returns. Even if shift time is already of short duration, small reductions have measurable gains in acceleration time.

SHIFT DELAY TIME IMPROVEMENT

On the 2003 Auburn FSAE vehicle, an electro-pneumatic shifting system was already in place. The system was simple and robust, with its main advantages being driver comfort and clutch-less operation. An ignition timing retard allowed full throttle up-shifts. Buttons placed on the steering wheel allowed shifting without hand removal. The circuitry was simple—as long as the button was depressed, the pneumatic cylinder applied pressure to the shift arm. Shift delay time with this system varied, but averaged about 0.25s (see Figure 3).

Although the existing shifting system was capable of fast shifts, two major problems with the design surfaced. The manual operation made shift delay time driver dependent—ignition retard continued for as long as the button was depressed. Additionally, even with short duration shift times it would be extremely difficult to properly time the shifts. With shifting rates reaching 1 shift per second, shift timing is critical.

For this reason, an automatic up-shift mode was created and implemented in the 2003 Auburn FSAE vehicle after the Detroit event. The automatic up-shift is activated at the driver's option with a momentary button on the steering wheel. If the driver is expecting a shift, holding down the momentary button will result in a gear change at the proper RPM, based on current gear. The shift delay is automatic and adjustable, which allows the shortest consistent shift time possible. If the button remains depressed after the shift, the vehicle will shift again if it reaches another shift point. Because the system is driven through the dashboard data logger, wheel slip ratio can be included in the logic to prevent shifts due to wheel spin. By adding an automatic momentary button, the driver is still in control, and still commands up-shifts. The previous system is still in place for downshifting, or if manual up-shifts are desired.

By adjusting the automatic up-shifting system, consistent up-shifts averaging 100ms were achieved. See Figure 6. The data was logged at 50 Hz, and remains unfiltered. Because of the shift time duration achieved, filtering of the data at 10 Hz results in loss of the shift delay time dip. 150 Hz+ logging with a 50 Hz filter would result in more accurate shift delay time determination, but the rate is beyond the output rate on the current sensor.

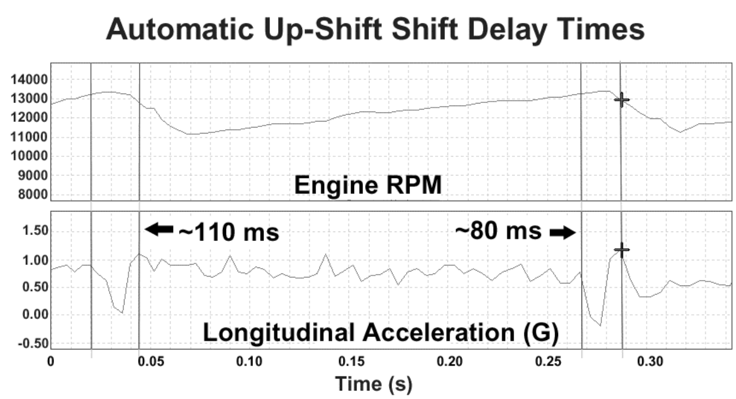

Figure 6: Quantification of automatic up-shift shift delay time improvement

VEHICLE PERFORMANCE RESULTS

The 2003 Auburn FSAE vehicle was designed to be comfortable and predictable. This design criterion included a broad torque curve, with above 48 N-m available from 6000-11000 RPM. The benefit is a car that responds well to driver demands, even if gear selection and other driver-induced conditions are not perfect. The drawback is a fairly low horsepower peak, and a powertrain design not considered favorable for acceleration only. Indeed, while the endurance event RPM histogram varies from 6000-14000 RPM, less than half of that RPM range is used in the acceleration event.

The powertrain design philosophy puts the car at a distinct disadvantage in the acceleration event. The 2003 vehicle had a mass of 223 kg without driver, and had a peak engine brake horsepower of around 72. In manual shifting testing, the car ran the 75m acceleration event in about 4.4s. In the 2003 event, under wet conditions, the manual-shifted car had a best time of 4.65s.

The automatic up-shift system mentioned earlier was implemented before the same vehicle competed in the FSAE-A event. The same 75m distance was covered in 4.13s in favorable conditions. The time was within .1s of the winning car, which was some 30 kg lighter, and had a higher claimed power output.

The 2004 Auburn FSAE vehicle mass was 7 kg lighter than 2003, with a similar peak power output. The final drive ratio was changed from 4.61 to 4.75, though packaging was the limiting factor. The 75m time was 4.19. This time was within .05s of the winning car, which was slightly lighter and claimed over 20 additional brake horsepower.

OTHER CONSIDERATIONS

Obviously, the Formula SAE event is more than an acceleration contest. This is easily shown by the compromises made in the torque curve.

SHIFTING IN OTHER EVENTS

In the endurance and autocross events, a large final drive reduction can result in more shifting. While up-shifting on the vehicle is automatic, multiple downshifts in relatively short braking zones can be difficult to master. The broad torque curve helps to offset this problem, allowing the driver to accelerate through an improper gear selection. Depending on the minimum speed of the course, this may not be an issue, but still important to consider.

TRACTION

Extreme final drive reductions result in very high thrust forces at the tire patch in the lower gears. While a good driver can use this to his advantage, for amateur drivers it may be a problem. Traction control becomes increasingly important as the final drive reduction increases, and should be a consideration.

CONCLUSION

A close analysis of shift delay time and its relationship with final drive reduction has resulted in significant gains in acceleration for a FSAE vehicle. Although some significant gains have been made in shift delay time, further gains will still result in significant acceleration performance improvement and should be pursued. While proper final drive ratio determination of a Formula SAE vehicle may be more complex than other racing series, it also offers greater returns in performance.

ACKNOWLEDGMENTS

Thanks sent to everyone on the Auburn Formula SAE team from 2000-2004. Appreciation goes to the faculty at Auburn University Samuel Ginn College of Engineering, especially Dean Larry Benefield. Individual thanks to Matthew Heffernan, Matthew Zorn, James Ray, Brian Freeman, Randy Whitehead, and Hugh Fellows.

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CONTACT

The Author was a member of Auburn University's Formula SAE team from 2000-2004, and team captain in 2003. The author is expected to graduate with a B.S.M.E from Auburn University in December 2004. You may contact Charles Ping at cpmaverick@gmail.com or 334-559-2689.

DEFINITIONS, ACRONYMS, ABBREVIATIONS

Shift Delay Time: Time during which a vehicle ceases normal acceleration during a gearshift. It is measured as the duration of the 'zero spike' in longitudinal acceleration.

Final Drive Ratio (FD): Gear ratio after the transmission but before the driveshafts; this ratio factors into all gears.

Pacejka94: A mathematical model that closely replicates the behavior of a tire on a rolling road. 'Pacejka94' is the 1994 version of the Pacejka models.

SIMULATION EQUATIONS

Slip Ratio (SR):

$SR = (\Omega R_\ell/V)-1$

Ω = Wheel Angular Velocity
R_ℓ = Loaded Radius of Tire
V = Forward Velocity

Input (Driveshaft) Torque (T_{in}):

$T_{in} = T_e * P_r * G * FD$

T_e = Engine Crankshaft Torque
P_r = Primary Reduction (Internal)
G = Transmission Gear Ratio
FD = Final Drive Ratio

Thrust Force (F_T):

$F_T = T_{in} / R_\ell$

Rolling Resistance (F_R):

$F_R = (SR+1)*(T_{in} /R_\ell) – F_T$

Aerodynamic Drag Force (F_D):

$F_D = D * V^2$

D = Drag Coefficient

Longitudinal Acceleration (A_X)

$A_X = (F_X-F_R-F_D)/(m+(I_e*G^2*F_D^2)/R_\ell^2)$

m = Mass of Vehicle & Driver
I_e = Inertia of Engine

PACEJKA94 EQUATIONS

Shape Factor (C):

$C = B_0$

Peak Factor (D):

$D = (B_1*F_Z^2+B_2*F_Z) * DLON$

Stiffness Factor (B):

$B = BCD/(C*D)$

$BCD = ((B_3*F_Z^2+B_4*F_Z)*\exp (-B_5*F_Z))* BCDLON$

Horizontal Shift (S_h):

$S_h = B9*F_Z+B10$

Vertical Shift (S_v):

$S_v=B_{11}*F_Z+B_{12}$

Composite (X_1):

$X_1 = (S_R+S_h)$

Curvature Factor (E):

$E = ((B_6*F_Z^2+B_7)*F_Z+B_8)*(1.D0-(B_{13}*SIGN (1.D0, X_1))))$

$F_X = (D*SIN(C*ATAN (B*X_1-E*(B*X_1-ATAN (B*X_1))))) +S_v$