

Investigating and Improving Vehicle Transient Handling Performance

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ABSTRACT

Steady-state handling design targets are generally defined in the design of suspension and steering systems, which are achieved through kinematic and compliance analysis. It has been found that some vehicles that meet all steady-state handling design targets do not necessarily perform well in subjective evaluations by experts or delight customers in the marketplace. The vehicles that customer find desirable exhibit the desired transient handling behavior, which is what the drivers experience for a majority of the driving time. It is therefore necessary to understand how to evaluate, and most importantly, how to design and tune the chassis for the desired transient handling behavior. In this study, the key mechanisms associated with transient handling performance are presented. The appropriate handling maneuvers are determined through which vehicle transient behavior can be evaluated. The proper response channels to monitor and measure are selected. Based on a comprehensive study using multi-body dynamics simulations, the influential design factors and tuning knobs to create the desired dynamics are examined and presented.

INTRODUCTION

Drivers experience transient handling behavior (referred to hereon simply as transients) of their vehicles during most of their everyday driving time. Even though the vehicle transient dynamics have been studied as early as Olley's era [1], researchers and engineers still try to fully quantify handling performance. Many of the widely used objective metrics are mainly biased toward steady-state behavior of the vehicle. In many cases, vehicle platforms developed based on such metrics are not necessarily recognized as having the desired characteristics in the marketplace. The challenge primarily

arises from the complex nature of vehicle transient dynamics and how drivers perceive such phenomenon.

Handling can be divided into steady-state and transient regimes, as well as linear and non-linear regimes. In a broad sense, steering performance can be included in the linear range handling performance, dealing primarily with vehicle's low to mid-range lateral acceleration response to steering wheel angular and torque inputs. The main focus is the driver's perception of steering feel. In this area, the performance metrics include steering sensitivity and its linearity, steering effort and effort buildup linearity, steering stiffness and its linearity, returnability, steering work gradient sensitivity, straight line tracking ability and minimum turn diameter. Many vehicle subsystems affect the steering performance, including but not limited to the steering system.

On the other hand, handling performance deals mainly with the vehicle's responses to steering wheel angular input. The inputs are usually of somewhat higher steer rate and larger amplitudes (up to the vehicle's handling limit). The handling performance aspects include both the steady-state and transient responses in the vehicle's yaw, roll, lateral, pitch and heave motions. Similar to the steering performance, the handling performance is influenced by all the vehicle systems characteristics, including vehicle architectural parameters, suspension characteristics, tire/wheel assembly, and steering system parameters.

Drivers have different expectations in various handling regimes. In general, in the linear and steady-state regime, responses should be linear. When nonlinearity is expected (e.g. large steer input), vehicles should be such designed to achieve a decreasing response gain (e.g. decreasing steering sensitivity or increasing understeer gradient) [2, 3]. During transient events, drivers first expect the quickness in the

lateral response of the front axle. The rear axle should be 'planted on ground' (i.e. the perceived vehicle yaw motion should pivot about the rear axle). Slow body roll motion should follow the front lateral motion, not precede it. Responses in all the channels should not have excessive overshoot (perceived as lack of precision and/or stability) and be smooth (without a two-stage feel). There should not be excessive roll, yaw, pitch or heave oscillation after the initial steer input. The above expectations during transient events require that the rear axle of the vehicle generate the cornering force in the final direction quickly and with minimal sideslip angle. The roll motion should be well controlled and phased with proper amount of roll damping. The coupling between the roll, yaw, heave and pitch motions should be minimized.

Although improvement of vehicle transient stability during severe cornering can be improved using active control systems [4, 5, 6], the fundamental design parameters, such as suspension geometry, springs and damping tuning, and tire property, play a big role in the transients for all acceleration regimes. Many studies have focused on the influence of chassis characteristics on vehicles' transients [7, 8].

This study intends to examine the mechanisms associated with transients, define the appropriate transient measures and events, determine the most influential design parameters, and finally suggest appropriate tuning strategies. Vehicles' transient response based on the bicycle model has been well documented [3]. Since lateral load transfer (LLT) effectively relates a vehicle's yaw and lateral response to its roll motions, and tire lateral load transfer distribution (TLLTD) [9] affects significantly the transients, the inclusion of roll degree of freedom is critical to fully assess the transients. Detailed suspension geometry, elastic elements and damping characteristics are necessary to simulate LLT and various jacking effects. Therefore, a full vehicle multi-body dynamic model is utilized in this study.

MECHANISMS ASSOCIATED WITH VEHICLE TRANSIENT BEHAVIOUR

This part of the paper examines the mechanisms associated with vehicle transients when the vehicle transits from one steady-state to another steady-state, including the effective relaxation length concept, suspension geometry induced force inputs to the body, lateral load transfer, yaw dynamic index, and jacking effects. Understanding the consequence after the steer input and the associated mechanisms enables improving vehicles' transients.

EFFECTIVE RELAXATION LENGTH

The steady-state force/moment properties of a tire depend on vertical load, slip and camber angles, pressure, temperature, construction, rim width and etc. Due to the relaxation effect of pneumatic tires, the generation of tire lateral force at the

contact patches lags that of the tire slip angles. Tire relaxation effect is the results of the complex deformation due to the flexible nature of the tire structure. The delay in the generation of lateral force affects the dynamic response of the vehicle. Assuming that in the initial steady-state, the vehicle operates at constant forward speed on a straight and flat road and vehicle responses at all channels remain zero, the steer angle input at the front road wheels generates front tire slip angles. With rapid application of tire slip angle, camber angle and/or braking torque, e.g. during a tire test, the deformation needs time to complete in order for a tire to generate its steady-state levels of forces and moments. Considering a unit step steer input, the tire lateral dynamic force, F_y , can be approximated by:

$$F_y = C_\alpha \cdot \alpha \cdot (1 - e^{-t/\tau}) \quad (1)$$

Where C_α is the tire cornering stiffness, α is the initial slip angle, and τ is the time constant. This equation accounts for the first order lag effect, which is sufficient to provide insight into its effect on transients. The relaxation length L_α is defined as the distance the tire has to travel to yield 63.2% (i.e.: $1 - e^{-1}$) of the steady-state force after a step input (Equation 2). V_x is the forward velocity. A simple formula (Equation 3) relates lateral relaxation length to tire cornering stiffness C_α and tire lateral stiffness K_y [10]:

$$L_\alpha = V_x \cdot \tau \quad (2)$$

$$L_\alpha = C_\alpha / K_y \quad (3)$$

Combining Equations 1, 2 and 3 yields the following equation:

$$F_y = C_\alpha \cdot \alpha \cdot (1 - e^{-\frac{K_y \cdot V_x \cdot t}{C_\alpha}}) \cong K_y \cdot \alpha \cdot V_x \cdot t \cdot (1 - \frac{1}{2} \cdot \frac{K_y \cdot V_x}{C_\alpha} \cdot t + \dots) \quad (4)$$

According to Equation 4, tires with increased lateral stiffness K_y and/or increased tire cornering stiffness C_α can generate increased lateral force at a given speed V_x , slip angle α and time t , which will translate into shortened response time. Any tire design features and application factors which enable higher lateral stiffness K_y and increased tire cornering stiffness C_α can improve transient handling response.

The suspension mechanism introduces additional delay in producing lateral force to the body, similar to that of the tire relaxation effect. The additional delay is associated with

suspension link bushings and unsprung mass and inertia. The lateral force transferred to the vehicle body is compared to the tire lateral force due to tire relaxation effect alone (Figure 1), with the tire subjected to a step steer input. The plot is generated from a simplified model of suspension/tire lateral compliance (Appendix A1). The steady-state tire lateral force is set to be 1400 N. Without the suspension lateral compliance, the lateral force transmitted to the body is a first order transient force. The tire lateral force reaches 885 N (i.e. 63.2% of 1400 N) at a relaxation length of 590 mm. With the suspension lateral compliance, the lateral force response has higher frequency components. The traveled distance for the lateral force to reach the same 885 N is increased to 722 mm. The additional increase in the relaxation length of 132 mm and the time delay explains why the lateral compliance of the suspension affects the transients of vehicles.

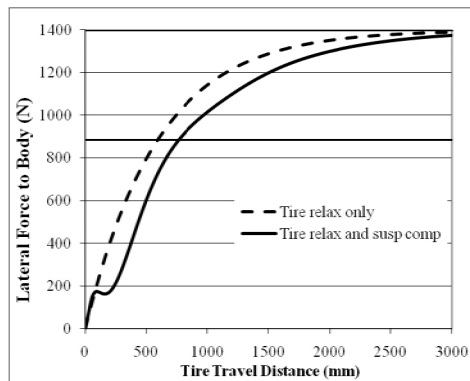


Figure 1. Tire lateral force response with and without suspension lateral compliance.

The model shows that low lateral compliance measured at the contact patch can shorten the Effective Relaxation Length (ERL) and makes the lateral force response close to that of the tire. It is also found that the camber stiffness, or the camber compliance due to camber moment, is the most significant factor influencing the total lateral stiffness as measured at the contact patch. The effective methods to increase the camber stiffness are to increase the span between the upper and lower attachment locations, as well as the bushings' lateral stiffness. Deflection of control arm bushings in the lateral direction leads to compliance steer and camber. Due to the required high lateral stiffness to shorten ERL, the tuning range for lateral force compliance steer and camber can be limited.

The above analysis also applies to the rear tires, except that the slip angle is now resulted from the lateral motion of the rear end of the body. This ERL then is a measure to describe how quickly the lateral forces are applied to the body at the rear axle. Short ERL is important to suppress overshoot in the response channels and also reduce the nonlinear two-stage feel.

SUSPENSION GEOMETRY AND FORCE INPUTS TO BODY

Suspension geometry affects how the forces generated at the contact patches are transferred to the vehicle body or frame (referred to as body hereafter). The forces generated at the contact patches act on the front of vehicle body through the suspension linkages, elastic elements and dampers. These force components generate the initial front lateral, yaw and roll motions.

A general force acting on the tire/wheel by the ground can be divided into two perpendicular components, one transmitted to the body by elastic elements and dampers and the other by the links. Conceptually, it can be shown that the application of forces by the suspension linkages to the body takes place at the roll center (RC). This assumes friction in the control arm joints, any moment from rubber bushings and the weight of links are neglected [11].

Assuming a right turn and initial roll motion to the outside of the turn, the lateral component acting on vehicle body at RC through links, F_{y_RC} , can be expressed as (see Appendix A2):

$$F_{y_RC} = (F_{yl} \cos^2 \theta_l) + F_{yr} \cos(\theta_r) + \frac{1}{2}(F_z \sin(2\theta_l) - F_r \sin(2\theta_r)) \quad (5)$$

This component is a function of the forces at the contact patches and the suspension kinematics. The first term, $(F_{yl} \cos^2 \theta_l) + F_{yr} \cos^2 \theta_r$, is the contribution from the lateral forces. Lower RC height (i.e. lower θ_l and θ_r) ensures that more lateral force at the contact patch is transferred to the body, resulting in quicker and higher lateral responses. Of course, this will induce more roll motion, since now the roll moment arm is longer. The second term is from the vertical forces. The two components may roughly cancel each other.

The vertical component acting on vehicle body at RC through links, F_{z_RC} , can be expressed as (see Appendix A2):

$$F_{z_RC} = -\frac{1}{2}(F_{yl} \cdot \sin(2\theta_l) - F_{yr} \cdot \sin(2\theta_r)) - (F_{zl} \cdot \sin^2(\theta_l) + F_{zr} \cdot \sin^2(\theta_r)) \quad (6)$$

The first term, $-\frac{1}{2}(F_{yl} \cdot \sin(2\theta_l) - F_{yr} \cdot \sin(2\theta_r))$, is the contribution from the lateral forces. During hard cornering, the lateral force at the left tire is much greater than that from the right tire ($F_{yl} \gg F_{yr}$), thus a net upward vertical force will be generated. This jacking force partially generates the initial body pitch and heave motions. The jacking force

generally also causes body roll, since the RC is not necessarily located on the centerline of the vehicle body. This jacking effect can be minimized by reducing the RC height h , (i.e. smaller θ_l and θ_r). When RC is at the ground level, lateral forces at the contact patches will not produce a net vertical input to the body. The second term is from the vertical forces. This component partially balances the static weight of the sprung mass.

BODY ROLL RESPONSE AND LATERAL LOAD TRANSFER

Since lateral load transfer (LLT) reduces the cornering power at a given axle, it thus couples the lateral and yaw motions to the roll motion. The transient LLT concept is the key to understanding transient dynamics. In steady-state, the sprung body roll angle ϕ with respect to unsprung masses is related to lateral acceleration a_y , sprung mass M_s , the front and rear wheel center roll stiffness $K_{\phi f}$ and $K_{\phi r}$, and the distance from roll axis to vehicle center of gravity h (Equation 7).

$$\phi = \frac{M_s}{K_{\phi f} + K_{\phi r} - M_s \cdot h \cdot g} \cdot a_y \quad (7)$$

There are three sources of LLT, as summarized in Table 1. The first is related to roll stiffness and body roll motion. The second and the third are related to the sprung and unsprung masses. In the table, T_f and T_r are the front and rear track width, respectively; M_{sf} and M_{uf} are the front sprung and unsprung masses, respectively; M_{sr} and M_{ur} are the rear sprung and un-sprung masses, respectively; h_f and h_r are the front and rear RC heights. h_{uf} and h_{ur} are the heights of the front and rear un-sprung mass center of gravity, respectively. An increase in RC height effectively reduces the body roll angle, thus resulting in reduced LLT due to roll stiffness, but increased LLT due to increased RC height.

Table 1. Three sources of steady-state lateral load transfer.

Sources of LLT	Front axle load transfer	Rear axle load transfer
Roll stiffness	$\frac{K_{\phi f} \cdot \phi}{T_f}$	$\frac{K_{\phi r} \cdot \phi}{T_r}$
Roll center height	$\frac{M_{sf} \cdot h_f \cdot a_y}{T_f}$	$\frac{M_{sr} \cdot h_r \cdot a_y}{T_r}$
Unsprung mass	$\frac{M_{uf} \cdot h_{uf} \cdot a_y}{T_f}$	$\frac{M_{ur} \cdot h_{ur} \cdot a_y}{T_r}$

The transient body roll motion is generated primarily by the lateral forces $F_{y_RC_f}$ and $F_{y_RC_r}$ at the front and rear RC's (the lateral acceleration is also a result of the lateral forces). A simplified expression is given in Equation 8.

$$I_{xx} \cdot \ddot{\phi} + (c_{\phi f} + c_{\phi r}) \cdot \dot{\phi} + (k_{\phi f} + k_{\phi r}) \cdot \phi = F_{y_RC_f} \cdot h_1 + F_{y_RC_r} \cdot h_2 \quad (8)$$

The roll moment due to lateral forces at RC's is counteracted by the body roll inertia, suspension roll stiffness and roll damping at both front and rear suspensions. The faster the anti-roll moments are generated, the better the roll motion can be controlled. The damper force is proportional to roll velocity and leads the roll angle by 90 degrees. This damper-generated anti-roll moment $(c_{\phi f} + c_{\phi r}) \cdot \dot{\phi}$ is an effective way to delay roll response and prevent excessive transient body roll overshoots. Low velocity damper coefficient can be dialed up to prevent early roll motion. The anti-roll moment generated by body roll angle, $(k_{\phi f} + k_{\phi r}) \cdot \phi$, will occur only after the body has undergone a roll motion.

During transients, there is an additional source of LLT due to

roll damping, expressed as $\frac{C_{\phi f} \cdot \dot{\phi}}{T_f}$ and $\frac{C_{\phi r} \cdot \dot{\phi}}{T_r}$. The cornering power at axles can be adjusted by varying roll damping distribution to achieve desirable transients, in addition to roll stiffness distribution tuning. The lead time in LLT due to roll damping makes it a more effective and flexible tuning element to influence the handling transients.

BODY LATERAL/YAW RESPONSES AND YAW DYNAMIC INDEX

The relationship between yaw moment of inertia of a vehicle and wheelbase affects how the initial lateral force is generated at the rear axle, thus influencing the transients. The

concept of $\frac{k^2}{ab}$ ratio about the vertical axis and its effect on transients were first investigated by Olley [1] (P247-P259) and further discussed and summarized in [12] (P386-P389). Since the concept is critical to understanding the transients, more detailed discussion can be found in these references. Only key points are referred to this paper.

As defined by Olley, the yaw inertia I_z and mass m are related by the radius of gyration k about the vertical axis through the center of gravity of the vehicle (see the picture in Appendix A3):

$$k^2 = \frac{I_z}{m} \quad (9)$$

The yaw dynamic index $\frac{k^2}{ab}$ is defined as the ratio between k^2 and the product of a and b , where a and b are the longitudinal distances from the center of gravity to the front axle and rear axle, respectively.

The theory states that the direction of the initial lateral motion at the rear end of the vehicle body depends on the yaw

dynamic index $\frac{k^2}{ab}$ ratio about the vertical axis: 1). When $\frac{k^2}{ab} < 1$, application of front lateral force moves the rear end outboard, thus the rear suspension and tires produce initial lateral force in the same direction as the front lateral force; 2).

When $\frac{k^2}{ab} > 1$, application of front lateral force moves the rear end inboard, thus the rear suspension and tires produce initial lateral force in the opposite direction as the front lateral force; and 3). When $\frac{k^2}{ab} = 1$, the rear axle does not move laterally and thus does not produce any lateral force.

To further elaborate, lateral forces applied to the front of the vehicle body at front RC causes the initial front lateral

acceleration and body yaw motion. When $\frac{k^2}{ab} < 1$, the initial lateral force at front RC pushes the rear end outboard, and thus generates sideslip at rear tires through links, which in turn generates lateral force in the same direction as the front lateral force. Lateral forces developed at rear tires are applied to body at rear RC. Since both front and rear lateral forces now work together to move the vehicle laterally, the body lateral acceleration builds up faster at any given time than if only the front axle generates lateral force.

On the other hand, when $\frac{k^2}{ab} > 1$, the initial lateral forces at front RC push the rear end inboard, and produce lateral force in the opposite direction of the front lateral force. The lateral force at the rear then changes direction to push the vehicle into the turn. This change of direction causes delay in the generation of lateral acceleration and gives a float feel of the rear axle.

When $\frac{k^2}{ab} = 1$, the rear end of the vehicle does not have any lateral motion. This may sound ideal, since the rear axle is 'planted on the ground'. However, since the rear axle is not generating any lateral force, its lateral capacity is not utilized. The body acceleration response time will be increased. All in

all, reduced body yaw inertia can effectively reduce $\frac{k^2}{ab}$ ratio, thus improving the transients.

BODY PITCH/BOUNCE RESPONSES AND JACKING EFFECTS

During transient maneuvers, various jacking mechanisms cause a pitch and bounce motions of the body, and influence the subjective feel of the vehicle. In addition to the jacking force due to suspension kinematics as shown in Equation 5, asymmetry in damper characteristics can cause jacking effect [13]. For example, the typical low compression and high rebound damping characteristics causes downward net force F_{d_net} to the body during the body roll motion (Figure 2 and Equation 10). Depending on the damper tuning at the other end of the vehicle, this may cause pitch motion of the body. F_{d_reb} and F_{d_comp} are the rebound and compression damper forces, associated with the damper's rebound and compression velocities, $\dot{x}_{d_reb}$ and $\dot{x}_{d_comp}$, respectively.

$$F_{d_net} = F_{d_reb}(\dot{x}_{d_reb}) - F_{d_comp}(\dot{x}_{d_comp}) > 0 \quad (10)$$

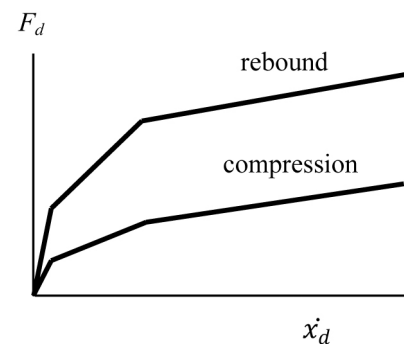


Figure 2. Jacking force due to asymmetric damping characteristics.

Additionally, stiffening suspension elastic elements (i.e. bushings, jounce bumpers and nonlinear springs) also cause jacking effects during body roll motion [8]. The compressed elastic elements generate more upward force than the rebound side loses (Figure 3 and Equation 11). The net vertical force F_{s_net} introduces the third jacking mechanism. F_{s_reb} and F_{s_comp} are the rebound and compression spring forces, generated at the damper's rebound and compression deflection, x_{s_reb} and x_{s_comp} , respectively.

$$F_{s_net} = -(F_{s_comp}(x_{s_comp}) + F_{s_reb}(x_{s_reb}) - 2 \cdot F_{s0}) < 0 \quad (11)$$

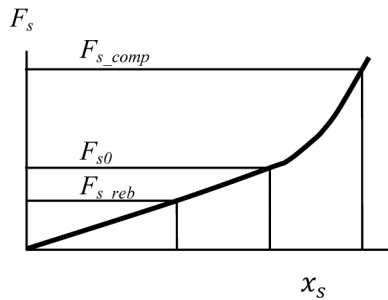


Figure 3. Jacking force due to stiffening elastic characteristics.

All the three jacking mechanisms (i.e. suspension links geometry, asymmetric damping characteristics and stiffening elastic elements), work together to cause body heave and pitch motions. The direction of the initial motions depends primarily on the overall effects from both front and rear suspensions. The transient pitch and bounce responses of the body are thus coupled to the roll motion. Among the three factors creating jacking effects, damper characteristics are the most tunable. More symmetric damping characteristics and relatively higher damping at low velocity can effectively prevent the early pitch and roll motion.

QUANTIFYING TRANSIENT HANDLING PERFORMANCE

Transient response of a vehicle represents how a vehicle's state variables transit from one steady-state condition to another. The transition is steering input and vehicle speed dependent. Different response variables may present different transient characteristics. Proper transient events and the key output channels need to be identified to quantify transients. These events should excite the transient behavior of the vehicle. The output channels selected should reflect certain aspects of the subjective feel of the transient state of the vehicle.

TRANSIENT MANEUVERS

Many events can demonstrate the transient behavior of vehicles, such as Evasive Lane Change, Consumer Reports Magazine's Avoidance Maneuver, Step Steer, Pulse Steer and etc. In this study, open-loop maneuvers, namely Step Steer and Pulse Steer are selected for their repeatability and simplicity. In the study described below, these events are simulated at a vehicle speed of 160 km/h. High vehicle speed tends to magnify the difference in vehicle transients caused by changes in design parameters.

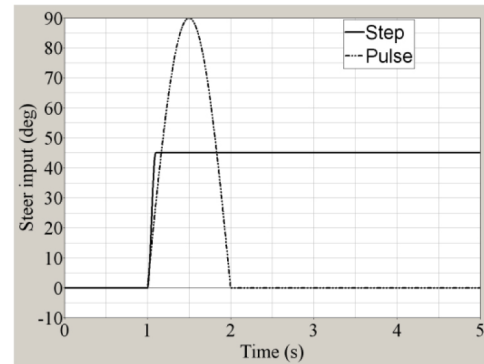


Figure 4. Step steer and pulse steer inputs.

The steer inputs are illustrated in [Figure 4](#). The amplitude of the steering angle input in step steer is 45 degree, resulting in roughly 0.7 g peak lateral acceleration at 160 km/h, and thus beyond the linear region. The ramp-up time in step steer input influences the transient behavior. Short ramp-up time can better excite the transient response. Pulse steer is a half sinusoidal wave. It is found that a pulse duration of 1.0 second can better distinguish the transient behavior with different tunings for the vehicle investigated and is thus selected. The amplitude of the input is 90 degrees which yields about 0.7 g of peak lateral acceleration at 160 km/h.

KEY CHANNELS TO QUANTIFY TRANSIENT RESPONSE

Transient state exists when a vehicle is subjected to steer input or external disturbances. The transient behavior is a function of steer input, vehicle speed and vehicle design parameters. Expert drivers use terms such as steering precision, stability, agility, and roll and yaw oscillation decay to describe the transient behavior. The typical steady-state metrics such as understeer gradient and roll gain, which are gains with respect to steady-state lateral acceleration, are not adequate. New channels need to be identified.

The relevant output channels include the lateral acceleration at both front and rear axles (for agility), sideslip angles at both front and rear axles (for steering precision and stability), roll angle (for roll stability and oscillation decay), heave and pitch motions (for oscillation decay). Yaw velocity Ω_z is an important measure for steady-state handling and is related to lateral acceleration. For transient performance, however, lateral acceleration a_y contains more information than yaw velocity, as shown in [Equation 12](#), thus it is a better measure. Besides, yaw velocity information is already included in front and rear sideslip angles, as shown in [Equation 13](#). Definitions of vehicle axle sideslip angles, axle reference sideslip angle and axle reference steer angle are given in [Appendix A3](#).