Development and Vehicle Integration of XWD Driveline Technology

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Abstract

The continuously ongoing development of new driveline hardware, improved vehicle state sensing capabilities and sophisticated vehicle dynamics control algorithms are strong enablers for improvements in both safety and functional performance of road vehicles [2]. The present paper describes one of the latest achievements in this area, namely the development and vehicle integration of a state of the art all-wheel-drive driveline, named XWD or cross-wheel-drive [1], in the Saab 9-3.

This all-wheel-drive driveline concerns an active on demand driven rear-axle, coupled to the front by means of an electronically controlled hydraulic clutch to distribute a variable amount of torque from front to rear. In addition to this front to rear distribution, the system is capable of transferring torque laterally across the rear-axle as well using an electronic Limited Slip Differential. In the next section of this paper, a complete system overview will be provided, detailing each and every component and explain both their mutual and external interfaces. Once having presented all key characteristics, the paper will further expand on how these characteristics determine system behavior on a vehicle level, i.e. especially on vehicle dynamics properties like traction and handling performance. Furthermore, the paper will present specific development implications related to the integration in the Saab 9-3, i.e. how on one hand the vehicle architecture needs to be changed in order to incorporate the described system, and how on the other hand, the system itself is tailored to excel in this particular vehicle.

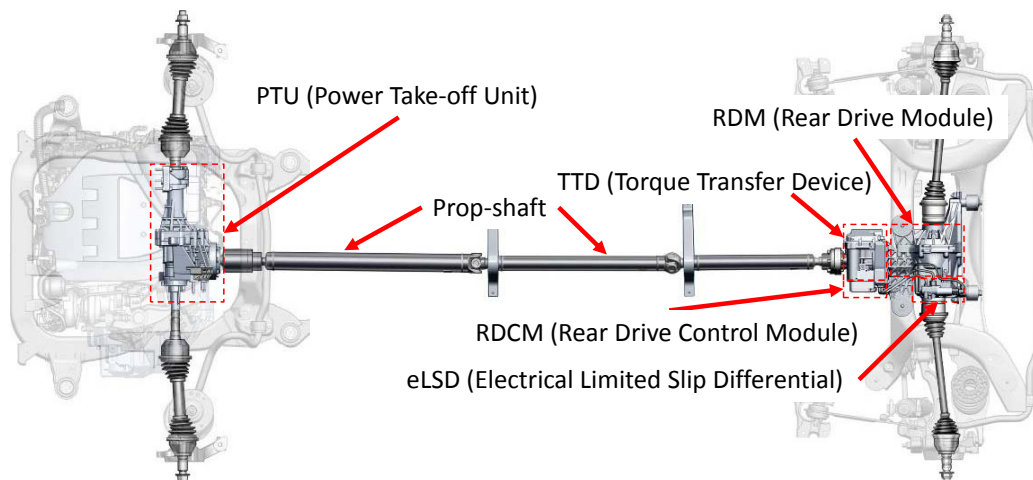*Corresponding author. e-mail: `hans-martin.duringhof-contractor@saab.com`

Figure 1: The Saab XWD System

1 System Design

1.1 Overview

The all-wheel drive system as developed by Saab can be classified as an active on demand all-wheel drive system. It is designed to fit a modified version of the GM Epsilon 1 platform and its recently released GM Global Midsize platform, both being front-wheel drive platforms from their origin.

The system is capable of transferring high torques from the front to the rear axle by means of an electro-hydraulic powerpack consisting of an electrically driven hydraulic pump and an electronically controlled hydraulic valve that controls pressure to the actual front to rear coupling. This pressure can be applied pre-emptively, unlike many systems on the market that rely on a differential speed between the input and output shafts of the front to rear coupling to build up pressure. A second electronically controlled hydraulic valve controls pressure from the same powerpack to the rear eLSD (electronically controlled Limited Slip Differential) to establish torque transfer laterally across the rear-axis.

An overview schematic is depicted in Fig. 1.

1.2 Hardware Components

Describing the system components from front to rear, the system can take power from the front axle driveline through a power take off unit and a propeller shaft that is connected to the torque transfer device which contains the front to rear coupling. The output shaft of this torque transfer device
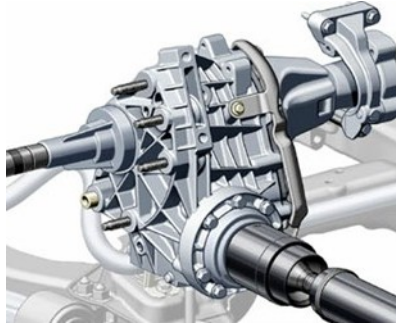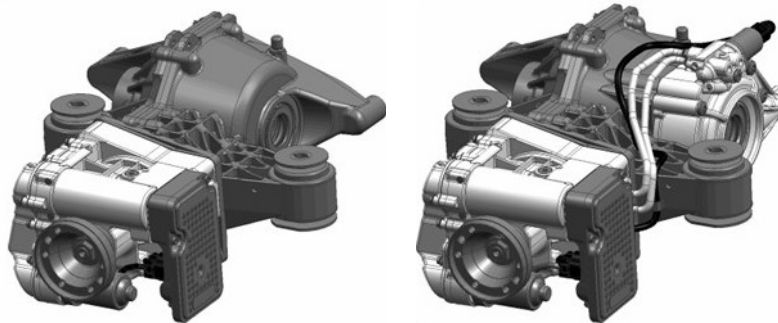
Figure 2: Power Take-off Unit

Figure 3: Rear Drive Module without eLSD (left) and with eLSD (right)

is subsequently connected to the rear differential, which is, together with the eLSD coupling embodied in the rear drive module. All components as mentioned here will be further detailed in the next subsections.

Power Take-off Unit   In order to transfer output torque from the transversely mounted transmission to the propshaft, a 90 degrees rotation is realized through the power take-off unit (PTU). Together with this rotation, also a 217:78 ratio is achieved.

Torque Transfer Device   A central part in the described system is the torque transfer device (TTD) shown in Fig. 3, consisting the electro-hydraulic actuator that varies the amount of torque that is transferred to the rear axle. It has one electric driven piston/revolver type pump, which establishes the working pressure of 30 bars. The pressurized oil is pushed through a paper based oil filter, into an accumulator to store energy which can be quickly released when needed. Besides providing the advantage of quick response, this accumulator also reduces overall current draw. The release of energy out of the accumulator is realized through two pulse-width modulated control

valves, each connected to a piston that presses the TTD and eLSD clutch packs respectively. As such, both clutch packs are energized by the same pump/accumulator source. The transferred torque through the clutch packs is closely correlated to the controlled oil pressure. Sensors in the system measure currents to the pump and the valves, which are used to feed an oil pressure and temperature model that facilitates an estimation of the transferred torque.

The locking level of the TTD is fully controllable between 0 and 100% independent of vehicle status, where the upper limit allows for a torque transfer equal to 1000Nm at the clutch, which, via the final drive, is equivelant of 2700Nm on the rear wheels (combined).

Rear Drive Module Right behind the TTD, the transferred torque is transmitted 90 degrees through the ring and pinion gearset in the rear drive module (RDM), including a ratio of 78:216. The output is subsequently split by the differential to the rear halfshafts. The RDM is assembled to the vehicle rear subframe at 4 mounting locations. Two bushings, located at the front of the RDM are vertical in orientation and retained by the RDM. The other two bushings are mounted in the subframe of the vehicle and have a horizontal orientation.

Electronically controlled Limited Slip Differential (eLSD) The RDM can optionally be equipped with an electronically controlled LSD (eLSD). This additional clutch pack can connect the differential carrier with the right half shaft via a disc package making it possible to have up to 1200Nm difference in torque level between the two half shafts. The eLSD acts as a hang on to the TTD and uses the ECU and power pack (feeder pump, filter and accumulator) on the TTD. The unit has its own pressure reducing valve which is used to control the pressure to the piston, similar as mentioned for the TTD. Two pipes connect the eLSD with the TTD; one is feeding the unit with pressurized oil and the other is the return pipe. The locking level of the eLSD is fully controllable between 0 and 100% during any driving situation.

Rear Drive Control Module The XWD system is controlled via an electronic controller called the rear drive control module (RDCM) that is mounted on the TTD/RDM assembly, as can be seen in Figure 3. This controller is connected to the vehicles CAN bus as well as battery system with a 12-pin Delphi GT150 connector. Internally the RDCM connects to the feeder pump via a 4-pin Delphi GT150 connector (2 pins used to connect to eLSD pressure reducing valve when applicable) and directly to the TTD pressure

reducing valve. The RDCM utilizes several signals present on the vehicle's CAN bus such as:

- Wheel speed signals

- Steering wheel angle

- Yaw rate and acceleration signals

- ABS signal

- ESP signals

- Engine speed

- Engine torque

Signals are used to detect vehicle status and to compute a suitable torque level for the present driving situation, according to the control logic as described in the next section.

Control Logic   The control of the TTD and eLSD couplings contains both feed forward as well as feedback based functionality (see Fig. 4). The main functional objectives of the feed forward controls may be divided between those used to control vehicle traction capability and those used to control vehicle lateral dynamics/handling, respectively. The feedback controls used have the overall objective of enhancing vehicle traction, not lateral dynamics. Apart from the sensor signals shown in Fig. 4, the sensor signals are pre-processed, failures detected and the states identified as well that control references and target values are calculated.

The TTD's feed forward controller decides on a torque split according to the desired longitudinal vehicle acceleration (which can be derived from driver input or a vehicle system like adaptive cruise control) and the measured lateral vehicle acceleration. On top of this, the feedback functionality aims to control the axle speed difference between the front and rear axle of the vehicle. The eLSD controls is purely feed forward with the main input signals being the requested longitudinal acceleration and measured lateral acceleration. A number of vehicle states (such as "Braking", "Handbrake applied", "Tight turn", etc) can also be identified by the algorithm and the controlled system behavior can be adapted correspondingly. Note that both the TTD and eLSD feed forward controls can utilize the system's pre-emptive capabilities to lock the clutch packs already at standstill or when driving without any axle speed difference between the front and rear wheels.
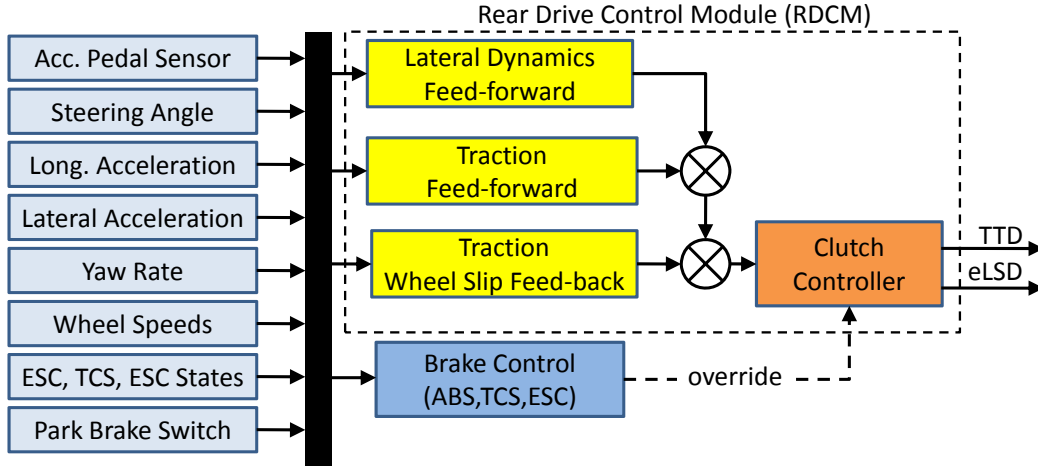
Figure 4: TTD and eLSD controls functional description

2 Traction Performance

2.1 Front-Rear Distribution

The most straight forward and obvious way to increase traction performance by an all wheel drive system is to utilize the driveline's ability to transfer torque from the front axle to the rear-axle. As such, this ability is an important design parameter for the torque transfer device that was discussed in Section 1.2. The amount of torque transfer that is actually required for a certain application, can be derived in several ways, mostly relying on load cases that represent real-life situations for which the customer expectations can be easily identified. Typical load cases are related to gradeability and/or trailer towing scenarios, as can be described by (1) [1].

$$M_R^{\text{req}} = R_w g((m_v + m_t) \sin \alpha - \mu m_v (l_R \cos \alpha + h \sin \alpha)/L, \qquad (1)$$

where m_v and m_t are the vehicle and trailer masses, respectively, α the road inclination angle, l_R/L the weight bias to the front axle, μ the road friction, h the z-position of the center of gravity and L the wheelbase.

When assessing a high performance vehicle, one might also consider the maximum torque potential from the internal combustion engine, together with the front-axle's friction limits to set a target for torque transfer capability. In Fig. 5 it is presented how much longitudinal acceleration could be realized purely from a engine/gearbox point of view, depending on vehicle speed, and for each and every possible gear. As can be seen as well, the limiting factor at low speeds (up to nearly 100km/h) is the front axle's friction, leaving a lot of potential for torque distribution to the rear axle.
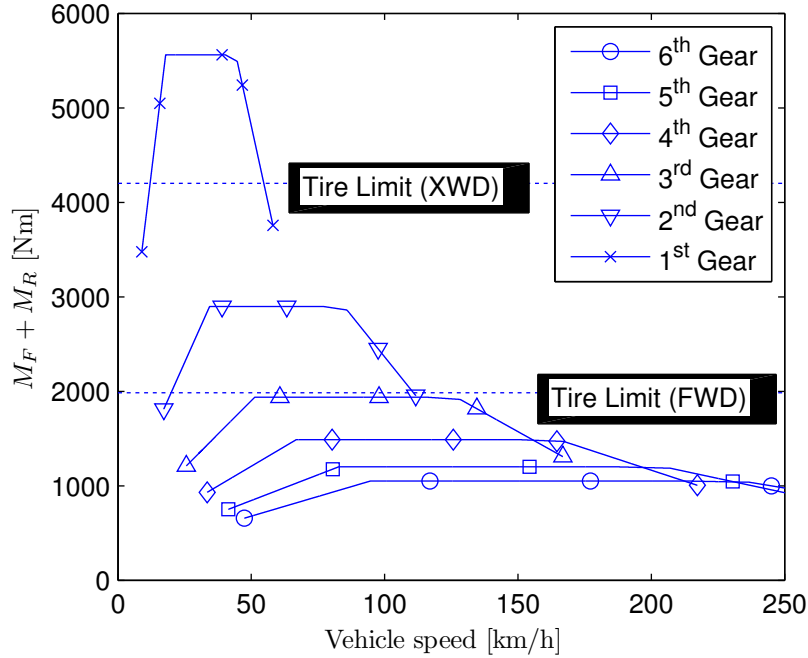
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Figure 5: Acceleration capacity in each gear

The above considerations have lead to a torque-transfer requirement of 2800 Nm, expressed as wheel torque, which is approximately the friction limit of the rear-axle on a Saab 9-3.

2.2 Left-Right Distribution

Split-μ Traction   In case of unequal friction levels on the left and right wheels, a regular open differential suffers from the fact that the traction torque on both wheels is equal to and therefore limited by the low friction side. The eLSD however can transfer torque from the low friction wheel to the high friction wheel, by limiting the wheel speed difference. As mentioned in section 1.2, the eLSD's clutch pack can hold a torque that results in 1200 Nm of torque difference between the wheels, meaning that almost all traction that one single wheel can provide on a high friction surface can be utilized, even if the other wheel is on ice.

Cornering Traction   Another occurrence, where a conventional open differential would limit traction, is when the vehicle is cornering. Due to load transfer from the inner to the outer wheel, the inner wheel will loose traction

more early than in the straight driving case, while the outer wheel is not able to utilize the additional traction it gets because of the increased load. Similar to the aforementioned split-? case, the eLSD can achieve a torque difference over the differential in favour of the outer wheel and hence the total traction. This torque-difference however, has to be controlled carefully in order not to waste lateral grip, which is becoming more and more dependent on the outer wheel as well when the lateral acceleration increases. This described effect will be further analysed in chapter 3, as this is clearly on the boundary of combined handling and traction performance.

3 Handling Performance

Besides the absolute torque transfer, that is primarily dimensioning the active on demand clutch, the relative torque transfer has to be considered as well. The reason is particularly because of its effect on combined longitudinal and lateral vehicle dynamics. In this section we will briefly summarize the main results in [1] that are relevant to this work. If we assume a front-wheel drive vehicle, the drive torque on the front axle will deteriorate the lateral grip of the front axle. Since the total lateral grip of the vehicle is determined by the axle with the least lateral grip, the lateral grip would be improved if the front and rear axles could share the total drive torque. If we study a steady-state turn, the lateral grip, $\bar{a}_Y$, can can be expressed as

$$\bar{a}_Y = \frac{L}{m_v} \min\left(\frac{\bar{F}_{YF}}{l_R}, \frac{\bar{F}_{YR}}{l_F}\right), \tag{2}$$

if we assume equal left/right drive torque distribution and small steering angles. The lateral grip limit $\bar{F}_Y$ for the front and rear axle is given by the lateral grip limit of the left and right wheels on that axle as

$$\bar{F}_Y = \sum_{i=1}^{2} \sqrt{(\mu(F_Z + \zeta(-1)^i \bar{F}_Y))^2 - \left(\frac{M}{2R_w}\right)^2}, \tag{3}$$

where $i = 1, 2$ for the left and right wheel respectively and ζ is the lateral load transfer coefficient and M is the drive torque on that axle. Solving for $\bar{F}_Y$ we have that

$$\bar{F}_Y = \begin{cases} \sqrt{(\mu F_Z)^2 - \frac{1}{1-\theta^2}\left(\frac{M}{2R_w}\right)^2} & \text{if} \quad |M| \leq R_w \mu F_Z(1-\theta^2), \\ \frac{1}{\theta}\left(\mu F_Z - \frac{|M|}{R_w}\right) & \text{otherwise,} \end{cases} \tag{4}$$

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where $\theta = 2\mu\zeta$.

In Figure 6 we can see a $\bar{F}_Y$ as function of the total drive torque, $M_F + M_R$ and the front/rear difference $M_F - M_R$. The boundaries of this diagram are where $\bar{F}_{YF} = 0$ along the bottom diagonal and where $\bar{F}_{YR} = 0$ along the upper diagonal. Along with these limits, we have a dark grey area where the rear axle limits the lateral grip, i.e. $\bar{F}_{YF} > \bar{F}_{YR}$ and vice versa for the light grey area. The dividing line is where $\bar{F}_{YF} = \bar{F}_{YR}$ and which front/rear drive force distribution maximizes the lateral grip for a given total drive torque. For reference, also some fixed drive force distributions are shown as dotted inclined lines. The vehicle we discuss in this work is front-wheel driven (100:0 front/rear distribution) when the TTD is open. When the TTD is locked, however, we will have a front/rear drive force distribution that closely follows the normal force distribution. The reason is that the traction stiffness of the axle is approximately proportional to the normal load on the axle. Further, when the TTD is locked, the front and rear wheels are forced to rotate at the same average wheel speed. This combined will result in a front/drive force distribution where

$$\frac{M_F}{F_{ZF}} \approx \frac{M_F}{F_{ZF}} \Rightarrow (M_F - M_R) \approx (M_F + M_R)\frac{g(L - 2l_R) - 2ha_X}{gL}, \quad (5)$$

which shown as a dashed line in Figure 6. The front/rear distribution changes if the ratio of the PTU and the RDM is not 1:1, which is the case in the system described here. In this application a gear ratio equal to 217:216 is chosen which means that the rear wheels are running 0.46% faster than the front wheels with a locked TTD. This will result a more rear-biased drive force distribution as can be seen with the line with square markers in Figure 6. For reference the effect of a front/rear gear ratio of 51:50 is depicted as well, which is very rear-wheel biased but requires excessive slipping of the TTD in order avoid the rear axle "pushing" the front wheels in the low-torque region.

3.1 Left-Right Distribution

To understand the realisation of torque difference between the left and right rear-wheels through the eLSD, we first analyse the optimality of the total optimal drive force distribution with certain system constraints. The optimal drive force distribution during acceleration in a turn is for our vehicle the one that

$$\max_{x \in \Re} a_Y, \quad (6)$$

with optimization parameters

$$x = \begin{bmatrix} F_{YFL} & F_{YFR} & F_{YRL} & F_{YRR} & \Delta M_R \end{bmatrix}^\top, \quad (7)$$
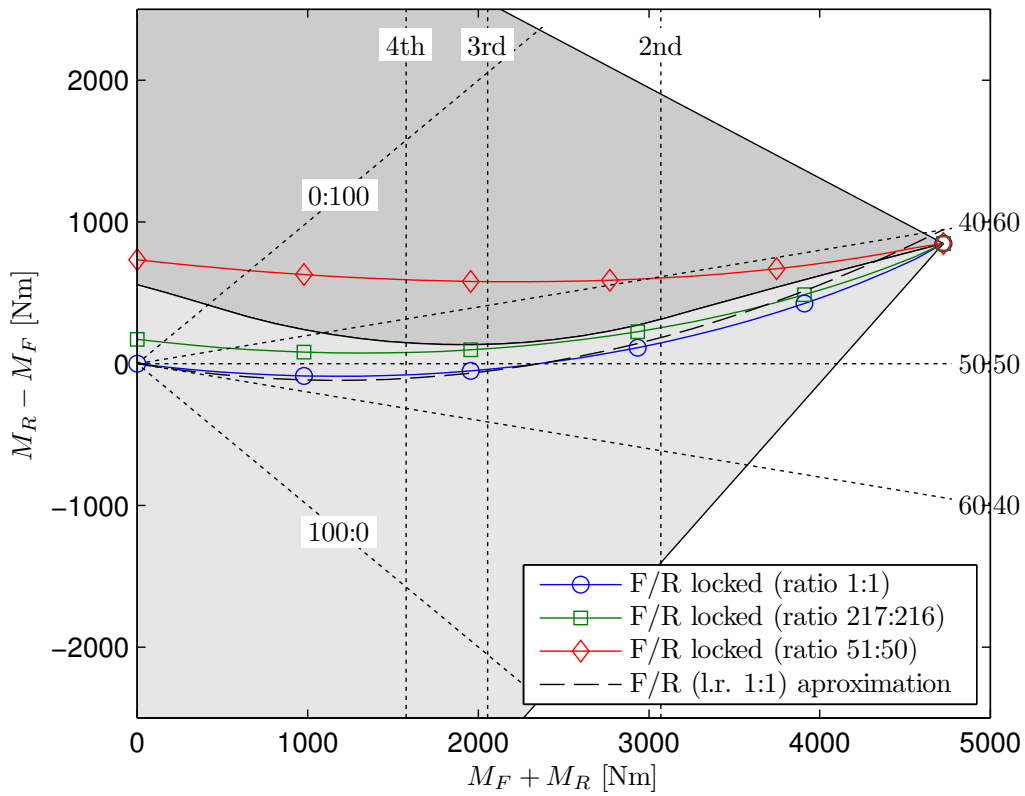
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Figure 6: Front/rear drive torque distribution as function of the total drive torque for a locked TTD with different speed ratios front/rear as well as some fixed front/rear distributions (diagonal dotted lines). The total drive force at maximum engine torque for different gears is shown with vertical dotted lines.
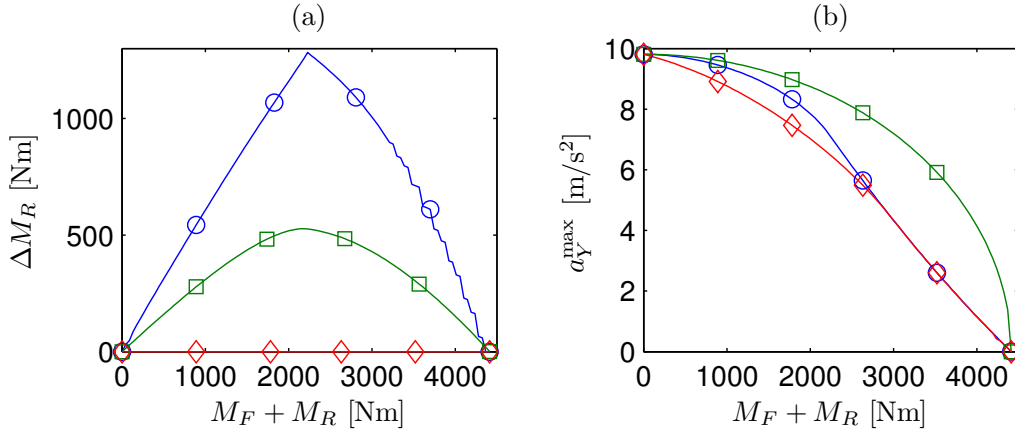
Figure 7: Optimal left/right distribution (a) for variable front and rear axle torque ((1) square markers), only differentiable rear axle torque ((2) circle markers) or open differential ((3) diamond markers). The maximum lateral acceleration for the these three configurations are shown as function of the total drive torque in (b).

subject to

$$F_Y^2 - \left((\mu F_Z)^2 - \left(\frac{M}{R_w} \right)^2 \right) \le 0 \tag{8}$$

for all four tires and while maintaining yaw balance which means that

$$\begin{bmatrix} l_F & l_F & -l_R & -l_R \end{bmatrix} x = 0. \tag{9}$$

The front/rear distribution is maintained according to (5). The result of this optimization can be seen in Fig. 7 where the left/right distribution was varied only on the rear axle as well as on both the front and rear axle. In Fig. 7.b the left/right distribution on the rear axle is shown for the case where the available grip of the front inner wheel is used for traction, there is no additional benefit of varying the left/right torque distribution only on the rear axle.

Next we study the effect of locking the rear differential with a given total drive torque and a front/rear distribution as previously given by (5). We are interested to study the torque difference versus lateral acceleration for the given total drive torque. As can be seen in Fig. 8, for a specific minimum total drive torque there exists a lateral acceleration where the difference in drive torque on the rear axle is zero. In [5] this point where the left/right wheel speed and drive torque is equal is called the cross-over point. Below
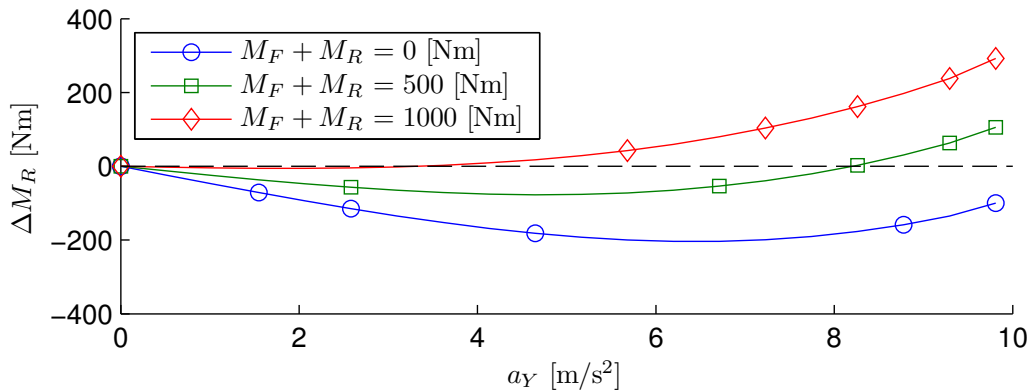
Figure 8: Left/Right Torque distribution of a locked rear differential versus the lateral acceleration for a given total drive force. The speed in this example is 70 km/h

this cross-over point, the eLSD has a yaw-damping effect by providing a turn-out yaw moment, while the eLSD provides a turn-in yaw moment for lateral acceleration above this cross-over point.

The results in Fig. 8 reflect the difference in drive torque as function of lateral acceleration for locked eLSD for given drive torque and lateral acceleration. For a given lateral acceleration, the torque required to reach the lock-over point is strongly dependent on vehicle speed as can be seen in Fig. 9.

4 System Integration

4.1 Physical Integration

The rear suspension is a 4-link type rear suspension as is the case for the FWD vehicle. The rear suspension was modified with a new geometry to fit the XWD system. Further a new wheel bearing was required to accommodate the drive shafts. This wheel bearing was chosen to be the same as for the front axle. In order to accommodate the driveline the vehicle was raised 5mm versus the standard chassis and with tires with a 13mm larger static radius, in total 18mm extra ground clearance was gained. This, however, was not sufficient to create a large enough fuel tank, so ride travel from design height was reduced from 113mm to 76mm. In order to compensate for the loss of travel, self-leveling dampers are provided standard. This ensures 74mm jounce travel from a maximum loaded position. A further important
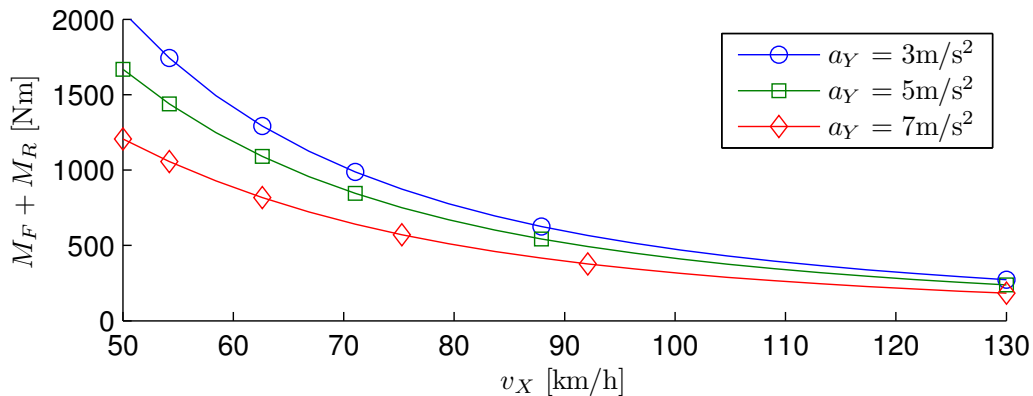
Figure 9: Torque required for the difference in left/right torque to be zero ($\Delta M_R = 0$) as function of vehicle speed for a locked eLSD and a given lateral acceleration.

change versus the FWD version was that the jounce bumper was moved from the spring link to the damper. From a characteristics point of view, the new wheel bearing and the change in suspension layout doubled the camber stiffness of the XWD suspension when compared to the FWD version. For more information on the systems integration, see [4] and [3].

4.2   Controls integration

While the selection and integration of hardware components obviously determines potential system behaviour to a very large extend, the system behaviour that is actually realized is more and more the result of the controls integration when it comes to electro-mechanic vehicle systems as the driveline discussed in this paper. For that reason as well as to accommodate proper integration with other chassis control systems like ESC (Electric Stability Control), Saab typically chooses for high involvement in this process as well as for maximum tuning/calibration flexibility. This high involvement was initiated already during the base development of software functionality, i.e. prior to the calibration and tuning of it, which provided the opportunity to guide the overall system development such that it perfectly matches the Saab philosophy in every part. Obviously, the tuning and calibration part, which took place on partly high friction surfaces but to a very large extend low friction (ice, snow) and gravel surfaces, was essentially governed by in-house Saab engineers.
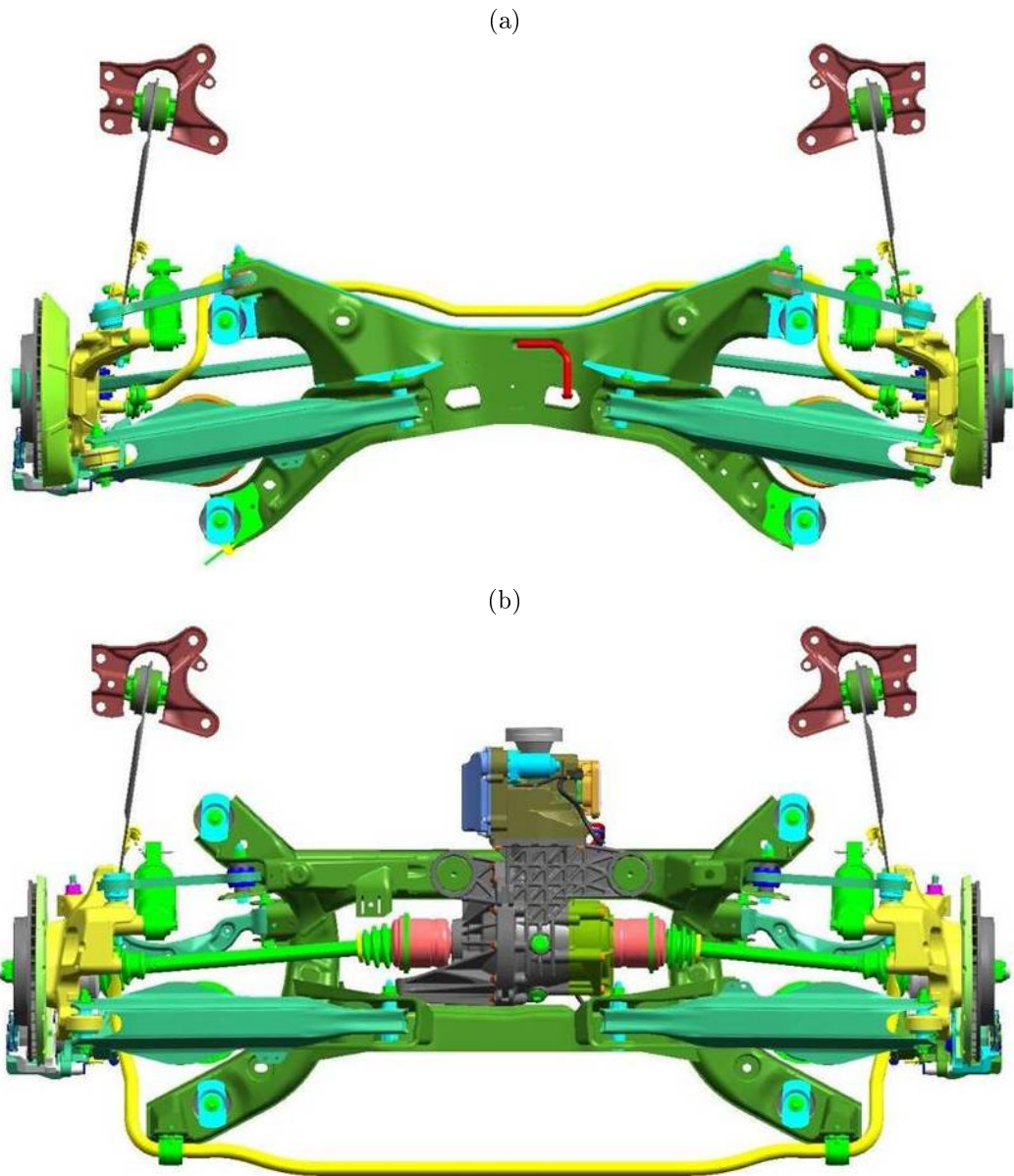
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(a)

(b)

Figure 10: Original FWD rear suspension (a) and modified XWD rear suspension (b).

5   Performance Evaulation

This section describes two vehicle tests, used to evaluate the performance of the all wheel drive system. Two driving scenarios are presented here to show the influence of both front/rear and rear/left torque distribution on vehicle dynamics. First there will be the effect of transferring drive torque to the rear in order to increase lateral grip on the front wheels as was described in Section 3.1. This effect can be effectively seen when performing an "Acceleration In Turn" (AIT) maneuver. A second case, now focussing at left/right torque distribution, concerns more or less the opposite manoeuvre, a "Drop Throttle In Turn" (DTIT) test.

5.1   Simulations Environment

In order to support development, a complete vehicle model has been used for preliminary performance evaluations. The vehicle model itself is made in CarSim [6], and can be used in co-simulation with a Matlab/Simulink environment where the controls are modelled, as well as part of the driveline hardware. The controls model actually uses roughly the same code as the production RDCM in the real vehicle.

5.2   Simulation results

Acceleration in Turn (AIT)   This maneuver starts with a steady state corner, i.e. constant speed and constant steering wheel angle, which is then followed by a sudden step on the throttle, requesting maximum longitudinal acceleration. The yaw-rate response during this maneuver is depicted in Fig. 11 for both a front wheel drive vehicle configuration as well as the XWD configuration, showing that the yaw-rate drop is much less pronounced in the latter case.

Drop Throttle in Turn   Now again the vehicle is initially kept at a constant speed and constant cornering radius, however, the throttle is now fully released at a certain point in time. This induces a sudden load transfer to the front of the vehicle because of the occurring deceleration, resulting in a shortly increasing yaw rate. When under such conditions the eLSD of the XWD system is locked, the yaw damping that is described in Section 3.2 achieves a smoother and more stable yaw response as shown in Fig. 12.
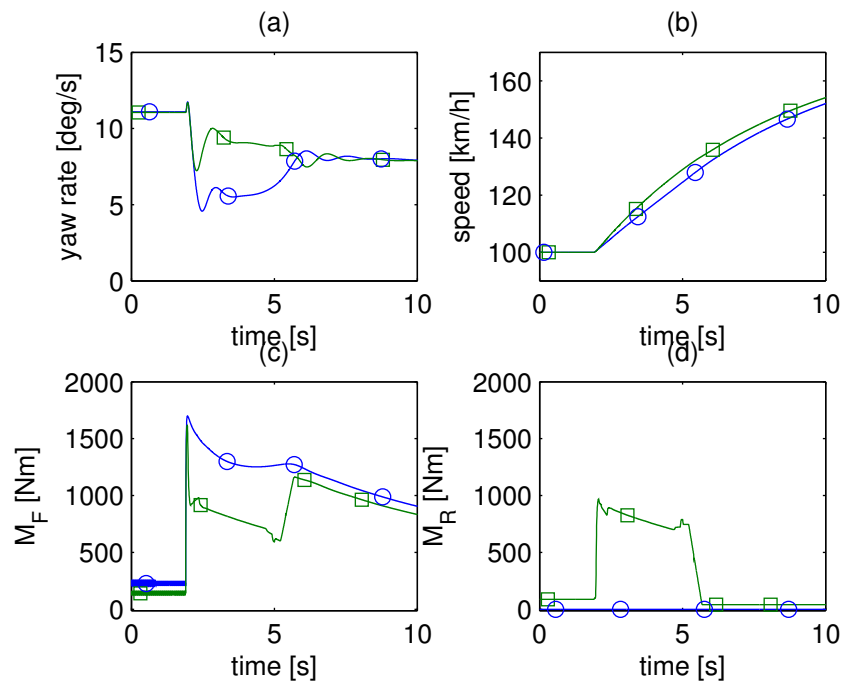
Figure 11: Yaw rate, speed and axle torque response under an Acceleration In Turn (AIT) maneuver with fixed steering input. The lines marked with a circle indicate an FWD reference vehicle and the lines with square markers are for a XWD vehicle, which distributes torque to the rear axle when the throttle is applied.
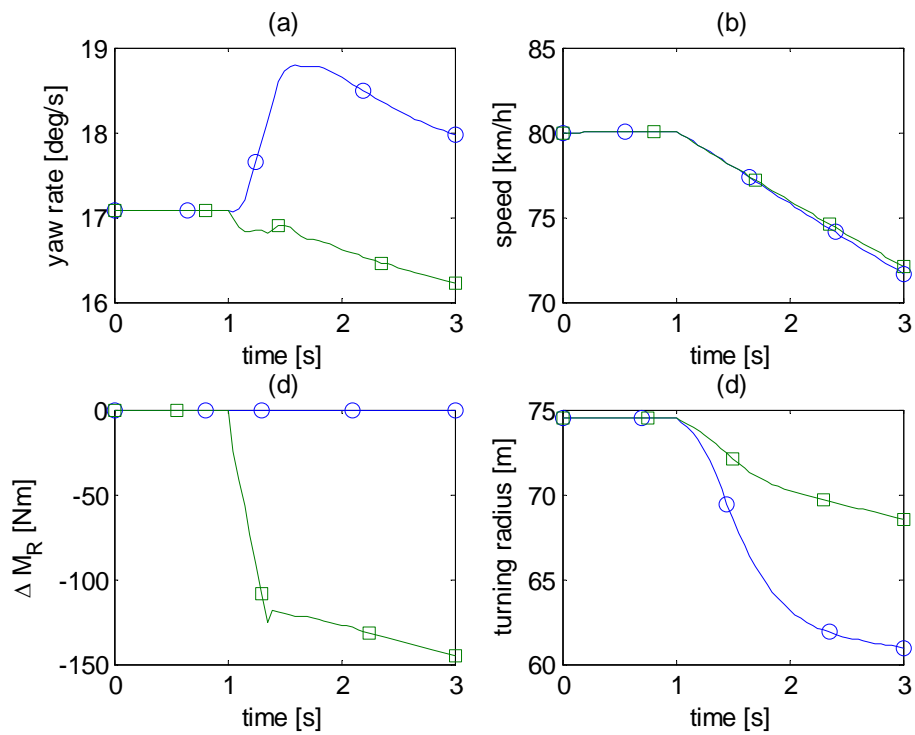
Figure 12: Yaw rate, speed and turning radius response under Drop Throttle In Turn (DTIT) maneuver under a fixed steering input (60°). The lines marked with a circle indicate a vehicle without eLSD and the line with square markers are a vehicle where the eLSD is locked when the throttle is dropped.

6 Conclusions

The present paper has revealed many details of the new Saab XWD system in terms of design, system characteristics and vehicle integration. Furthermore, theoretical background was provided for the understanding of possibilities to enhance traction and handling through managing drive torque distribution. This drive torque distribution in this case was only considered within the constraints of an active on demand all wheel drive system with an electronically controlled limited slip differential. Dependencies on total drive torque, lateral acceleration and vehicle speed of these effects were derived and graphically shown. Finally, two sample performance verification tests were shown to assess state of the art capabilities of this driveline system in real-life driving scenarios.

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Description	Symbol	Unit	Value
Vehicle mass	m	[kg]	1675
Yaw radius of gyration	ξ	[m]	1.32
Wheel base	l	[m]	2.675
Distance of front/rear axle to mass center	l_F/l_R	[m]	$0.4L/0.6L$
Distance ground to mass center	h	[m]	0.5
Front/rear track width	w_F/w_R	[m]	1.524/1.506
Front/rear axle lateral load transfer coefficient	ζ_F/ζ_R	[-]	0.17/0.16
Tire radius	R_w	[m]	0.32

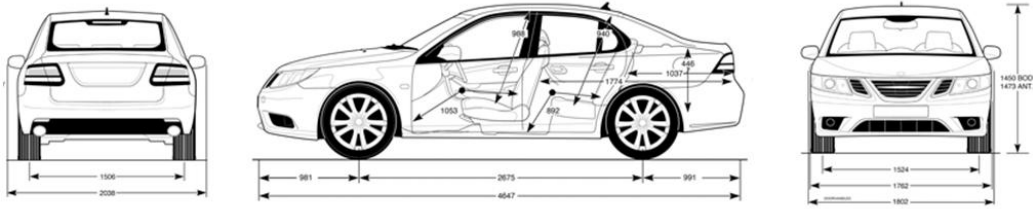Table 1: Vehicle Dimensions

Figure 13: Overall Vehicle Dimensions

A Vehicle Data

The vehicle data used in the performed simulation studies are given in Tab. 1 with the overall vehicle dimensions shown in Fig. 13.