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A CASE STUDY OF AN INTEGRATED STARTER GENERATOR FAILURE AND BENEFITS OF TIME DOMAIN TORSIONAL VIBRATION ANALYSIS

John Srodawa[1]

[1]DEVCOM, Ground Vehicle Systems Center, Warren, MI

ABSTRACT

An Integrated Starter Generator (ISG) was integrated between an opposed piston two stroke engine and a 32 speed binary shift transmission for use in the Advanced Powertrain Demonstrator (APD). The initial design of the ISG integration and accompanying frequency domain torsional vibration analysis was performed considering driveline characteristics within the normal operating speed range of the engine. After a short period of time, the ISG suffered a catastrophic failure. The root cause of this failure is analyzed with special attention to the torsional behavior of the system. Multiple methods are employed to assess the torsional behavior including time domain torsional analysis. The time domain torsional analysis revealed that a significant number of torsional vibration cycles were occurring outside of the normal operating range of the engine as the engine accelerated from engine cranking speed to engine idle speed. The cycle accumulation during these short excursions through resonance ultimately lead to ISG rotor failure. The methods and lessons learned are presented.

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1. INTRODUCTION

The Advanced Powertrain Demonstrator (APD) uses a four cylinder opposed piston two stroke engine (OP2S) [1]. The engine contains two crankshafts which are coupled together through a series of gears. This geartrain also introduces a gear ratio between the engine crankshafts and the engine output shaft such that the output shaft makes 41 turns for every 37 turns of the engine crankshafts. The engine output shaft is driven by the center gear in this gear train. This engine is coupled to a 32 speed binary shift transmission without a torque converter. An Integrated Starter Generator (ISG) is integrated between the engine and transmission with a rotor that rotates at engine speed. The output shaft of the engine is an external spline with limited bearing support such that radial loads should be kept to a minimum. The transmission input shaft is supported by tapered roller bearings providing substantial support for cantilever loads. The ISG rotor is directly coupled to the transmission input shaft via a flywheel. A torsional coupling is bolted to this same flywheel. The torsional coupling provides torsional compliance, torsional damping, and radial compliance up to 0.5mm. A Torsional Vibration Analysis (TVA) of the engine including the engine gear train, torsional coupling, ISG rotor, and transmission input shaft inertia was conducted. The primary focus of the TVA was to determine the desired spring rate and damping characteristics of the torsional coupling to ensure that the engine gear tooth loads were kept within acceptable design limits.

2. FAILURE DESCRIPTION

The ISG suffered a catastrophic failure while the APD was being tested in a chassis dynamometer. After disassembly, it was found that the rotor had suffered mechanical failure consistent with excessive shear loads in torsion. The rotor consists of a stack of laminated steel sheets held together with a series of small diameter through bolts located near the periphery of the rotor. Several of these through bolts had loosened with some having the nuts come free and the bolt heads protruding from the rotor. These loose nuts and protruding bolt heads interfered with the stationary stator components causing damage to the coolant ports and windings of the stator. The ISG ultimately flooded with coolant while also experiencing resolver failure and winding shorts. This resulted in the ISG operating at excessive temperatures and losing starter and generator functionality. The failure of the bolted joints was consistent with excessive shear loading leading to slippage within the bolted joint and subsequent deformation of the bolts leading to a loss of bolt preload.

3. INITIAL PRE-FAILURE TVA

The initial torsional vibration analysis (TVA) was performed in the frequency domain using techniques similar to those first introduced by F.P. Porter [2]. This technique breaks the oscillatory engine forcing function up into a Fourier series of sine and cosine components each at an increased order (multiple) of the fundamental frequency. Essentially, the complex waveform of the engine torque as a function of crank angle is broken up into a number of discrete sine and cosine waves, each having a fixed amplitude and frequency for any given engine speed. The torsional system is then analyzed under steady state conditions for each one of these discrete sine or cosine waves of fixed frequency and amplitude over the range of engine operating speeds. The benefit of this frequency domain analysis is that the torsional system response is reduced to an algebraic form eliminating the need for intensive calculation of differential equations as a function of time. This analysis technique was essential prior to the advent of computers and, even today, significantly reduces computation time. The consequence of using a frequency domain TVA is that the actual oscillatory waveform as a function of time is lost. The frequency domain results provide one with the amplitude of vibration as a function of engine speed, along with the thermal dissipation of the torsional damper, but it is difficult to understand the number of oscillations experienced during any given engine speed transient. Long standing design convention states that the primary natural frequency of the engine-transmission system be set in the speed range between engine cranking and engine idle speed. This allows the engine to experience extended cranking events without risk of resonance and allows the engine to operate at idle speeds and above without risk of resonance excitation by the fundamental frequency or higher orders of the fundamental. It is understood that the engine will pass through resonance on every engine start, but common convention implies that the engine accelerates very quickly from cranking speed to idle speed spending very little time in resonance. Since the time spent passing through resonance is so short, it is assumed that the cycle count is also limited, eliminating the need for fatigue analysis in resonance. Based on this convention, the initial torsional vibration analysis focused on the expected engine operating speed range between engine idle speed and engine governed speed including expected torsional vibration amplitude, torsional torque amplitude, maximum engine gear train tooth loads, and torsional coupling thermal dissipation. The natural frequency was set at ~26Hz corresponding to an engine output shaft speed of 390rpm which lies safely between the maximum engine cranking speed of 250rpm and the engine idle speed of 1000rpm. There was nothing in the results that would indicate that there were any design issues.

4. POST-FAILURE TVA

The initial investigation of the failure quickly focused on the torsional behavior of the powertrain, especially during engine cranking and engine shutdown. The engine Electronic Control Unit (ECU) determines engine crankshaft speed using a 60-tooth tone wheel located on the engine crankshaft. The test cell data acquisition system was recording the ECU reported engine crankshaft speed at 50Hz which is just barely 2x the natural frequency. This results in some aliasing of the speed signal as a function of time and results in a low fidelity engine speed trace. However, some interesting data was still discovered. Figure 1 shows the recorded engine speed trace during the engine start. One can see that the powertrain passes into full resonance between 400 crank rpm and 600 crank rpm which corresponds to 443 rpm to 665 rpm at the ISG rotor due to the engine gear ratio. One can also see that the amplitude of the engine crankshaft oscillation is 200rpm peak-to-peak, however, due to aliasing it is not clear that this is the true amplitude of the oscillation. The data also shows that there are about six full oscillation cycles experienced while passing through resonance which is greater than expected prior to the failure.

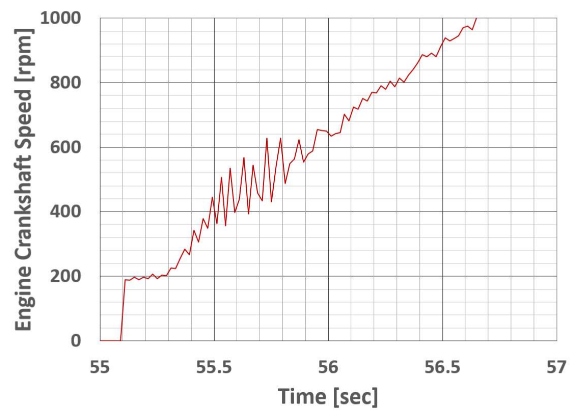

Figure 1: Engine crankshaft speed during an engine start recorded at 50Hz

The engine shutdown data was also recorded in the same way and is portrayed in Figure 2. As the engine spins down after the fuel injectors are turned off, the powertrain enters full resonance at crankshaft speeds between 500rpm and 390rpm. The amplitude of the oscillation is a bit lower at approximately 140rpm peak-to-peak, however the accuracy of this estimate is subject to signal aliasing. It should also be noted that the powertrain system experienced about seven full oscillation cycles while passing through resonance during this engine shutdown.

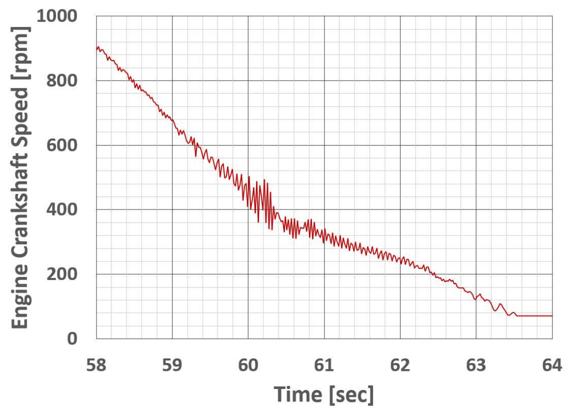

Figure 2: Engine crankshaft speed during engine shutdown recorded at 50Hz

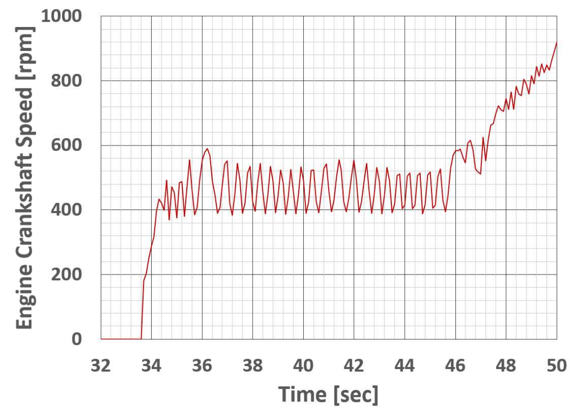

Figure 3: Engine crankshaft speed while starting with an air bubble in the fuel system

There was also one event where the engine fuel injection system experienced a delay in building fuel pressure in one rail due to air in the system. The engine has two fuel injectors per cylinder with one of these injectors being supplied fuel from the upper fuel rail and the other injector being supplied fuel by the lower fuel rail. Since one of the rails experienced an air pocket, the engine started but then struggled to achieve idle speed for about 12 seconds until both fuel rails were able to build pressure. Figure 3 shows the engine speed during this event. The data shows that the engine operated in resonance for the extent of this 12 second event accumulating 25 complete cycles.

At this point it became apparent that the common convention that the engine passes through resonance quickly enough not to cause appreciable damage was false. This low fidelity engine speed data shows that the powertrain system accumulates approximately six cycles on every engine start and another six cycles for every engine shutdown. This is a non-trivial number of cycles when considering fatigue failure. The focus then shifted towards creating a more accurate time domain model of the powertrain system to assess the actual environment that the ISG is exposed to.

4.1. Time Domain TVA Model

The powertrain architecture is depicted in Figure 4. In this figure one can see the two crankshafts, the eight pistons, the four cylinders, and the geartrain that couples the two crankshafts together while driving the engine output shaft. The torsional coupling is located between the output shaft of the engine and the combined ISG rotor inertia and transmission input inertia. Also note that there are two crankshaft dampers on the free ends of the crankshafts.

The primary concern of the analysis is the behavior of the torsional coupling and the operating environment of the ISG rotor. The engine manufacturer had already assessed the torsional behavior of the crankshafts and the crankshaft dampers. It was noted during the initial TVA that the natural frequencies of components within the engine assembly are significantly higher than the natural frequency of the torsional coupling. It was also noted that since the torsional coupling is the lowest natural frequency of the system, that changes in crankshaft inertia had little to no effect on the torsional behavior of the ISG or transmission. For these reasons, the internal aspects of the engine were ignored in

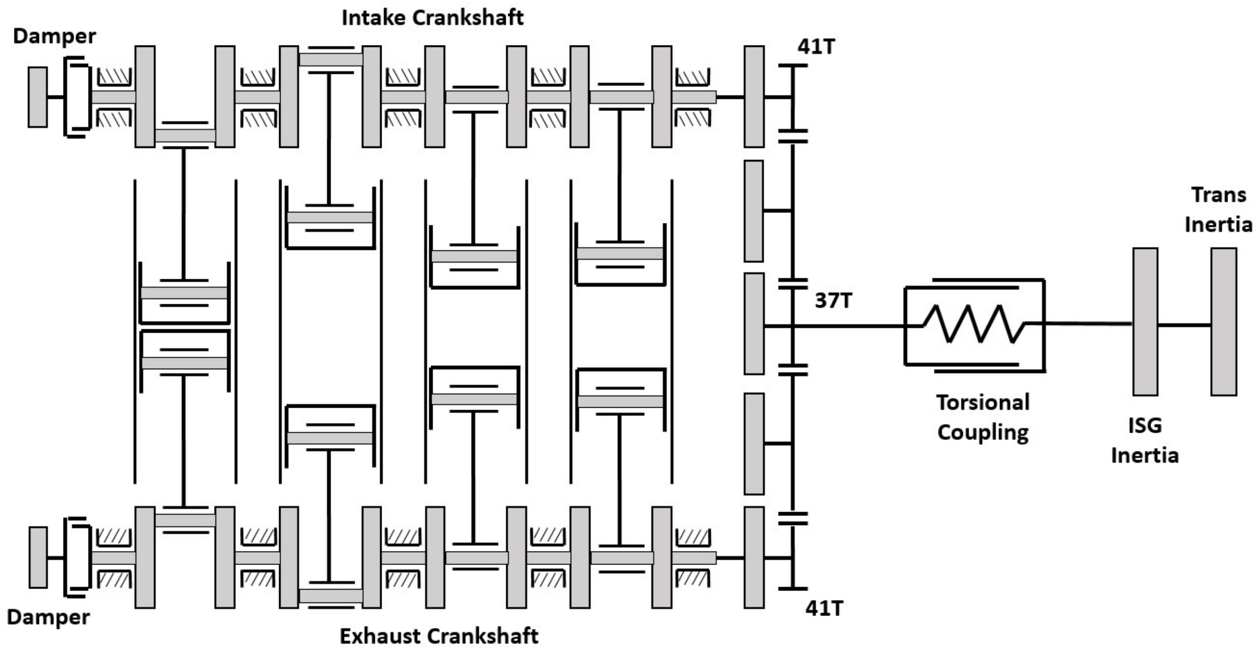

Figure 4: Schematic of actual engine components

the time domain model making the model significantly less complex. The model reduces to a simple two mass problem with an engine forcing function applied. To simplify the analysis further, the engine forcing function is modeled assuming that the engine has a single crankshaft, four cylinders, four pistons, and a conventional cylinder head. The cylinder head valve events are timed to Mimic the OP2S port opening and closing events. The resulting cylinder trace is multiplied by a factor of two to account for the pressure induced forces on the omitted crankshaft.

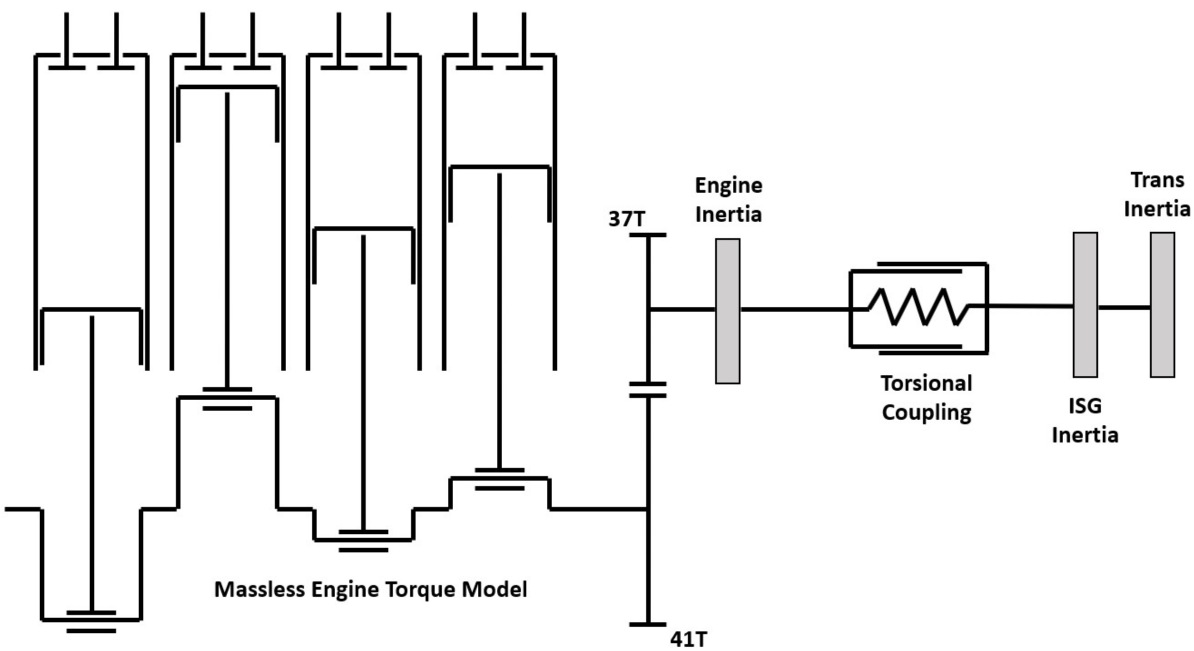

Figure 5: Simplified two mass torsional model with cylinder pressure forcing function

The engine forcing function was generated using a simple crank angle-based piston cylinder model based on the polytropic process. Since the model predicts the starting of the engine, in-cylinder conditions were set to standard atmospheric pressure at the start of the model execution. This is true even if the piston is in mid stroke, as cylinder pressure is likely to equalize with atmospheric conditions over a reasonable amount of time due to piston ring leakage. The cylinder pressure is also set to atmospheric conditions whenever the cylinder ports are open. It is assumed that during cranking and during the initial acceleration to idle speed, the intake and exhaust manifold pressures are still at atmospheric conditions even though the superchargers are engaged. It is assumed that at this point the superchargers are just facilitating the movement of fresh charge through the cylinders rather than building any significant manifold pressure. The piston-cylinder-crank model parameters used in this analysis are presented in Table 1.

Table 1: In-cylinder pressure model parameters

Bore	130	mm
Stroke	135	mm
Conrod Length	226	mm
Clearance Volume	0.18	liters
Inlet Port Closure	115.4	deg BTDC
Exhaust Port Opening	115.4	deg ATDC
Polytropic Exponent	1.3	-
Peak Cylinder Pressure	38	bar

The in-cylinder combustion pressures were simulated with a simple step change in cylinder pressure at exactly Top Dead Center (TDC) assuming that injection timing and duration of injection have been properly established to place peak pressure rise at or near TDC. The peak cylinder pressure at TDC is adjustable to influence the overall mean torque applied to the engine mass element. The engine speed traces captured from the ECU, ignoring the resonance region, show a fairly linear acceleration from 50 rpm to 900 rpm. The amount of time required to accelerate from 50 rpm to 900 rpm was measured from the traces to determine an average system acceleration rate. This acceleration was multiplied by the total system inertia to develop an average torque during the acceleration to idle. The peak cylinder pressure was then adjusted to obtain this same average torque on the engine mass element in the model. The in-cylinder pressure trace for a single cylinder is shown in Figure 6. The fluctuating torque of the four individual cylinders is shown in Figure 7. The resulting total torque waveform after accounting for an engine frictional torque of 220Nm is shown in Figure 8. This engine friction value was measured using a torque wrench during powerpack assembly and matches engine friction data at idle speed from the engine supplier.

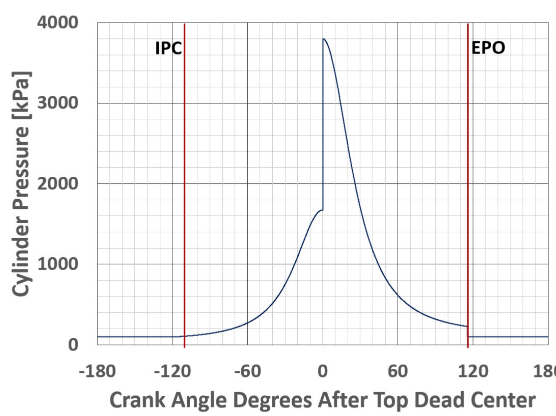

Figure 6: Estimated in-cylinder pressure trace

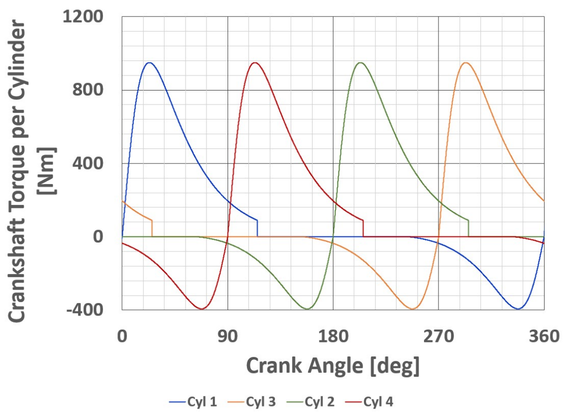

Figure 7: Crankshaft torque trace for each cylinder

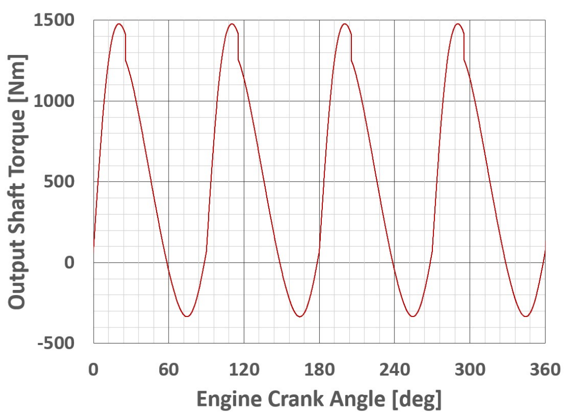

Figure 8: Engine output shaft torque trace

The model was constructed in Excel using a simple Euler time step approach. A 20µs time step was used which is equivalent to a 50kHz sample rate. The initial conditions assume that the engine and ISG are at complete rest and that the engine is positioned such that the #1 cylinder is 90 degrees BTDC. The engine inertia was set to 1.56 kg-m^2 based on engine manufacturer data. The combined ISG rotor and transmission input inertia was set to 6.9 kg-m^2 based on supplier data and 3D CAD model data of additional integration components. The torsional coupling spring rate was set to 51700 Nm/rad per the Torsional coupling drawing. The ISG starter cutout speed was set to 200 rpm and the fuel injection cut-in speed was set to 50 rpm. The appropriate damping constant was not available from supplier data, so the damping constant was varied until the amplitude of the crankshaft speed oscillation matched the maximum amplitude of 200 rpm peak-to-peak seen in the ECU traces. The damper was modeled as a pure viscous damper with a damping constant of 50 N-m-s. The resulting Excel model contained 73 columns and 117,144 rows. The Excel spreadsheet requires about 35 seconds to update after an input parameter change and requires about 48 seconds to save on a medium performance laptop. The model construction effort was similar to or a bit less time consuming than similar MATLAB Simulink based models. Model execution time is shorter than similar Simulink models and there were no issues with algebraic loops which can be very challenging for time domain torsional vibration models in Simulink.

4.2. Time Domain Model Results

The resulting engine starting speed trace is shown in Figure 9 below. This trace has been developed for the crankshaft speed so that it can be directly compared to the ECU speed traces presented earlier.

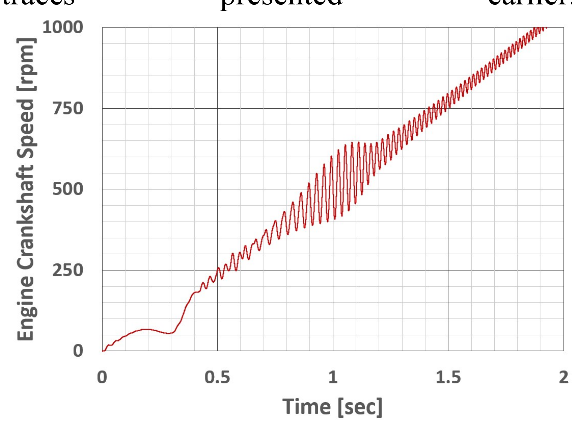

Figure 9: Time domain model engine speed trace results

The speed trace presented in Figure 9 has significant high frequency content and has a very different appearance than the ECU speed traces. Oscillation begins almost immediately upon engine firing and persists all the way to engine idle. The amplitude of vibration reaches a peak at an engine crankshaft speed of approximately 520 rpm, which corresponds well to the ECU traces. To better compare the predicted speed trace against the measured ECU speed traces, the signal processing related to a 60 tooth speed pick up wheel and the associated 50 Hz ECU recording rate was simulated. The simulated ECU speed trace is shown in Figure 10. The signal processing involved with the ECU speed pickup removes a significant amount of the high frequency content and results in a speed trace that is nearly identical to that presented in Figure 1. The step spike in speed between 0 and 0.1 seconds is due to the signal processing equations not developing a full synchronization with the 60 tooth wheel at such low crankshaft speeds.

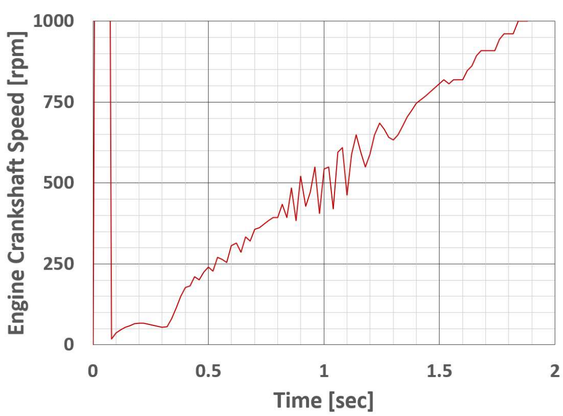

Figure 10: Time domain model results with simulated 50Hz data recording of ECU tone wheel

Referring back again to Figure 9 one will note that the speed trace shows approximately 10 cycles of oscillation while passing through resonance, which is more than that indicated by the speed trace recorded from the ECU. It is clear that the real hardware is more torsionally active than the ECU speed trace would suggest.

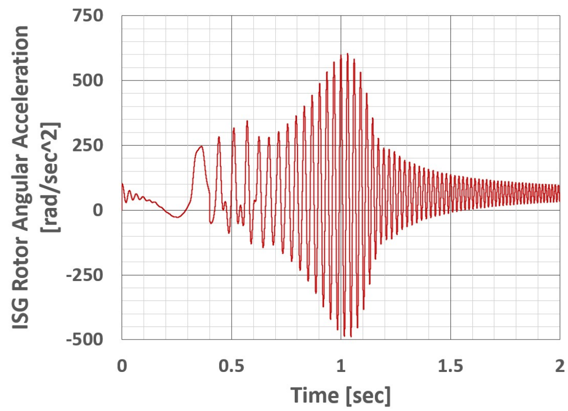

Figure 11: Time domain model predictions of ISG rotor angular acceleration

Figure 11 shows the angular acceleration experienced by the ISG rotor as the rotor passes through resonance during a normal engine start. This is the most important data to the ISG rotor designer. This angular acceleration is the source of the torsional shearing forces between different elements of the ISG rotor construction. It is this angular acceleration that ultimately stresses the bolted joints of the system and generates the stresses that can lead to a fatigue failure. In assessing the design for fatigue, one should note that there are a large number of cycles of varying intensity experienced during a normal engine start. Careful examination of Figure 11 will reveal that there are over 20 cycles at an acceleration that exceeds 250 rad/sec^2 which exceeds the angular acceleration at idle by a factor of 2.5. As ISGs are normally associated with anti-idle technology and reduced engine idle times during silent watch, powerpacks with ISGs will experience more engine starts than traditional powerpacks without ISGs. Depending upon vehicle usage, one can expect between 10^5 and 10^6 oscillations per decade. It is thus evident that the common convention that the engine accelerates quickly enough through resonance that fatigue can be ignored is a risky assumption. A proper fatigue analysis would require one to create a histogram of the time domain data to capture the number of cycles as a function of cycle amplitude during the engine start and engine shutdown and then utilize Miner's rule to determine the overall fatigue life of any fatigue sensitive components. Another option, if optimizing for weight and volume is not critical, is to design for infinite fatigue life at the worst case angular acceleration experienced at peak resonance. However, this option is only valid for materials that exhibit a fatigue strength endurance limit. Other materials will always have a finite life requiring the full Miner's rule based fatigue analysis.

It should be noted that bolted joint failure is not contingent upon a large number of accumulated cycles as is fatigue failure. Rather, a bolted joint can fail after exposure to a relatively small number of alternating force conditions that are strong enough to produce relative motion in the bolted joint. To ensure the safety of bolted joints used in the ISG rotor construction, it is important to verify that the bolted joint has sufficient preload and friction to prevent bolted joint slippage at the worst case expected angular acceleration. In this case, the time domain TVA is predicting a peak angular acceleration of 600 rad/sec^2 as depicted in Figure 11. However, caution should be used as the predicted amplitudes in resonance are highly dependent upon the damping values used in the model. Since the damping values used in this model were tuned to match the ECU recorded speed traces, there was an inherent level of uncertainty around the damping assumptions.

The same time domain model can be used to evaluate the torque amplitude of the torsional coupling and the deflection of the torsional coupling to ensure that the torsional coupling is being kept within safe operational limits in accordance with the torsional coupling specifications.

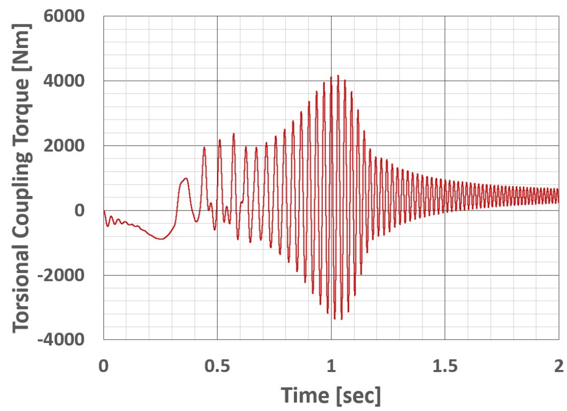

Figure 12: Time domain model prediction of Torsional coupling torque

Figure 12 presents the torque across the torsional coupling in the time domain. Figure 13 presents the angular deflection of the torsional coupling during a normal engine start. Both of these parameters were well within the torsional coupling limits and were not associated with the failure.

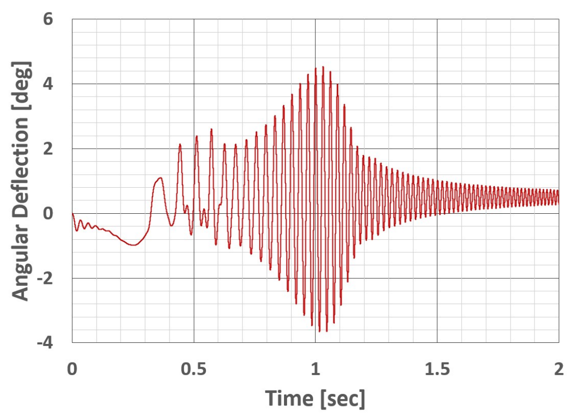

Figure 13: Time domain model prediction of Torsional coupling angular deflection

4.3. Impact on ISG Design

Based on the time domain TVA results, it was possible to identify that the peak angular acceleration imposed on the ISG rotor was 600 rad/sec^2. This value is dependent upon the damping values used in the model which had a level of uncertainty associated with them. For this reason, the ISG supplier was given an angular acceleration requirement of 1500 rad/sec^2 at a natural frequency of 30Hz. The ISG supplier redesigned the rotor to provide a more robust bolted joint along with adhesives to eliminate any chance of bolted joint slippage and relative motion between the laminated sheets of the rotor. The new rotor design was confirmed using a Finite Element Analysis using the 1500 rad/sec^2 angular acceleration assumption. The FEA showed that the new rotor design was safe and new ISG rotors were procured per the new design. These new rotors were installed into the Advanced Powertrain Demonstrator powerpacks and were subjected to over 200 hours of in vehicle testing without any further issues.

5. CONCLUSIONS

Following the catastrophic failure of an ISG rotor during powerpack dyno testing, a detailed review of the torsional vibration analyses was conducted. Previous analyses had ignored torsional behavior below engine idle speed. This decision was based on the long-standing convention that the natural frequency of the driveline should be placed between the engine cranking speed and the engine idle speed. This convention also implies that the driveline spends so little time between engine cranking and engine idle speeds that operation in resonance can be ignored.

Traditional methods of evaluating driveline torsional vibrations are based on frequency domain models. These modeling techniques were developed before the advent of computers to reduce the computational burden of the torsional vibration analysis. These frequency domain models have continued to be used throughout the last century. With the advent of increased computing power, it is now possible to make relatively simple time domain torsional vibration models on a desktop computer or laptop using MATLAB, Simulink, or even Excel.

A simple time domain model was developed for the ISG failure analysis. This time domain model revealed that peak angular acceleration in the driveline as the engine passed through resonance exceeded the bolted joint capacity of the ISG rotor design. The time domain model also revealed that the driveline experiences a significant number of vibration cycles while passing through resonance. Based on these findings, it was determined that operation in resonance can no longer be ignored, especially as ISGs are used in powerpacks to facilitate enhanced silent watch capabilities combined with anti-idle technology. Driveline bolted joints must be designed not to slip when exposed to the worst case angular acceleration at resonance. Fatigue sensitive components must be designed to withstand the number of cycles and the cycle amplitudes determined from a time domain torsional vibration analysis.

6. REFERENCES

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