

High Performance Stop-Start System with 14 Volt Belt Alternator Starter

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ABSTRACT

As the industry strives to achieve improved fuel economy, stop-start systems for internal combustion engines are receiving additional focus. Studies and system proposals have been made for various electrical configurations ranging from 12 volts to 42 volts and higher [1, 2, 3]. Both cranking motor and belt alternator starter configurations have been proposed, with some concerns regarding the customer acceptance of the cranking motor solution [1] which were subsequently addressed [3]. Typically, 14 Volt Belt Alternator Starter (BAS) applications are limited to 1.6 liter gasoline (0.4 liter per cylinder) and 1.4 liter diesel (0.35 liter per cylinder) engines due to BAS torque output constraints. Some previous work extended this range to 0.7 liter per cylinder by utilizing a boosted supply voltage and auxiliary energy storage device [1]. This paper details a 2.4 liter 4 cylinder GDI (0.6 liter per cylinder) application of a non-boosted 14 V BAS system with integrated battery state of health monitoring.

A key enabler for the application is a high (3.24:1) pulley ratio which reduced cranking current by 43%. The lower cranking speed did not result in longer start times as a fast-start algorithm allowing fuel injection and spark on first partial compression stroke was employed which actually reduced start time by 22%. Functionality was further improved by using the higher rotation speed of the alternator together with battery state of health monitoring for improved battery recharging and efficient energy recovery.

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INTRODUCTION

The BAS system configuration was selected for this application due to its superior noise & vibration characteristics. It also has no restart waiting period (to let the engine come to rest) due to a change in driver input while entering a vehicle stop condition. It has high durability since there are no mechanical engage / disengage events. Mueller et. al. suggested design durability targets of up to 450,000 start/stop events for an automatic transmission equipped vehicle [3]. The electrical machine has high efficiency in the generating mode (>75%) thereby reducing parasitic power loss during run mode. In addition, it has functional expandability to provide energy recuperation during deceleration or braking events.

Torque available when cranking with a typical 14 Volt Belt Alternator Starter will be limited to around 35 Nm at 100 RPM. The 2.4 liter 4 cylinder GDI engine used in this study required a peak torque of 82 Nm to overcome compression derived torque, which occurs just before the first Top Dead Center transition. Typical applications use a BAS pulley of 57 mm diameter, for packaging reasons the minimum pulley

diameter is 46 mm. A 46 mm drive pulley, while practical, imposes significant challenges with regards to belt tensioning, routing and durability. In this study a 46 mm pulley was successfully applied using 190 degrees of belt wrap and a single automatic belt tensioner providing 270 N of belt tension.

The standard crank shaft accessory drive pulley had a diameter of 149 mm which coupled to the 46 mm BAS pulley yields a 3.24:1 ratio and a nominal available cranking torque of 113 Nm. While this is sufficient to overcome the peak compression load alone there is insufficient margin to overcome other parasitic loads such as valvetrain. An unintended byproduct of the higher-than-normal pulley ratio is the reduced cranking speed and higher maximum speed when generating. The BAS used in this study had a maximum rated speed of 18,000 RPM which was acceptable for this application since an engine down speeding strategy was applied, limiting peak speed to 5,000 RPM. In the body of this paper we will present the modeling techniques used to determine peak starting torque, followed by the impact of various strategies which enable reliable, fast restarts. In order

to maximize fuel economy benefit and assure reliable restarts a Battery State of Health algorithm was applied.

Robinette and Powell provide a comprehensive discussion of 12 V stop/start issues including vehicle integration, startability, noise, vibration, and vehicle launch [2]. One of their conclusions is to avoid dwelling at engine speeds that align with high vehicle sensitivity (typically 150 - 450 RPM), and to transition through this speed range quickly during start events. To achieve this goal, their system was designed to achieve fast cranking speeds (300 - 400 RPM) and to synchronize the ECM, fuel and fire on the second compression event. They also concluded that achieving the fast cranking speed was difficult due to the high current draw and correspondingly reduced system voltage. Thus, for drivability and noise reasons, we concluded it is desirable to achieve 600 RPM within 400 ms after cranking commences, and maintain that minimum speed for the duration of the start event. For our system configuration, this requires the ECM to maintain synchronization through the stop event and to fuel and fire on the first compression event. This requires knowledge of the engine coast down stop position and the ability to estimate cylinder oxygen mass in the absence of a consistent intake stroke. Three methods by which engine stop position may be determined were examined together with an early combustion control technique.

PUMPING MODEL

Much of the work described in this paper requires estimation of current crankshaft state in terms of the torques applied; for this reason a Simulink® [5] discrete time model was developed. The model, while similar to earlier models [6], was intended for embedded application in a real time controller being used to determine cylinder pressure and volume for fuel injection quantity. Figure 1 illustrates the architecture of the model; a single cylinder is shown in the figure though the model contains all cylinders.

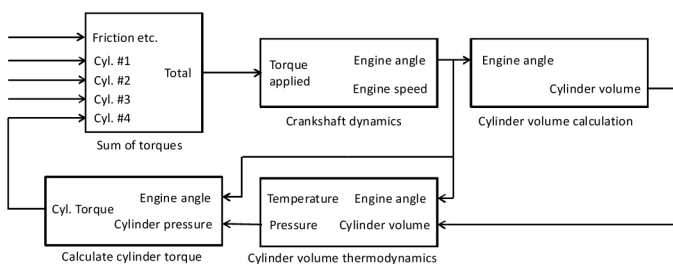


Figure 1. Discrete time engine model

At the heart of the model is a crankshaft dynamic block containing the 3 integral terms required to estimate acceleration, speed and angle using total applied torque. Using the crankshaft angle, the volume in each cylinder is calculated. Knowing valve timing the polytropic gas law is used to estimate cylinder air pressure and temperature. Using engine angle together with cylinder pressure and connecting rod geometry, individual cylinder torque can be calculated.

The sum of these torques provide the input to the crankshaft dynamic block which through the discrete time method is iteratively solvable.

ENGINE STOP POSITION DETERMINATION

Typically, crank angle is determined by means of a toothed timing wheel attached to the crankshaft whereby each successive tooth is detected by use of a single magnetic-field-based proximity sensor, either variable reluctance, or increasingly, a hall effect type. There will be a discontinuity in the tooth spacing with one or more missing teeth which allows an absolute timing position to be established. Thus, with each successive tooth edge transition, engine angle is known. Crank angle sensors of this type are unable to determine direction of rotation, so when slowing to rest after a fuel shutoff, a reversal in direction cannot be measured directly. For this reason when the engine is re-started there will be a period of time where the true engine angle is unknown and cannot be established until one or more 'gap' transitions occur. The consequence of this is that unless true engine angle can be determined after a reversal, the scheduling of combustion events must wait until a gap transition occurs and synchronization is restored. Thus a strategy to use zero speed combustion as a cranking assist method is prohibited. Additionally, at re-start the potential to use fast-start which can inject and combust on as many as 3 or more compression strokes is lost, leading to extended start times. Knowing and retaining engine stop position is therefore critical to achieving fast, reliable restarts. It was determined during development that to be effective, the estimated stop position needs to be accurate within ± 12 degrees. Various methods potentially capable of achieving this have been evaluated and will be discussed in the next three sections.

ADAPTIVE-MODEL-BASED STOP POSITION ESTIMATION

Using a suitably calibrated and validated pumping model, it is possible to force the model state (speed, angle) to be the same as the physical engine. At a suitable time prior to the sensed crank angle becoming unreliable, the simulated engine angle is delinked from the physical engine and becomes the source of the estimated true engine angle. Most of the torques acting on the crankshaft (pumping, cam etc.) as it slows to rest can be calculated or determined under test conditions. Coulomb friction, however, is highly dependent on engine conditions and age. Errors in estimating coulomb friction can lead to significant error in stop position estimation. To mitigate this problem, the model was modified to include a learning mode such that during the coast down, at each compression stroke TDC, the model-estimated crank angle and actual crank angle are compared. The difference between estimated and actual crank angle is used as an error function to adjust the coulomb friction term for the next compression

cycle. Coulomb friction is typically updated 5 or more times during each coast down to rest event.

Equipping the engine with an absolute crank angle sensor allowed the accuracy of the model to be assessed. **Figure 2** shows a full coast down to rest with 5 TDC transitions. The top trace shows crank angle derived from the absolute reference sensor (red), the normal crank sensor (blue) and the model-based estimation (green). This coast down event concluded with an engine reversal. The model-based estimator accurately followed the true engine position while the unadjusted crank sensor response deviated substantially. The middle trace shows engine speed (crank-sensor-based in blue, model-based in green) and indicates the TDC transition points (red) where the model state is synchronized and the coulomb friction updated. The bottom trace shows the coulomb friction value adapting after each TDC event.

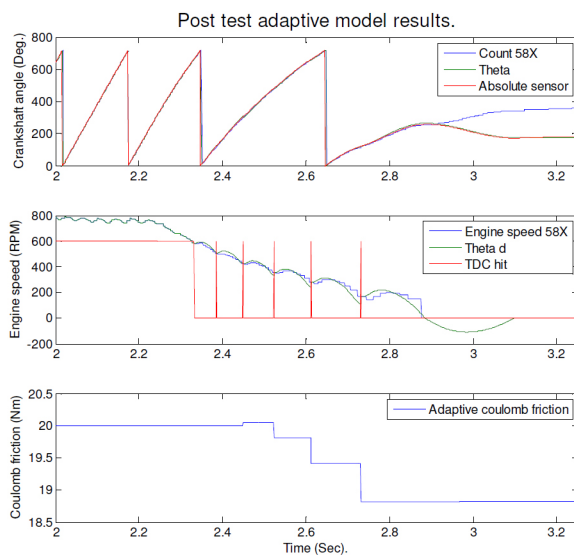


Figure 2. Adaptive model results

To be effective, it is necessary to accurately represent all engine torque loads, which in practice can be difficult. In real world applications the model based estimation was unable to meet the desired accuracy targets, particularly when approaching TDC at near-zero speed. At this condition the smallest error in torque can result in model crankshaft rotating in the opposite direction of the physical crankshaft resulting in large stop position error. A need for an improved method was evident.

CRANK-SENSOR-BASED STOP POSITION ESTIMATION ALGORITHM

A stop position estimation algorithm has been developed which identifies the characteristic signature of a crankshaft reversal of rotation using crankshaft speed and position. The estimation algorithm uses two methods of detection combined with location logic which prevents false reversal detection at or after TDC.

Robust Detection Method

The 'robust' detection method requires that the second of three sequential velocity samples be below a calibrated threshold and that both velocity samples 1 and 3 are greater than sample 2. **Figure 3** is an example of the early robust method in operation.

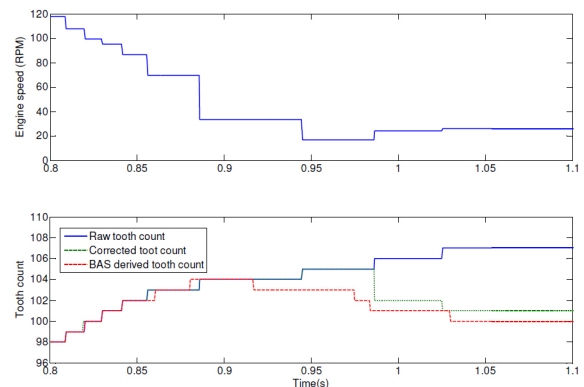


Figure 3. Reverse detection by means of robust detection method

In this example, sequential speed updates at 0.87, 0.94 and 0.97 seconds yielded sequential sample speeds of 33.3, 16.8 and 24.2 RPM, thus triggering a 'robust' reverse detection ($16.8 < \text{threshold}$, 33.3 and $24.2 > 16.8$). As such the robust detection method identifies a reversal after the fact. This can be observed in the figure at 0.94 seconds; although a reversal had occurred, the corrected tooth count incremented. To adjust for this when a 'robust' reverse is detected (1 tooth after the actual reversal) 2 additional counts are subtracted from the corrected count. The accuracy of the reverse detection point is confirmed by using the BAS hall sensor's derived rotation angle, and in this case the final rest position and the BAS derived tooth count disagree by one tooth count. The source of the BAS derived tooth count is higher resolution than the raw crank sensor (2.5 vs. 6 degrees) so a crank rotation of less than one tooth can trigger a BAS derived tooth count change. In the case of **Figure 4** the final corrected tooth was exactly correct in response to the raw tooth count updates.

There are cases where the extra speed update (sample 3) after reversal required by the 'robust' method is either missing or, due to engine friction, is not greater than sample 2. In these cases an additional detection method is employed, namely the 'early' detection method.

Early Detection Method

The 'early' detection method requires that second of two sequential velocity samples is below a calibrated threshold and the difference between the first and second sample be greater than a second threshold. **Figure 4** is an example of the early detection method in action. Engine speed is updated upon each crank sensor tooth transition; the top chart shows the crankshaft decelerating through approximately 0.93