Proceedings
of the 4th International Modelica Conference,
Hamburg, March 7-8, 2005,
Gerhard Schmitz (editor)

M. Tiller
Ford Motor Company, USA:
Evaluation of Motor and Battery Requirements for Hybrid-Electric Powertrains during Cranking
pp. 477-483

Evaluation of Motor and Battery Requirements for Hybrid-Electric Powertrains during Cranking

Michael M. Tiller
Research and Advanced Engineering, Ford Motor Company
mtiller@ford.com

Abstract

Hybrid electric vehicles (HEVs) are an emerging technology for improving fuel economy and emissions. However, hybrid powertrains are expensive to manufacture because of the sophisticated electronics required. In particular, the motor and battery requirements must be carefully considered because of the cost and weight of these components.

For this reason, it is important to conduct an upfront analysis to determine the minimum requirements for the motor and battery [1]. Such an analysis ensures that the requirements for the vehicle (acceleration, fuel economy, *etc*) can be met while minimizing the incremental cost to the consumer.

This paper describes the development of engine and transmission models used to perform such an analysis for a research vehicle project. The model must take into account several important effects such as crankshaft position, engine damper design, motor design, control strategies and so on. The multi-domain modeling capabilities of Modelica allow us to formulate a model with which all these important effects can be captured [2].

This paper will show that such a model is not only capable of helping hardware designers evaluate the performance of different electrical components but also allows experimentation with various control strategies for controlling the launch clutch and drive motor.

Keywords: System engineering, hybrid electric, VMA

1 System Engineering Process

Development of a complete vehicle is a daunting task. There are numerous regulations and constraints on the development process. In addition, while the attributes of the vehicle as a whole (performance, fuel economy, emissions, *etc*) may be specified, there is a complex relationship between the design specifications for individual components and the performance of the entire vehicle system.

For this reason, system engineering principles are used to formalize the design process [3]. As part of this process, requirements are identified during the early stages of development. These requirements are then used to define performance targets for each of the vehicle subsystems (and, in turn, their constituent components). The process is often represented by the system engineering "V" shown in Figure 1.

For the application described in this paper, we are concerned with the initial requirements cascading. Based on fuel economy analysis, we know what size motor and battery are required and how much power they need to handle. However, fuel economy is only one attribute to be considered. Because we would like to eliminate the cost and weight associated with a dedicated starter motor, we also need to verify that the motor and battery combination we have chosen will be sufficient for starting the engine.

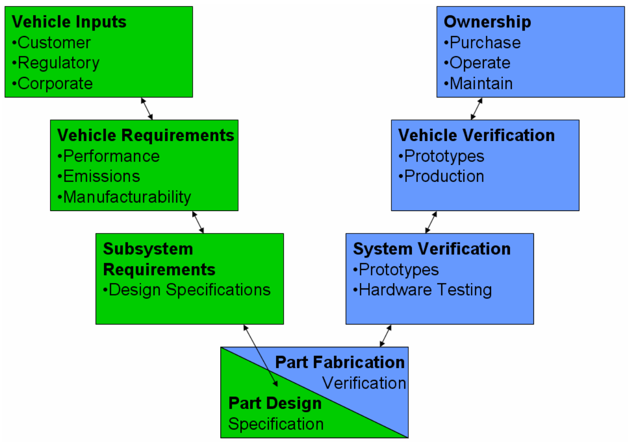

Figure 1: System Engineering Process

While it is possible to rely on "rules of thumb" or knowledge-based engineering solutions to determine requirements for conventional vehicles, it is nearly impossible to apply these to research projects. In such cases, physically-based models of the underlying systems with sufficient levels of detail and fidelity can be created that reasonably approximates the response of a physical incarnation of the design. Because design specification details should result from this process, such models need to be detailed enough to capture the effects of design changes. We term such models “design-oriented” models. In order to capture such effects it is typically necessary to make first-principles based models of the various components and use constitutive relationships based on design parameters (*e.g.* compliance, inertia, mechanical limits, *etc*) to characterize these components.

2 Powertrain Architecture

This section describes some of the relevant details about the powertrain architecture. This analysis was conducted for a research vehicle. Many of the components were relatively new and they had never been used in this particular configuration before. For this reason, models were necessary to analyze the requirements and determine component specifications.

The overall vehicle model used the Modelica Vehicle Model Architecture library [4]. Starting with the conventional vehicle architecture (shown in Figure 2), specific models for the engine and transmission were supplied that captured the physical effects required for the analysis of engine cranking. The remainder of this section will discuss each model in detail.

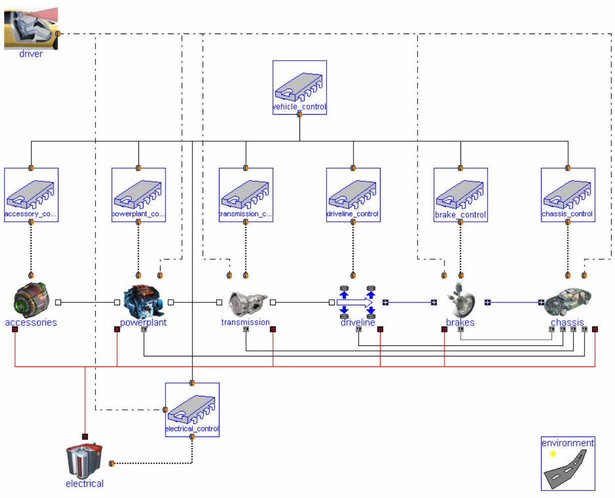

Figure 2: Vehicle Model Architecture

2.1 Engine Model

Modeling engines can be quite complicated because many factors contribute to the dynamics of the engine [5]. Because, in this application, our goal is to reach a critical engine speed in order to begin injecting fuel, we do not need to be concerned with the combustion dynamics of the engine. Instead, we focus on only those dynamics that are present before fueling begins. The engine model used is shown in Figure 3 and includes typical crank-angle based dynamics. For our analysis, two effects are particularly important.

The first effect involves the engine design itself. In particular, the compression ratio of the engine and the valve timing will determine exactly how much “resistance” is felt as we try to crank the engine. The engine configuration we are studying is a V-6 configuration so for every 720 degrees of motion in the crankshaft we will go through 6 compression events. These events normally correspond to the compression of the air-fuel mixture in preparation for combustion and the amount of work that must be done in order to perform such compression is strongly influenced by the compression ratio and valve timing of the engine.

The other effect we consider is friction. Friction is very sensitive to both engine speed and ambient thermal conditions. Friction is extremely hard to quantify because of the various non-linear effects involved (viscosity, thermal expansion, wear) and the fact that it is typically only calculated under steady state conditions for normal operating points. Because our analysis was conducted for an engine that was still in a prototype stage (without complete friction data), we will assume a conservative friction relationship and show how sensitive our results are with respect to this estimate.

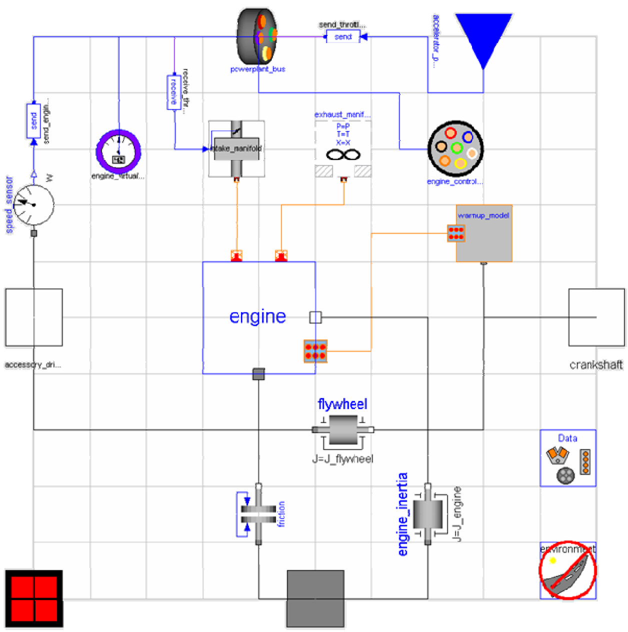

Figure 3: Engine Model

2.2 Damper Model

While the damper model is technically part of the transmission model, its design and behavior can be described independently of the other transmission components. The purpose of the "damper" is to prevent engine torque fluctuations from being propagated into the transmission and driveline. In addition to preventing these vibrations from being "felt" by the driver of the vehicle, the isolation also protects downstream components from experiencing torque reversals between combustion events leading to gear rattle and other NVH phenomena.

The damper design must be concerned with two kinds of dynamics. The first is the normal engine "torque signature" under steady operating conditions. In these cases, the damper should be as efficient as possible in transmitting energy to the transmission (to avoid a fuel economy penalty) but still isolate the fluctuations of the engine so they do not lead to downstream disturbances. The other mode involves damping of large scale disturbances (*i.e.* those that might occur as a result of pressing the accelerator pedal). It is desirable that in such circumstances the damper should "extract" energy so that these disturbances are quickly damped out.

These two, seemingly contradictory, goals are accomplished by a design, shown in Figure 4, that combines a compliant (typically multi-stage) spring in parallel with a hysteretic element surrounded by inertia elements on either side. Because of the backlash deliberately designed into the damper, the hysteresis is only triggered for large deflections (determined by the magnitude of torque carried by the element and the compliance of the spring). As a result when large disturbances are generated by the engine, the hysteresis loop removes the energy, via friction, as heat.

The damper must be tuned so that the natural frequency of the driveline is below the idle speed of the engine to avoid excitation of resonances in the driveline. However, there is also a dynamic aspect to this tuning. Because of the multi-stage design of the spring, large deflections result in the stiffer stage of the spring being involved. This increases the "effective stiffness" of the device and, as a result, raises the effective natural frequency. This leads to an interesting phenomenon. As you approach the natural frequency of the spring (for small deflections) from below, the spring will start to resonate. If this resonance leads to deflections that are large enough, the stiffer stage of the spring will begin to participate and the natural frequency will increase. If the natural frequency increases that means that a greater portion of the engine cranking will occur below the natural frequency and more resonance will occur. If this process is gradual enough, the resulting dynamics can become quite violent. To avoid this, it is desirable to move through the resonance as quickly as possible.

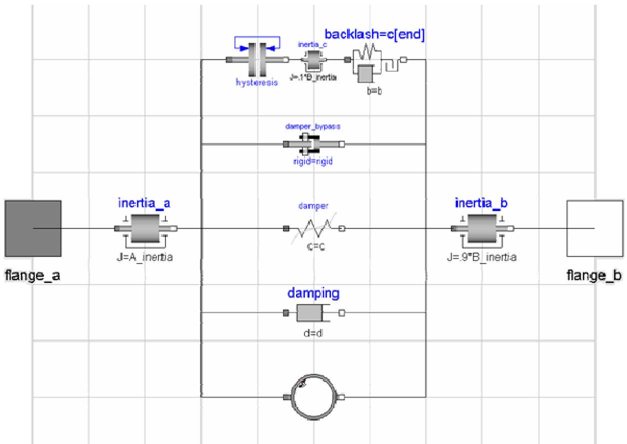

Figure 4: Damper Model

2.3 Transmission Model

Ultimately, the purpose of this analysis is to establish the power and torque requirements to crank the engine in this vehicle. Because the electric motor used for this process is contained in the transmission [6], the transmission plays an unusually key role in the starting process for this vehicle. A schematic of the transmission is shown in Figure 5.

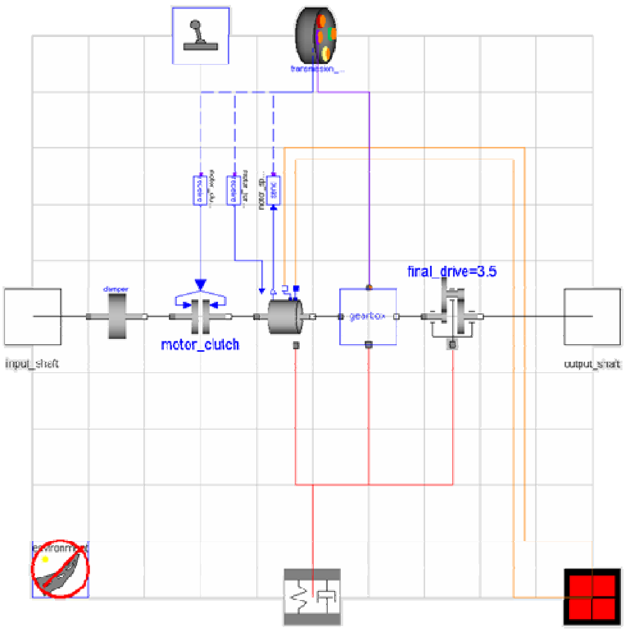

Figure 5: Transmission Model

The design of the gearbox itself is not particular important here because the transmission will not be engaged during our analysis (although it would be in a further assessment of "rolling starts"). What is important is the engagement of the motor clutch (*i.e.* the clutch that connects the motor to the engine). Another important factor is the additional inertia of the motor rotor. Although it would be nice to choose how much inertia to include in the rotor, this is largely determined by packaging constraints and the performance targets of the motor.

It is important to note that all torque used to start the engine must come from the electric motor in the transmission. There is no starting motor on the accessory side of the engine as there is in a conventional powertrain. This means that the motor design must be able to deliver tractive torque (when driving the vehicle) or cranking torque (when starting the engine). Because the peak power requirements are different, these two goals do not necessarily lead to similar designs for the motor.

2.4 Control

Starting the engine involves several discrete phases [7]. These stages are shown at the top of Figure 6. For our analysis we assume that before the engine is started the electric motor is disconnected from the engine (*i.e.* the motor clutch is disengaged) and the engine is completely at rest (phase 1). Before this motor clutch is engaged, the controller uses a PID strategy to bring the motor speed up to a specified value (phases 2 and 3). Once that setpoint has been achieved, the motor clutch is engaged (phase 4). As the clutch is engaged, torque is transmitted to the engine. The critical issue is making sure that the engine "turns over". In practice, this means that sufficient starting torque must be delivered to the engine to overcome the resistance caused by the first compression event in the engine (phase 5). The PID strategy attempts to hold the motor speed at the same setpoint during this process (phases 2-5). Once the engine has reached the desired speed (phase 6 and 7), the motor torque requirement is considerably reduced because it only needs to maintain the desired speed.

The control strategy relies on sensing two different speeds, the motor speed and the engine speed. The motor speed is known to a great degree of accuracy with very little delay. Unfortunately, the same cannot be said of the engine speed. The engine speed sensing relies on a traditional engine speed sensor which is relatively low resolution (as compared to the motor speed), is unreliable at low speeds, and is subject to considerable lag due to its design and implementation.

Using the information about the motor and engine speed, the control strategy can use two actuators, the electric motor and the launch clutch. The control strategy can specify the torque to be generated by the motor and the pressure applied to the launch clutch. Physically, this clutch pressure translates into a "clutch capacity" (*i.e.* how much torque can be transmitted through the clutch). As a result, the clutch is also effectively a torque actuator.

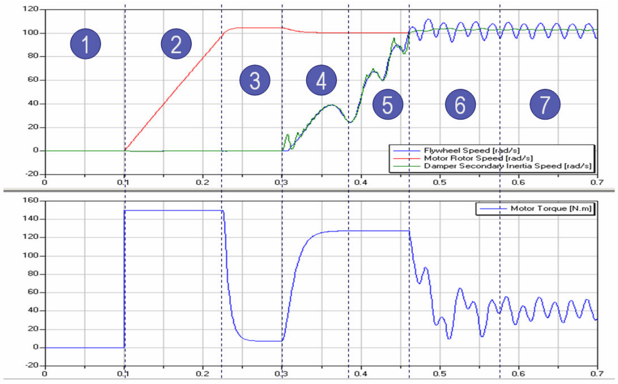

Figure 6: Baseline Analysis Results

3 Validation and Interpretation

While the use of models in system engineering to assist with target cascading and requirements analysis is useful, establishing the validity of the models used in the process is difficult. This is because, by the nature of the process, the system being engineered has not been built yet. Because design-oriented models are built using first principles, they do not rely heavily on empirical data. Instead, design data can be used to directly characterize the model.

In our case, there was existing data showing how similar hardware and control strategies functioned on a research prototype [7]. As a result, our validation focused on making sure that the response from our models matched, qualitatively, the response from actual hardware (albeit different hardware).

Figure 6 shows a typical result. Interesting qualitative features shared with in-vehicle test results:

- Saturation of motor torque during phase 2
- Magnitude of 'parasitic' loses during phase 3
- Motor torque limited to clutch capacity in phase 4 and 5
- "Brake" torque required in phases 6 and 7

A key feature of Figure 6 is the transition from phase 4 to phase 5. The boundary between these phases is defined as the point where the engine crosses the first compression event. Of particular importance during this event is the deceleration of the engine. If the engine speed approaches zero, the engine may stall (*i.e.* the vehicle will fail to start).

4 Analysis

4.1 Baseline results

Figure 6 shows a baseline response for our system. We can see the various phases of the control strategy and the results clearly indicate a successful starting of the engine (*i.e.* the engine achieved the critical speed necessary to begin fueling). We also see no evidence of any serious resonance in the driveline during cranking.

While such results highlight the important features of the experiment, there are other results that are available to us in our design-oriented model that are also useful. For example, these results confirmed that response of our damper did not exceed any of its design constraints (*i.e.* maximum deflections, maximum torque, energy dissipated, *etc*).

Another interesting feature of the simulation is the deflection of and torque transmitted through the engine mounts. While not particularly interesting for target setting and requirements analysis of the electric motor and battery, the model could be used for an additional analysis involving target setting and requirements analysis for the engine mounts.

4.2 Friction Sensitivity

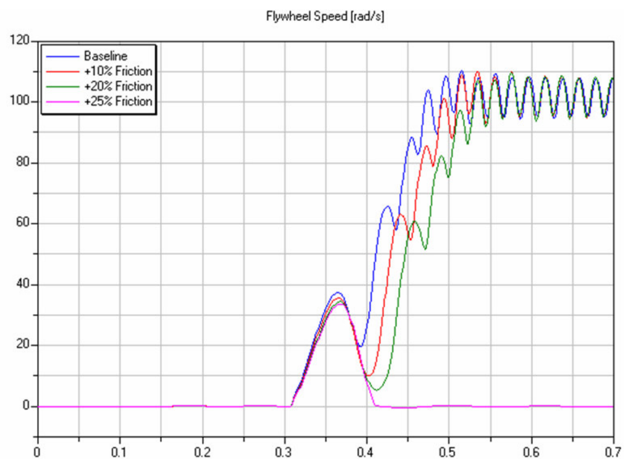

Figure 7: Friction Sensitivity

As mentioned previously, friction is a complex quantity to measure and it changes as a function of engine operating conditions. As such, we would like our analysis to be as robust as possible to our friction estimate. For this reason, it is useful to understand the sensitivity of our baseline response to different amounts of friction. From Figure 7, we see results of several different simulated experiments with different amounts of friction. All conditions are identical between Figure 6 and Figure 7 except the amount of friction. As the amount of friction is increased, the important feature to notice in Figure 7 is the dip in engine speed during the first compression event. Although the baseline case shows a successful start, an increase in friction of only 25% leads to an unsuccessful result. From this we can see that there is significant sensitivity to friction. This analysis can help us establish an upper bound on acceptable friction.

4.3 Crankshaft Position Sensitivity

Another important factor in cranking an engine is the initial position of the crankshaft. Ideally, the engine should be given as much time as possible to build up momentum as it approaches the first compression event. As shown in Figure 8, by taking our baseline case and “backing up” the starting position of the crankshaft we can significantly increase our tolerance to friction (and thereby improve our robustness to friction).

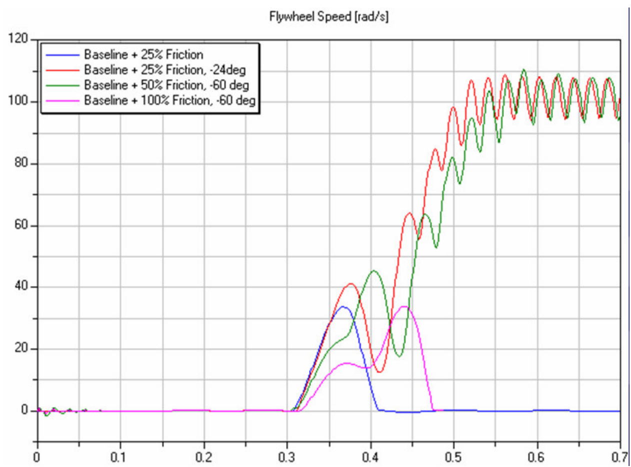

Figure 8: Crankshaft Position Sensitivity

The difficulty in this approach is that we cannot directly control the crankshaft position prior to starting the engine. So this analysis only gives us information about the fact that the results are sensitive to initial crankshaft position and highlights a need to understand the statistical variation in engine shutdown patterns (something we could also use the model to study in detail).

4.4 Performance Limits

Sections 4.2 and 4.3 addressed noise factors in the engine starting process and established sensitivities to help us gage the robustness of the procedure. Now we will turn our attention to the control strategy itself to see what we can achieve with the sensors and actuators we have available.

We will focus on two cases which we will label “Best Case” and “Worst Case”. The “Best Case” scenario is important because it shows us how much excess capability we have in our electric motor under the best circumstances. This excess capacity gives us some metric by which we can gage the potential of the system to implement rolling starts (*i.e.* starting the engine while the vehicle is moving). The “Worst Case” scenario helps us to gage the limits of our design by trying to start the engine under very difficult circumstances.

Let us first consider results from the “Worst Case” analysis shown in Figure 9. For this analysis we have specified that the setpoint for motor speed control during phase 3 (see Figure 6) should be twice the speed at which the engine should be started. This “overspeed” gives us additional momentum (built up during phases 2 and 3) that we can use to generate additional torque. Our “Worst Case” corresponds to the green line in Figure 9. What this result shows us is that by using a clutch with a torque capacity of 350 N.m., we can still start the engine in the face of 200% more friction than the baseline case. Note that the additional torque used to crank the engine comes from sacrificing momentum in the motor rotor as exhibited by the deceleration of the motor rotor during motor clutch engagement.

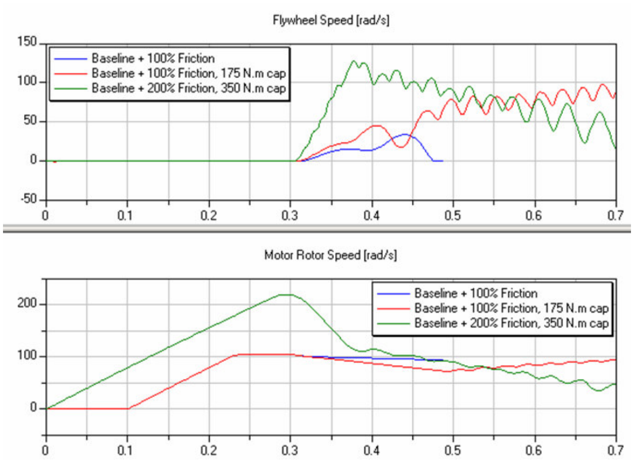

Figure 9: Worst Case Scenario

Looking at Figure 10, we see the results of our “Best Case” scenario. In this case we assume that the amount of friction to be overcome has been reduced by 40% (due, for example, to engine warm up). In such a case we see that we no longer need to use all of our motor torque to start the engine (as demonstrated by the difference in the two traces at the bottom of Figure 10). This is important because it means that we could provide some drive torque to the wheels (through the transmission gearbox) and still have enough torque left over to crank the engine. This analysis gives us some indication of how much excess is available (*i.e.* that could be used to move the vehicle forward)[1].

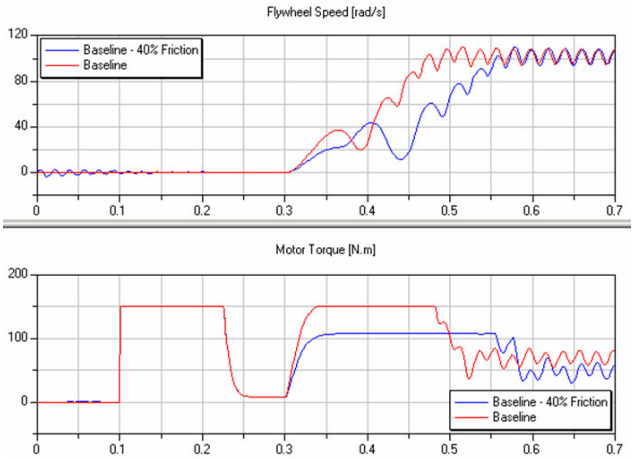

Figure 10: Best Case Scenario

5 Conclusions

The analysis in this paper supports the idea that this particular system is relatively robust with respect to the motor and battery requirements. While the response of the system is sensitive to friction and crankshaft position, the control strategy and the actuators available to it can handle the most extreme cases with enough of a safety margin.

From the analysis presented in this paper, we can see how design-oriented models can be used to guide the development of both hardware and software in the vehicle development process. Although this paper shows how this process was applied to a hybrid electric vehicle, the principle holds not only for other types of vehicles but for many product development activities in general. The key is the ability to quickly develop design-oriented models to help with upfront evaluations. This not only saves time in the development process but can save a considerable amount money by reducing or even eliminating the

[1] Of course, there are significant issues with starting the engine under such circumstances without causing significant (*i.e.* driver perceptible) driveline disturbances. However, this is beyond the scope of this paper (although **not** beyond the scope of this model).

need for prototype hardware that might have to be fabricated to support real-world testing aimed at answering the same questions.

References

[1] "Advanced Hybrid Vehicle Powertrains 2003", *SAE World Congress, 2003*. Detroit, Michigan. SAE-SP-1750.

[2] Tiller M., Tobler, W. E., Kunag, M., "Evaluating Engine Contributions to HEV Driveline Vibrations", *Proceedings of the 2nd International Modelica Conference*, Oberpfaffenhofen, Germany, 2002.

[3] Ford Design Institute, "Systems Engineering Fundamentals", 1999.

[4] Tiller M., Bowles P., and Dempsey M., "Development of a Vehicle Model Architecture in Modelica". *Proceedings of the 3rd International Modelica Conference*, Nov. 3-4, 2003. Linköping, Sweden.

[5] Batteh, J., Tiller M., Newman, C. E., "Simulation of Engine Systems in Modelica". *Proceedings of the 3rd International Modelica Conference*, Nov. 3-4, 2003. Linköping, Sweden.

[6] Ortmann W., Colvin D., Fozo S. R., Encelewski M., Kraska M., "Incorporating an Electric Machine into the Transmission Control of Ford's Modular Hybrid Transmission", SAE2004-01-0069. Society of Automotive Engineers, Inc. Warrendale, PA.

[7] Colvin D., Masterson B., "Challenges of Engine Starts and Drivability in a Parallel Hybrid-Electric System", SAE2004-01-0063. Society of Automotive Engineers, Inc. Warrendale, PA.