WHITE PAPERS

UNDERSTANDING THE EFFECTS OF ENGINE DOWNSPEEDING ON DRIVETRAIN COMPONENTS

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INTRODUCTION

The drivetrain configuration consisting of engine, transmission, drive shaft and rear axle for Class 8 Linehaul trucks has been evolving over the past few years. This change is being brought about in an effort to lower emissions and increase fuel efficiency. The trend is to run engines slower while at cruising speed to achieve better fuel efficiency and lower emissions. To accomplish this, a selection of fast rear axle ratios combined with direct drive transmissions is gaining popularity. While this approach has some merit, the resulting engine torque that the drivetrain components (transmission, drive shafts and rear axles) have to transmit is much greater when coupled with faster numeric rear axle ratios. Greater torque has a direct impact on the structural performance of the drivetrain components.

At first there was no real concern for the drivetrain since the Gross Axle Weight (GAW) and Gross Combination Weight (GCW) ratings were not changing. However, for rear drive axle ratios less than 3.0, the amount of peak torque the drivetrain experiences during low speed maneuvers (such as start up, trailer positioning and trailer tug) and at road speed is much greater. The increase in peak torque can result in fractured drive shaft universal joints, transmissions, rear axle components and inter-axle drive shafts if not considered in the initial design.

Although there was no discernible connection between any reported drive shaft fractures and engine downspeeding, Meritor initiated a task force of engineers and specialists to investigate drive shaft fracture issues. This White Paper presents the results of Meritor's investigation to help the commercial truck industry understand the effects of engine downspeeding on drivetrain components; focusing on engine downspeeding, rear drive axle and transmission ratios, component fractures, dynamic simulation modeling, vehicle test data and dynamic simulation to vehicle test correlation.

ENGINE DOWNSPEEDING

Engine downspeeding has evolved over recent years. Important changes and improvements were introduced to diesel engines to be EPA 2010 emission regulation compliant. The evolution came about primarily by architectural engine changes; and as a result, the EPA 2010 engines are more capable and quicker to respond than previous engine models. For a detailed discussion, consult SAE 2013: L. RAY BUCKENDALE LECTURE 2013-01-2421 *"Systematic Development of Highly Efficient and Clean Engines to Meet Future Commercial Vehicle Greenhouse Gas Regulations"* by Donald W. Stanton – Cummins Inc. The lecture illustrates engine downspeeding and fuel efficiency improvement, and the following underlined topics from the lecture have the greatest impact on engine ramp-up rate, torque and inertia effects.

- Air-to-Air Charge Cooling
- <u>Electronic Control Systems</u>
- <u>Combustion Systems</u>
- <u>Variable Geometry Turbocharger</u>
- Cooled EGR
- <u>High Pressure Common Rail Fuel System</u>
- Diesel Particulate Filter (DPF)
- Selective Catalytic Reduction (SCR)

**SAE 2013/ L. Ray Buckendale Lecture 2013-01-2421: "Systematic Development of Highly Efficient and Clean Engines to Meet Future Commercial Vehicle Greenhouse Gas Regulations" by Donald W. Stanton, Cummins Inc. The lecture Illustrates engine downspeeding and fuel efficiency improvement.*

Peak Cylinder Pressure and Power Density had gradually increased over time. When EPA 2010 engines were introduced, Peak Cylinder Pressure and Power Density rose sharply. This steep increase has a direct impact on the drivetrain components being used in trucks today. See Figure 1.

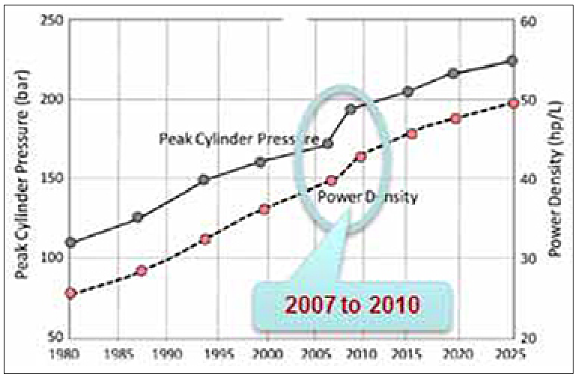

Figure 1 – Peak Cylinder Pressure and Power Density Increase
SAE 2013/L. Ray Buckendale Lecture: 2013-01-2421

The engine architectural changes previously stated significantly altered the engine torque curve. The rate by which an EPA 2010 engine reaches full-rated engine torque after the clutch is engaged is significantly faster. See Figure 2.

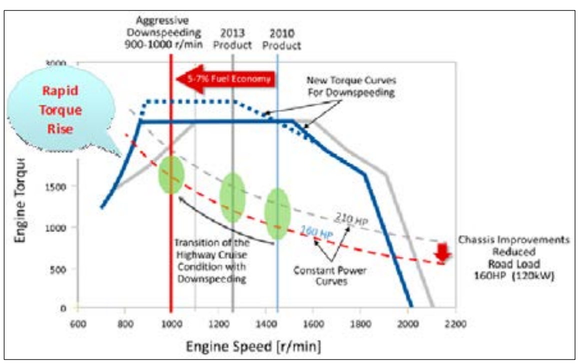

Figure 2 – Engine Downspeeding and Effects on Fuel Efficiency Improvements
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The graph depicts an aggressive downspeeding trend that lowers engine speed to 900-1000 RPM at highway cruising speed. The EPA 2010 and newer engines have a very rapid torque rise from idle until reaching full-rated engine torque. The solid gray curve represents an engine torque curve for a pre-EPA 2010 engine. You will notice that the engine torque increases gradually from about 800 RPM until full engine torque at about 1100 RPM. The solid and dotted blue curve represents an EPA 2010 and newer engine torque curve that shows a very steep and rapid torque increase from about 700 RPM until full engine torque at about 900 RPM. The rate of torque increase happens in less than one-half (0.5) second and is very different to a truck driver who has not been previously exposed to the newer engines.

The trend to continue downspeeding for fuel efficiency gains has limitations for drivetrain components. Trucks equipped with manual transmissions become more difficult to drive, Automated Manual Transmissions (AMT) becomes more limited, dual clutches become more necessary for operation, and rear drive axle and drive shafts components become more stressed. See Figure 3.

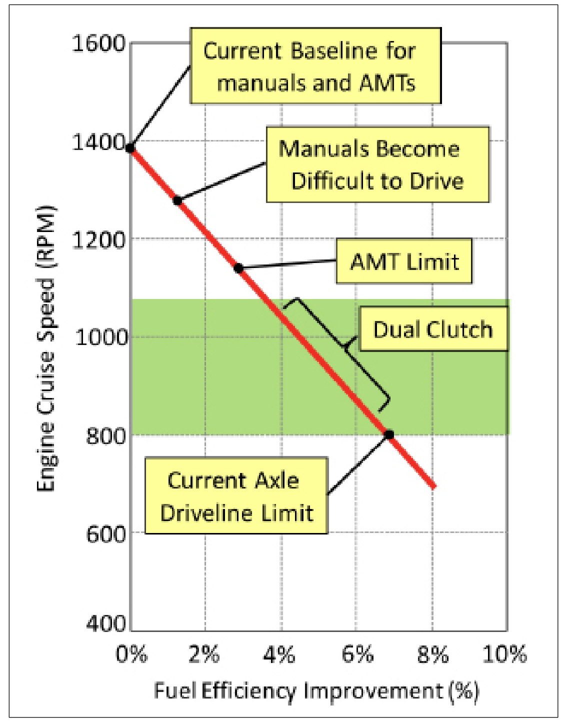


Figure 3 – Drivetrain Limitations Over the Engine Speed Range for Linehaul Truck at Cruising Speed
SAE 2013/L. Ray Buckendale Lecture: 2013-01-2421

REAR AXLE RATIO AND TRANSMISSIONS

Downspeeding engines to save fuel and comply with regulatory greenhouse gas (GHG) emissions standards is rapidly being implemented by several fleets. Engine downspeeding is made possible by selecting a faster numeric rear drive axle ratio - less than 2.64 - and coupling it to direct drive transmission. This combination of fast axle ratio and direct drive transmission gives the operator a truck that has excellent startability (18-20%) and slower engine rpm at cruising speed for fuel economy.

The drivetrain torque required to maintain an acceptable startability increases substantially as the rear drive axle ratio is reduced. See Figure 4. The chart shows the engine and drivetrain torque required for different rear axle ratios between 3.55 to 2.47 and various startability values from 15 to 20 percent. The Static Loaded Radius (SLR) of the tires is 18.9 inches, and the GCW is 80,000 pounds. The engine torque required to maintain startability going from a 3.55 to a 2.47 rear axle ratio is 23 percent higher. Correspondingly, the drivetrain torque is 44 percent higher, and drivetrain components are now exposed to a much greater load and stress. This happens when a truck accelerates from a stop, positions a trailer at the terminal, docking a trailer, and at on-highway steady-state cruise speeds. The consequence of these occurrences can be more frequent fractured components or much lower fatigue life of the drivetrain components.

	Startability (%):	15	16	17	18	19	20
	Tire SLR (inches):	18.9					
	GCW (lbs):	80,000					
	Axle Ratio	Required Torque (lb-ft)					
Engine Torque	3.55	886	921	957	992	1,028	1,063
	3.36	936	973	1,011	1,048	1,086	1,123
	3.08	1,021	1,062	1,103	1,143	1,184	1,225
	2.85	946	984	1,022	1,060	1,097	1,135
	2.79	967	1,005	1,044	1,082	1,121	1,160
	2.64	1,021	1,062	1,103	1,144	1,185	1,225
	2.47	1,092	1,135	1,179	1,222	1,266	1,310
Driveline Torque	3.55	11,244	11,693	12,142	12,591	13,040	13,489
	3.36	11,879	12,353	12,827	13,302	13,776	14,251
	3.08	12,957	13,474	13,992	14,509	15,027	15,544
	2.85	14,004	14,563	15,123	15,682	16,241	16,800
	2.79	14,305	14,876	15,447	16,019	16,590	17,161
	2.64	15,116	15,720	16,324	16,927	17,531	18,135
	2.47	16,155	16,800	17,446	18,091	18,736	19,382

Axle ratios faster than 3.0 use a direct drive transmission with a 14.8:1 first gear.
Axle ratios slower than 3.0 use an overdrive transmission with a 12.69:1 first gear.

Figure 4 – Engine and Drivetrain Torque Required for Startability

COMPONENT FRACTURES

Increased engine torque and driver influence can adversely affect the performance and reliability of the drivetrain system. A truck driver who suddenly accelerates at start up or during a trailer maneuvering event at a terminal, such as a docking event or during winter snow and ice conditions, can inadvertently apply too much torque, due to the contribution of engine, clutch and transmission inertia. As a result, a fracture can occur to a drivetrain component; for example, the universal joints and drive shafts, rear axle assemblies and transmissions. See Figure 5. Fractures can often be very expensive to repair and can take a truck out of service for several days.

Universal Joint | **Transmission Gearing** | **Rear Axle Pinion Stem**

Figure 5 – Fractured Drivetrain Components

When these types of fractures happen, the fractured surfaces exhibit evidence of a sudden overstress condition. Most often this is a single event. This is different from a fatigue fracture that is depicted by a series of witness marks that progress through the part, until the point that the ultimate strength of the part is exceeded and full fracture occurs. These progressing witness marks, called beach marks, can been found at the fracture location. See Figure 6.

Sudden Overstress | **Fracture Fatigue Fracture**

Figure 6 – Examples of Overstress Fracture versus Fatigue Fracture

DYNAMIC OVERSTRESS

A great deal of effort goes into determining the torque that a drivetrain system will encounter to safeguard against overstress. The formulas and inputs to calculate maximum component torque depend on the drivetrain component under consideration. For example, the maximum axle input torque will include the gross engine torque, transmission low gear ratio (forward or reverse) and an efficiency value for both the engine and transmission.

The values obtained by these calculations are compared against the wheel slip torque. If an application has very high multiplied engine torque, the assumption is that the wheels will slip before the drivetrain components are overstressed.

Once the drivetrain torque values are determined, the drivetrain components are selected in part by the maximum rated torque rated by the component manufacturer.

Historically, this method of calculating drivetrain torque and selecting components was successful. With the introduction of the EPA 2010 engines, however, some truck operators and fleets started experiencing a greater occurrence of component fractures.

Data collected from trucks equipped with EPA 2010 engines, direct drive transmissions and axle ratios less than 2.64 exhibited instances where the measured drive shaft torque exceeded 20 percent greater torque than the calculated torque using traditional formulas. In some cases, the measured torque values far exceeded the rated maximum torque values of certain drivetrain components.

A test truck equipped with an EPA 2010 engine rated at 1550 lb-ft, 410 HP, a direct drive transmission second gear ratio at 10.95 and a rear axle ratio of 2.47 has a calculated maximum torque of 13,700 lb-ft. However, during a second gear aggressive start up, the measured drive shaft torque, using an instrumented drive shaft, was 21,600 lb-ft. The measured drive shaft torque value was 58 percent greater than the calculated torque. As a result, the test truck fractured a drive shaft universal joint cross. See Figure 7.

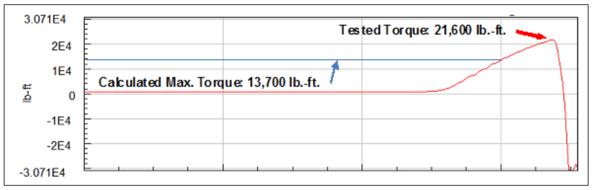


Figure 7 – Calculated Maximum Torque versus Torque Measured in a Test Truck

DYNAMIC SIMULATION MODEL

A better understanding between the calculated maximum drivetrain torque values and the measured test values is required. To accomplish this, a dynamic simulation model was developed by Meritor that simulates the dynamic transient inertial behavior of the drivetrain system. As demonstrated by test, the drive shaft torque can greatly exceed the calculated steady-state torque values. The simulation model is a tool to predict and help better understand the dynamic load behavior in the drivetrain system.

The model combines the engine, transmission, drive shaft and rear axle into a simple spring, mass, damper system. The model takes into account the transient inertial effects of the system. See Figure 8.

Figure 8 – Original and Simplified Drivetrain Dynamic Model

The model contains some generic assumptions for engine inertia and system dampening. However, data collected during the truck test verified the assumptions. The model provides the ability to examine the influence of parameters, such as engine ramp-up time and engine speed, to overall system level torque response.

Varying the engine ramp-up time to full torque with the model for an aggressive launch event has a remarkable impact on the peak torque to which the drivetrain system is exposed. For example: an engine ramp-up time of 0.5 seconds to full rated torque can yield a peak torque of 24,500 lb-ft. Slowing the ramp-up time to 2.0 seconds yields a peak torque of 20,100 lb-ft or an 18 percent reduction in peak torque. This is in contrast to a truck that has a calculated steady-state torque of 18,500 lb-ft. See Figure 9.

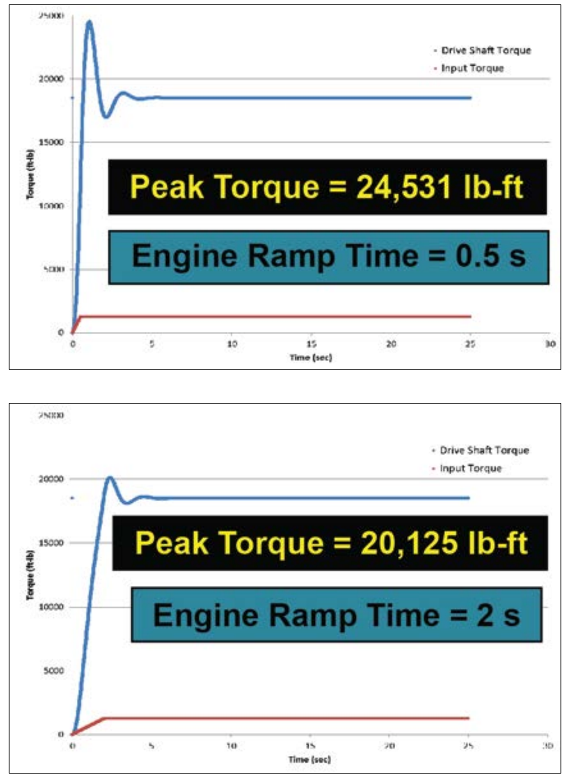


Figure 9 – Peak Torque Effect Due to Engine Ramp Up Time

Similarly, varying the engine speed for a drop-clutch event results in a significant difference in drivetrain peak torque. For example: an engine speed of 1,750 RPM at clutch engagement can yield a peak torque of 29,300 lb-ft. Slowing the engine speed down to 1,000 RPM yields a peak torque of 18,400 lb-ft or a 37 percent reduction in peak torque. See Figure 10.

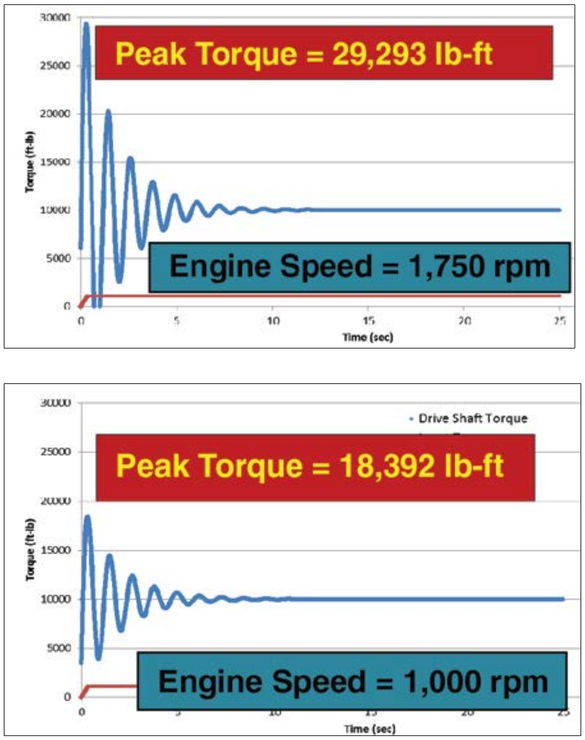


Figure 10 – Peak Torque Effect Due to Engine Speed

As rear axle ratios become faster, the peak transient torque increases at a much greater rate as the engine speed response time decreases. For rear axle ratios decreasing from 3.55 to 2.85, the percent of peak transient drivetrain torque increases at a fairly linear rate as engine ramp-up time decreases from 2 seconds to 0.5 seconds. For rear axle ratios decreasing from 2.79 to 2.19, the percent of peak transient drivetrain torque increases at a much greater rate as engine ramp-up time decreases from 2 seconds to 0.5 seconds. The percent increase of peak transient drivetrain torque affects transmissions, drive shafts and rear axle components. See Figure 11.

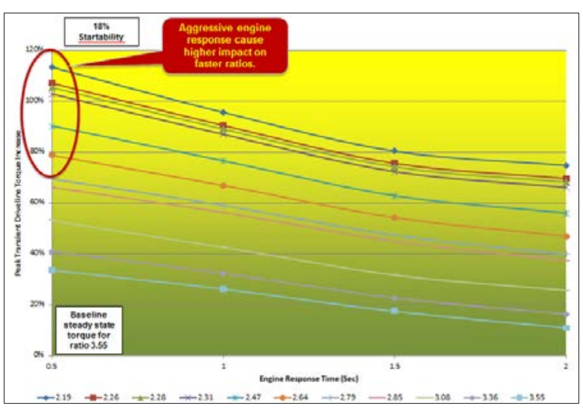


Figure 11 – Percent of Peak Transient Drivetrain Torque Increase versus Engine Ramp Up Speed for Faster Rear Axle Ratios

VEHICLE TEST

Truck tests were conducted using different chassis and engine combinations. Each truck was equipped with a rear axle ratio less than 3.0 and a manual direct drive transmission. The purpose of the test was to gain a better understanding of the conditions that lead to an overstressed component and evaluate engine control settings to prevent an overstress condition. The test procedure consisted of three block events.

Block 1: Torque Build-Up Event:
Test Scope: Evaluate if the drivetrain torque controls are efficient and working.
Test/maneuver description: With vehicle rolling slowly, driver to apply brake and acceleration until the torque reaches the torque limit settings.

Block 2: Aggressive Release of the Clutch Event:
Test scope: Evaluate optimal engine setting that will prevent drivetrain torque to exceed rated torque.
Test/maneuver description: With vehicle standing, apply acceleration and release the clutch abruptly.
Execute 1st gear maneuvers. Monitor drive shaft torque and if it exceeds drive shaft rated torque, stop the test; revert to the setting before the drive shaft rated torque was exceeded.
Execute 2nd gear and reverse maneuvers to confirm torque is not exceeding drive shaft rated torque.

Block 3: Trailer Slide (trailer brakes on) Event:
Test scope: Evaluate optimal engine settings during trailer slide events (trailer brakes on).
Test/maneuver description: With vehicle controls set at engine speed and torque control resultant from Block 2, apply acceleration and release the clutch abruptly to stall the engine.

The events simulate maneuvers that were most often reported by drivers at the time a drive shaft universal joint cross fractured. The rated torque of the Meritor RPL25SD drive shaft is 18,500 lb-ft. Any test event that exceeded the drive shaft rated torque constitutes an unacceptable condition.

Data collected from the truck tests shows that events can occur that generate drivetrain torque values far greater than the calculated steady-state torque values. Each test event produced drivetrain torque values that exceeded the rated torque of the drivetrain components and overstress fractures occurred. These events are low speed maneuvers that are not uncommon at a terminal or loading dock.

Engine settings affecting torque control and engine speed were repeatedly modified for truck speeds below 5 KPH. The tests were duplicated and the impact of the changes on measured drivetrain torque were analyzed. Iterations were performed until settings that satisfactorily limited maximum drivetrain torque were identified.

The engine control strategies are different among engine manufacturers. Meritor's investigation is not intended to define the appropriate engine control parameter values. Rather, it is to demonstrate that by limiting parameters like low speed torque and engine speed, the overstress drivetrain conditions can be reduced or eliminated.

Subjective evaluation by drivers found no discernible difference for startability and drivability when peak torque mitigation engine control settings were applied.

DYNAMIC SIMULATION TO VEHICLE TEST CORRELATION

Data obtained during vehicle tests shows a strong correlation with the simplified dynamic simulation model. The dynamic simulation model for an aggressive launch event predicts that the drivetrain torque will increase rapidly from zero to 20,000 lb-ft and dampen to a steady-state torque of 18,500 lb-ft within 5 seconds. The comparable vehicle test shows a very similar result. In a similar fashion, the dynamic simulation model for a drop-clutch event predicts a torque spike followed by dampening within 5 seconds. The vehicle test shows the similar result. See Figures 12 and 13.

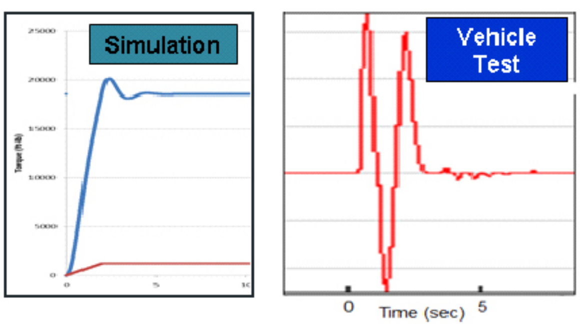


Figure 12 – Aggressive Launch Event Simulation Compared to Vehicle Test

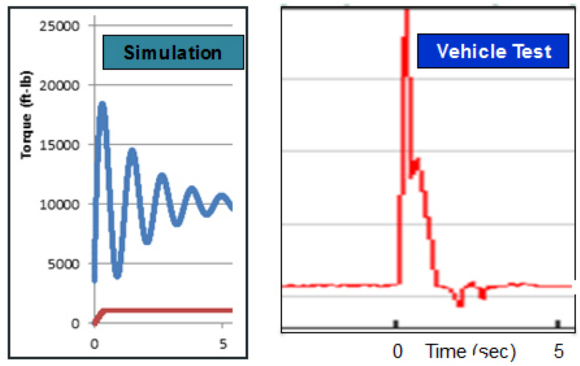


Figure 13 – Drop Clutch Event Simulation Compared to Vehicle Test

To further illustrate the dynamic simulation model correlation to vehicle test, actual test results of a truck with an instrumented drive shaft subjected to two drop-clutch events are overlaid against the simulation model. The dynamic simulation model predicts a very rapid torque increase on the drive shaft that exceeds 20,000 lb-ft of torque followed by a decaying oscillation. Data from second and third vehicle tests are overlaid upon the dynamic simulation model. Data from one test shows a very rapid torque increase exceeding 20,000 lb-ft followed by a dampening state. The data from the other test also shows a very rapid torque increase exceeding 20,000 lb-ft terminating in a drive shaft fracture. Subsequent model refinements have demonstrated improved correlation. It should be noted that the drive shaft that fractured had previously been subjected to a number of events that exceeded the rated torque capacity. See Figure 14.

The dynamic simulation model was also used to calculate the peak torque values across varying engine speeds in the range of 650 to 1000 RPM. The simulation was run for drop-clutch events in both first and second gear. The simulation results are overlaid against test data collected during vehicle tests. The dynamic simulation model and vehicle test show very strong correlation throughout the engine speed range. See Figure 15.

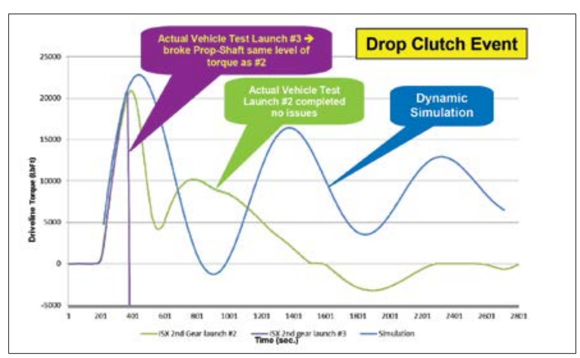
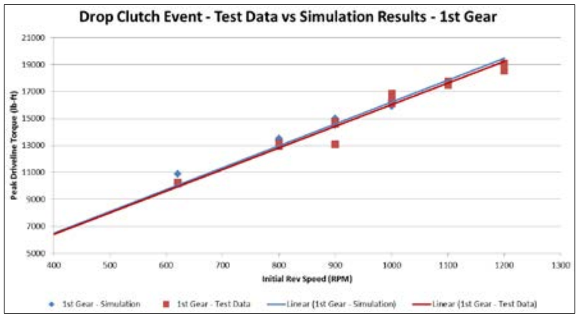


Figure 14 – Drop Clutch Data Compared to Dynamic Simulation



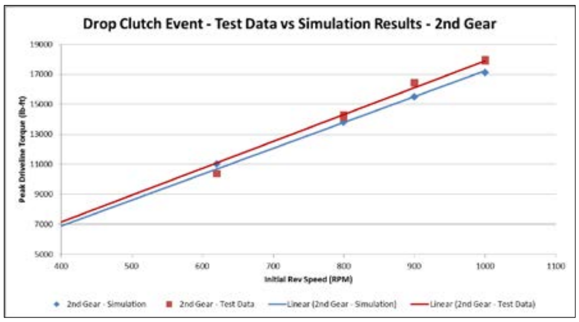


Figure 15 – Dynamic Simulation of Peak Torque at Varying Engine Speed Compared to Vehicle Test Data for Drop Clutch Event in 1st and 2nd Gear

SUMMARY

Engine downspeeding has been evolving in recent years since the introduction of the EPA 2010 engine and significantly alters the engine torque curve. The rate by which an EPA 2010 engine reaches full-rated torque is much quicker, which places a much greater load on the drivetrain system. The torque carried through the drivetrain system reaches and, in some cases, exceeds the structural integrity of the drivetrain system. As a result, drivetrain components are being subjected to an overstress condition that is manifesting into sudden component fracture. These fractures occur more often during vehicle start up and acceleration, positioning maneuvers and docking events at a terminal or during winter time ice and snow conditions. Historically, the method used to calculate steady-state drivetrain torque was satisfactory for sizing drivetrain components. Drivetrain data collected from trucks equipped with EPA 2010 engines having rear axle ratios less than 3.0 and direct drive transmissions exceeds 20 percent greater torque than traditional steady-state calculation methods.

A dynamic simulation model was developed to better understand the transient inertial effects of the drivetrain system. The simulation model is a tool used to predict the large differences between calculated steady-state torque and vehicle dynamic torque events.

Several truck tests were conducted. The purpose of the tests was to measure the amount of torque produced during various truck maneuvering events. Test results show there are events that are not uncommon that expose the drivetrain to torque values that exceeds the rated torque limits of the components. These events can overstress and fracture the drivetrain components.

Torque and system response data measured during truck tests correlated closely with values predicted using the dynamic simulation model. The close correlation between test data and predicted data validates the usefulness of the simulation model. The model can now be used to more accurately predict the behavior of a truck system for future downspeeding applications.

CONCLUSION

The trend to downspeed engines will continue as a way to improve fuel efficiency and lower emissions. The new engines will be combined with rear axles having ratios as low as 2.19 or less and direct drive transmissions. The amount of torque transmitted through these new drivetrain combinations is much greater than former combinations. As a result, the drivetrain components will be more frequently exposed to overstress conditions that can fracture a drivetrain component.

The dynamic behavior of a vehicle is dependent on the characteristics of the drivetrain components. The characteristics include mass/inertia of the system, system stiffness and system dampening. Transient torque behavior varies by vehicle configuration. Changes to any drivetrain component or subsystem can significantly impact the transient behavior of the drivetrain system.

Dynamic simulation modeling and vehicle tests confirm that the transient behavior of the drivetrain system can produce peak torque values far in excess of calculated steady-state torque. Peak transient torque becomes the origin of drivetrain component fracture.

Vehicle tests confirm the need for controls that effectively manage the powertrain behavior to limit drivetrain peak transient torque. Successful vehicle test events produced results that turned the problem on by fracturing drivetrain components. Reproducing the same vehicle test events using revised engine controls to mitigate peak transient torque turned the problem off by preventing drivetrain component fractures.

Truck drivers reported no perceived impact on start-up or drivability with engines having controls activated that effectively managed the peak transient torque.

Truck OEMs, engine manufacturers and drivetrain component manufacturers need to continually work together to develop and implement control strategies that satisfactorily protect the drivetrain components. A control strategy is more effective than simply up-sizing fractured drivetrain components. Up-sizing, though it will solve the problem with that particular component, will only transfer the problem to the next weakest component in the drivetrain system. The outcome is larger, heavier and more costly components to overcome the unintended effect of downspeeding for fuel efficiency gains and greenhouse gas emissions standards.

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