A Survey of Today’s CVT Controls

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Abstract

For over one hundred years, the principle of continuously and infinitely variable transmissions (C/IVT) has eluded practical automotive utility. With the advent of durable materials, sophisticated electronic controls, and improved lubricants, a viable assortment of C/IVT powertrain systems with innovative ratio shift control strategies is now feasible. A few of these progressive system designs are described in this survey. Pertinent fundamental relationships of C/IVT powertrains are also discussed.

1 Introduction

Internal combustion engines (ICE) have characteristic operating points of maximum fuel efficiency, minimum emissions, maximum power, and maximum torque. These features can be exploited in power transfer to the loads of automobile wheels by the use of the continuously varying transmission ratio defined as

\[ r_\omega(t) = \frac{\omega_c(t)}{\omega_e(t)} \quad \text{or} \quad r_T(t) = \frac{T_c(t)}{T_e(t)} \]

(1)

where \( r_\omega \) is referred to as the speed ratio, \( r_T \) is referred to as the torque ratio, \( \omega_e \) is the engine speed, \( \omega_c \) is the wheel speed, \( T_e \) is the engine torque, and \( T_c \) is the wheel torque. Henceforth, \( r = r_\omega(t) = r_T(t) \) denotes the CVT ratio. The pursuit of continuously variable transmissions (CVTs) began as early as 1897, when Jo-habert L. H. Maugras invented his split-torque version [6]. But only recent advancements in tribology, material science, and electronic controls have made CVTs sufficiently practical to warrant fervent interest in their design.

Traditionally, a powertrain or driveline that includes a CVT consists of an ICE, the CVT, a differential, and axles that join the differential to the wheels. Since only the transmission has variable ratios, \( \omega_c \) and \( T_c \) in Eq. (1) essentially represent the speed and torque at the transmission output as indicated in Figure 1. A system of this form usually entails combining independently designed components, while the powertrain controller may be either a system design or an amalgamation of individual component controllers. Recently emerging are powertrain system configurations designed to capitalize on the CVT and to have integral powertrain controllers.

![Diagram of CVT system](image)

**Figure 1:** Traditional arrangement of powertrain components

For any powertrain configuration, there exists numerous operating scenarios that present challenging control issues. One example is launching a vehicle from rest, which requires an infinite ratio per Eq. (1). Frictional or hydraulic coupling devices are used to launch vehicles equipped with conventional automatic transmissions having discrete ratios. These devices can also launch CVT powertrains. However, the advantages of having a CVT include maintaining a rigid connection at all times, allowing no losses even through the zero output speed operating point. Such designs are considered infinitely variable transmissions (IVTs), and the zero output speed condition is termed geared neutral. Some IVT configurations and control strategies are explored in [4, 15, 17]. Unfortunately, like this vehicle launch issue, many other intricate aspects of powertrain control and design examined and published in the surveyed works must be omitted here due to space limitations.

This survey focuses on the basic shift control strategies applied once the vehicle is already launched and operating in forward driving conditions. For each surveyed design, a brief description and some implications of the chosen hardware configuration is provided. And the shift control strategy is examined relative to the implemented design solution. Section 2 establishes the fundamental relationship that governs the dynamics of ratio shifts, engine operations, and output responses. Section 3 contains the survey of powertrain designs organized by their apparent shift strategy philosophies. Section 4 concludes the observations.
2 Fundamentals

For steady state vehicle conditions, when the driver is holding a constant throttle that is interpreted as constant power demand, amongst those surveyed, the unanimous choice for shift control strategy is to maintain engine operations on a predetermined map. This map is usually a sequence of intersections between constant power curves and minimum fuel consumption contours on an engine torque versus engine speed plot; hence, the map is termed the peak efficiency curve. A sketch from [1] is reproduced as Figure 2. Under these conditions, the system is in equilibrium; engine torque scaled by the ratio balances the torque load from wheel traction, vehicle wind resistance, road incline, etc.

\[ rT_e = T_l \]  
(2)

where \( r \) is the speed ratio defined in Eq.(1), \( T_e \) is the engine torque, and \( T_l \) is the torque load.

For transient vehicle maneuvers, or deviations from steady state driver inputs, the nonlinear dynamics are derived as follows.

\[ r(T_e - I_e \dot{\omega}_e) = T_l + I_c \dot{\omega}_c \]  
(3)

where \( r, T_e, T_l \) are the same as in Eq.(2), \( I_e \) is the engine inertia, \( \dot{\omega}_e \) is the engine acceleration, \( I_c \) is the vehicle inertia, and \( \dot{\omega}_c \) is the CVT output acceleration. Engine acceleration subtracts from the engine torque generated to drive the load and additional output acceleration through the transmission ratio. Substituting the time derivative of a re-arranged Eq.(1)

\[ \dot{\omega}_e = \dot{\omega}_c r + \omega_c \dot{r} \]  
(4)

into Eq.(3) then solving for the output acceleration, the result is

\[ \dot{\omega}_c = \frac{1}{I_c + I_e r^2} (rT_e - I_e \omega_e \dot{r} - T_l) \]  
(5)

Figure 3 demonstrates the inverse response of numerical solutions to this equation when various constant ratio rates are assumed for a particular engine torque level. Because the response starts in the opposite direction before heading toward the commanded input, it is like the nonminimum phase behavior of linear systems. For sufficiently high ratio change rates, the system does not respond in the direction of the commanded input within reasonable ratio range. This problematic behavior constrains all of the shift control strategies surveyed.

Remarkably, every implementation of a shift control strategy suits best only one of multitudes of unpredictable transients the system may incur. Because shift control strategy dictates the performance of the powertrain during transient maneuvers, and transient performance, such as acceleration and deceleration, defines the character of a vehicular system for a driver, the shift control strategy governing the transient performance of a driveline determines the character of the vehicle that a driver experiences. This renders appropriate innovation and management of strategies very important to the successful development of the powertrain.

3 Samples of Design Solutions

This section summarizes the three shift control philosophies discerned amongst the designs surveyed. Within each school of thought, strategies are revealed to vary with system configuration.

3.1 Single Track

A simple transient shift control philosophy is to maintain engine operations along the peak efficiency curve used for steady state operation, the single track approach. Chan et al. [1] achieve this strategy by controlling the speed of the ICE with an engine throttle governor and controlling the engine torque independently with the ratio variations of a generic CVT. The engine throttle governor is inferred to relate the driver input; axle torque measurements are required for feedback. Figure 2 shows two linear sections of a representative
single track. At low vehicle speeds, the axle torque is approximately constant while at high vehicle speeds engine torque is approximately constant. Where the track corners the CVT ratio control regime switches from proportional axle torque control to proportional output power control, with increasing vehicle speed. To ensure a consistently acceptable simulated response for any operating condition despite the nonlinear and time varying nature of Eq. (5), proportional controllers must be tuned for optimal gain scheduling.

Ten years after Chan et al, Wittmer et al [18] introduce the ETH hybrid vehicle, for which Shafai and Geering [14] later demonstrate the ability to maintain the ICE operating at either the peak efficiency point of maximum torque or the ultimate efficiency point of being shut down. This ICE duty-cycling is made possible by a variable speed pulley CVT with a large ratio range, which can compensate losses incurred when only a flywheel drives the powertrain while the ICE is off, as well as compensate excess gains incurred while the ICE is on. Detailed descriptions and models of variable speed pulley drives are widely available [9]. Additionally, an electric motor that is a primary source of driving power in other modes is used in this mode to cancel the CVT ratio change rate term in Eq. (5). The shift control strategy maintains the two points of ICE operation and the vehicle response to drive-by-wire input signals by varying the CVT and other power sources that do not directly consume fossil fuels. By using the Lyapunov approach to adaptively estimate uncertain parameters (e.g. load torque, driving torque, etc.), and output feedback linearizing the nonlinear state space model derived with the ratio change rate term cancelled, Shafai and Geering successfully simulate the system to meet the setpoint torque at the CVT output, the interpretation of the driver input. When the ratio change rate term cannot be completely cancelled by the electric motor, severe adverse effects result.

In their shift control strategy, Kim et al [10] use a predetermined optimal CVT speed ratio map, shown in Figure 4, to set their ICE operation along the peak efficiency curve, or their Optimal Operation Line (OOL). By equating driver input signal to a percent throttle opening, which is actuated by a stepper motor, the desired CVT ratio regulates the engine speed such that engine operation tracks the OOL. By developing effective electro-hydraulic control over the hydraulic pressure that provides their variable speed pulley CVT with the desired torque capacity, the desired CVT output torque is acquired also through the calculated optimal CVT speed ratio, which is by definition the inverse torque ratio per Eq. (1). With fuzzy feedback the actual CVT ratio is controlled to match the calculated optimal ratio setpoint. Their experiments show that the chosen shift strategy successfully results in ICE operation that is close to the OOL for gradual and relatively small input demand ranges (i.e. three 8-20% periodic throttle cycles over 150 seconds.)

In demonstrating the versatility of his simulation package, which does not use any experimental maps, Hong [8] designs powertrains matching an ICE with a manual, a conventional automatic, and a continuously variable transmission for comparison. For the CVT model, he uses the single track approach similar to that of Chan et al; the driver input is simulated as proportional and integral control on engine speed error and the predetermined ratio is a smooth function of engine speed, depicted in Figure 5. His simulation technique side-steps the nonminimum phase behavior of the ratio change rate by assuming that all ratio changes are made gradually with the following discrete model.

$$\omega_c = \omega_{c,0} + (\omega_{c,0} + \omega_{c,0} - \Delta \omega) \cdot (1 - e^{-\Delta t / \alpha})$$  (6)

where $\Delta t$ is the time step during acceleration or deceleration, and $\alpha$ is the time constant of the rotational system including inertias and damping.

3.2 Speed Envelope
Extending the idea of setting the desired speed ratio as a smooth function, Deacon et al [2] choose to represent their shift control strategy in the form of Figure 6. The two curves in the figure, each approximated by a
polynomial in the controller, form a speed envelope in which the CVT regulates powertrain responses. For a given vehicle speed, the driver input relates almost linearly to engine speed. From the plots validating their simulation to experimental data collected from a diesel engine connected to a variable speed pulley drive CVT, further details of their speed envelope philosophy can be gleaned. When transients occur (i.e. the driver input varies), engine torque is permitted to immediately respond, followed quickly by the CVT ratio change that generates the engine acceleration corresponding to the driver input but within the confines of the speed envelope. Hence, the ratio change rate seems systematically limited by the rate of the driver input, the engine speed response, and the established speed envelope. Once the driver input reaches a steady state, a seemingly preferred engine torque is reached and maintained by continuous engine speed modulation affected by the CVT ratio. Because the stated project objective of this system is overall improvement in emissions without compromising driving performance, predilections for regions of the engine operations map can be inferred.

A slightly different implementation of the speed envelope philosophy is used by Honda [5] in their recent commercial gasoline engine vehicle. Included in their CVT shift control strategy are various driving mode selections to aid in driver input interpretation. A given mode at a given vehicle speed and throttle opening will force the CVT to modify engine speed only within a particular portion of the overall speed envelope. For example, Figure 7 shows that a portion of the envelope at high CVT input speeds, or simply engine speeds, is virtually excluded from the control solution when the driver chooses the fuel economy mode. A portion of the envelope at low engine speeds is virtually excluded from the control solution when the driver chooses the sport or performance mode. The publication details and justifies speed envelope apportioning for other modes. A PI controller regulates the variable speed pulley CVT ratio to actually meet the prescribed setpoints. To mitigate the effects of the otherwise uninhibited ratio change rates, Honda restricts the shift speed response to the two settings represented in Figure 8. For engine braking and other high power demand situations, the shift control defaults to provide maximum permissible performance and ratio change rate.

Figure 7: A simplification of the Honda multiple modes CVT shift control speed envelope

Figure 8: Slow and quick ratio rate responses

The part of the Bosch [3] Cartronic Powertrain Structure shown in Figure 9 is a flowchart of their speed envelope shift control strategy. The engine electronic throttle interprets the driver gas pedal input as an output torque demand commensurate with the adaptively identified driver style; this automates the driving mode selection manually available in Hondas. The chosen mode and input throttle interpretation prescribe the variable speed pulley CVT ratio setpoints that govern the engine speed within the speed envelope. The ratio changing rate is determined by the driver style, driving
conditions, and other available operating information processed in the “dynamic driving program.” And like the Honda, high power demands cause the shift control to default to maximum permissible performance and ratio change rate.

3.3 Off the Beaten Track
The philosophy of providing a shift control strategy for each driving mode incorporated into the design can also be reflected on the engine characteristics map with respect to the peak efficiency curve instead of on a speed envelope. Vahabzadeh and Linzell [17] propose their economy and sport mode shift strategies in the form of the two different traverses off the peak efficiency curve in Figure 10. With available engine electronic

![Figure 10: Mode dictated shift strategies off the peak efficiency curve](image)

throttle, the driver gas pedal input is correlated to the demand for engine power. Power level requirement changes are resolved by, first, finding the destination operating point where the desired power level meets the peak efficiency curve. In the fuel economical control scheme, this destination point coincides with a particular throttle level which is immediately satisfied by opening the throttle. Then the newly attained specific throttle level is maintained by modulating the engine speed through the variable speed pulley IVT ratio until the engine reaches the destination operating point. In the performance control scheme, the interpreted power level is immediately satisfied by changing the throttle as necessary at the current engine speed. This is considered utility of the available torque margin when the torque is increased. The newly attained power level is then maintained through the combination of throttle and engine speed variation until the engine reaches the destination operating point. If the torque margin is insufficient to allow the power level demanded at the given engine speed, or if the power demand is high in either driving mode, both throttle and engine speed, and hence IVT ratio, are modulated as necessary until the engine reaches the required power level and the destination operating point. Positive simulation results are achieved with the design model and shift control strategy that directly controls the ratio change rate to monitor its negative effects.

One step further off the beaten track is that in addition to providing multiple driving modes that exploit the use of C/IVTs, the engine can be redesigned to maximize fuel efficiency, minimize emissions, and optimize driving performance in concert with the C/IVT design. Smith et al. [15] propose ample redesign suggestions for the engine complemented by a double-sided, full-toroidal traction IVT. Detailed descriptions and models of toroidal traction C/IVTs are available in [11, 12, 13, 16]. Fellows and Greenwood [4] describe the shift control strategy for the toroidal IVT powered by a conventional gasoline engine. The spirit of their strategy is similar to that expounded by Vahabzadeh and Linzell for the variable speed pulley system, including monitoring of the ratio change rate effects. Imposing this shift control strategy on a redesigned engine with the characteristic map having the peak efficiency curve at significantly lower torque levels, allowing significantly higher torque margins particularly at lower engine speeds, the transient powertrain traverse resulting from driver power demand change can be pictured as in Figure 11. The traverse shown is the same power

![Figure 11: Shift control strategy traversing off the peak efficiency curve of a redesigned engine compared to the characteristics of a conventional engine](image)

level change as the performance mode traverse in Figure 10, indicated by the endpoint markers. The redesigned engine can potentially require less throttle and engine speed changes, plus, complete the traverse without reaching any engine parameter limits.

4 Conclusions

It is clear that shift control strategies for ICE-C/IVT powertrains vary with system configuration, operat-
ing philosophy, and design resolution. And the non-minimum phase dynamics of continuously varying ratios considerably constrain all design strategies. However, successful implementation of shift control strategies for ICE-C/IVT powertrains improve vehicle performance over conventional engine-transmission combinations. This achievement inspires development of criteria that quantifiably predict effectiveness of strategies and help determine choice controllers.

References


[16] Torotrak (Development) Limited, "Torotrak CVT."
