AWD/4WD Transfer Case for Active Torque Split
Between the Axles

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ABSTRACT

In the recent years, the all-wheel drive transfer system has been developed increasingly. This thesis mainly focuses on various means of AWD/4WD transfer case for active torque split. This paper mainly discusses about transferring torque into front and rear axles by different couplings.

The first part of thesis illustrates several AWD/4WD transfer cases. In this step, the thesis shows description of each AWD/4WD transfer case. The second part, the paper shows the mathematic analysis about torque and rotational speed distribution of input, front and rear axles for each transfer cases. In the third part, the paper presents Simulink models of transfer cases in AWD/4WD vehicles. In this part, the thesis also shows the comparison and analysis between the Simulink models and schematic layout of transfer cases. The last part of thesis shows the condition and results of Simulink model’s running. In this part, the thesis also presents show the analysis of the results and conclusions.
Table of contents

1 Introduction ................................................................................................ 8
  1.1 Limited slip differential ................................................................. 8
    1.1.1 clutch-type LSD (the ZF differential) ........................................... 8
    1.1.2 Gear-type LSD (torsen) .......................................................... 9
    1.1.3 viscous-type LSD ............................................................... 10
  1.2 Haldex(Audi TT quattro) ............................................................. 11
  1.3 viscous-coupling system (Audi R8 sports car) .......................... 12

2 The layout and analysis of transfer case .................................... 13
  2.1 Limited slip differential ............................................................... 13
    2.1.1 ZF differential ........................................................................ 13
    2.1.2 Torsen C differential ............................................................ 15
  2.2 Open differential ........................................................................ 16
  2.3 Haldex clutch(Audi TT quattro) ................................................ 17
  2.4 Viscous coupling .......................................................................... 18
  2.5 Torque Vectoring Transfer Case ................................................ 19
  2.6 Chain drive transfer case ............................................................. 20
  2.7 Transfer case for active torque bias System ............................... 22
  2.8 Transfer case for AWD on-road vehicle (Mercedes) .................. 24

3 Simulink model with Simscape Driveline ................................. 26
  3.1 The model of 2WD vehicle ............................................................ 26
  3.2 4WD vehicle with different central differential ......................... 29
    3.2.1 Open central differential ..................................................... 29
    3.2.2 Self-locking central ZF differential ....................................... 31
    3.2.3 Torsen center differential .................................................. 35
  3.3 The model of differential Test ..................................................... 38
  3.4 Four-wheel drive vehicle with Viscous coupling ....................... 46

4 SIMULATION RESULTS ................................................................. 48
  4.1 Working condition ................................................................. 48
4.1.1 Parameters.................................................................48
4.1.2 Road surface .............................................................48
4.2 Results ..............................................................................51
4.2.1 Engine Performances .....................................................51
4.2.2 Performance of coupling devices .................................53
4.2.3 VEHICLE PERFORMANCE .............................................61

Conclusion..............................................................................62

Bibliography............................................................................63
Table of figures

Figure 1-1: ZF differential ........................................................................................................ 8
Figure 1-2: schematic diagram of ZF differential ............................................................... 9
Figure 1-3: Torsen differential ................................................................................................. 9
Figure 1-4: Torsen C differential .......................................................................................... 10
Figure 1-5: Viscous-Coupling Differential ........................................................................... 10
Figure 1-6: driveline of Audi TT quattro ............................................................................ 11
Figure 1-7: Haldex differential ............................................................................................ 11
Figure 1-8: Viscous clutch ....................................................................................................... 12
Figure 2-1: ZF differential ......................................................................................................... 13
Figure 2-2: schematic diagram of upper part .................................................................. 14
Figure 2-3: schematic diagram of Lower part ......................................................................... 14
Figure 2-4: schematic diagram of self-locking differential ............................................. 15
Figure 2-5: schematic diagram of open differential .......................................................... 16
Figure 2-6: stick diagram of Haldex AWD system ........................................................... 17
Figure 2-7: schematic diagram of Viscous coupling .......................................................... 18
Figure 2-8: Layout of TV transfer case ................................................................................ 19
Figure 2-9 stick diagram of TV transfer case ................................................................. 19
Figure 2-10 Chain drive transfer case ................................................................................. 20
Figure 2-11: stick diagram of chain drive transfer case ............................................... 21
Figure 2-12 layout of Transfer case for active torque bias System .................................. 22
Figure 2-13: stick diagram of Transfer case for active torque bias System ................... 22
Figure 2-14: Torque distribution versus clutch torque ..................................................... 23
Figure 2-15: layout of transfer case AWD on-road vehicle ......................................... 24
Figure 2-16: stick diagram of transfer case of AWD on-road vehicle ......................... 25
Figure 3-1: the model of FWD vehicle................................................................................ 26
Figure 3-2: Parameters of engine torque............................................................................... 27
Figure 3-3: Parameters of engine dynamics ....................................................................... 27
Figure 3-4: Parameters of vehicle body block................................................................. 28
Figure 3-5: The model of 4wd vehicle................................................................................ 29
AWD/4WD transfer case for active torque split between the axles

Figure 3-6: schematic diagram of open differential .......................................................... 29
Figure 3-7: Model layout of Self-locking central ZF differential .................................... 31
Figure 3-8: The model of Torque to pressure .................................................................. 32
Figure 3-9: The parameters of Input Pinion Crown Gear ............................................ 33
Figure 3-10: The parameters of Clutch pack ................................................................. 33
Figure 3-11: The parameters of Sun-Planet Bevel ....................................................... 34
Figure 3-12: 3-12: Torsen center differential ................................................................. 34
Figure 3-13: Model layout of torsen differential ............................................................. 36
Figure 3-14: the layout of Sun-Planet Worm Gear ....................................................... 36
Figure 3-15: Parameters of Sun-Planet Worm Gear ...................................................... 37
Figure 3-16: Model layout for differential testing ......................................................... 38
Figure 3-17: Locking coefficient versus input torque for Limited-slip differential ................. 39
Figure 3-18: T Partial enlargement of Figure 3-22 .......................................................... 40
Figure 3-19: TBR versus input torque for Limited-slip differential ............................... 40
Figure 3-20: ΔT VS Δω for Limited-slip differential .................................................... 41
Figure 3-21: Locking coefficient versus input torque for open differential ............... 42
Figure 3-22: TBR versus input torque for open differential .......................................... 42
Figure 3-23: ΔT VS Δω for open differential ................................................................. 43
Figure 3-24: Locking coefficient versus input torque for torsen differential .............. 44
Figure 3-25: TBR versus input torque for torsen differential ....................................... 44
Figure 3-26: ΔT VS Δω for Torsen differential ............................................................. 45
Figure 3-27: Model layout of 4WD vehicle with Viscous coupling .............................. 46
Figure 3-28: Model layout of Viscous coupling testing ............................................... 46
Figure 3-29: Output torque VS Δω for Viscous coupling ............................................. 47
Figure 4-1: friction coefficient of different surface in different slip ............................. 49
Figure 4-2: schematic diagram of Simulink test .......................................................... 50
Figure 4-3: Engine torque of different transfer case ..................................................... 50
Figure 4-4: Engine speed of different transfer case ..................................................... 51
Figure 4-5: Torque/speed ratio between input and output axles .................................... 52
AWD/4WD transfer case for active torque split between the axles

Figure 4-6: Input torque of differentials
Figure 4-7: Rotational speed front and rear axles (OPDI)
Figure 4-8: Output torque of front and rear axles (OPDI)
Figure 4-9: Slips of front and rear wheels for OPDI
Figure 4-10: Rotational speed of front and rear axles (OPDR)
Figure 4-11: Partial enlargement of Figure 4-9
Figure 4-12: Output torque of front and rear axles (OPDR)
Figure 4-13: Slips of front and rear wheels for OPDR
Figure 4-14: Rotational speed of front and rear axles (TCD)
Figure 4-15: Output torque of front and rear axles (TCD)
Figure 4-16: Slips of front and rear wheels for TCD
Figure 4-17: Rotational speed of front and rear axles for viscous coupling
Figure 4-18: Output torque of front and rear axles (Viscous)
Figure 4-19: Slips of front and rear wheels for Viscous coupling
Figure 4-20: Slips of front wheels
Figure 4-21: Slips of rear wheels
Figure 4-22: The vehicle accelerations of different transfer cases
Figure 4-23: The vehicle speeds of different transfer cases
1. Introduction

Four-wheel drive (4WD)/all wheel vehicle (AWD) vehicles are almost as old as the vehicle itself, however for more than half a century, the main use of the 4WD/AWD system has been to improve off-road traction. The technology advances over the recent decades resulted in more advanced and light weight solutions shifting the focus more towards sports cars and performance passenger vehicles.

There is a little difference between 4WD and AWD. Four-wheel drive is a two-axle vehicle drive that provides torque to all of its wheels simultaneously. A four-wheeled vehicle with torque supplied to both axles is described as “all-wheel drive” (AWD). There is no clear distinction between AWD and 4WD, but usually 4WD vehicles contain a transfer case, which has a central differential and an optional two-gear 8eductor (LO-low and HI-high).

There many different kinds of coupling devices connected the front and rear axles in 4WD/AWD transfer case. In this part, I will introduce several devices.

1.1 Limited slip differential

There are 3 types of limited slip differential: clutch-type LSD (the ZF one), gear-type LSD (torsen) and viscous-type LSD.

1.1.1 clutch-type LSD (the ZF differential)

This type of LSD works like a multi-plate clutch in a high-performance car. The carrier consists of 2 parts: clutches and gears. The gears include the gearset like a open differential and 2 extra pinion gears.

Figure 1-1: ZF differential

This type of LSD works like a multi-plate clutch in a high-performance car. The carrier consists of 2 parts: clutches and gears. The gears include the gearset like a open differential and 2 extra pinion gears.
There are plate, friction and pressure ring in each side of the side gears. The pinions extend out and contact the pressure rings. When pressure rings open, it means the pinions pass through are shaped to move the pressure rings against the clutches and “lock up” the axle.

The “lock up” occurs when the clutches and plates squeeze against each other, it can increase the amount of torque to let them to spline.

1.1.2 Gear-type LSD (torsen)

Torsen is from Torque Sensing, a limited-slip mechanical differential. This differential is manufactured by Gleason Corporation. They can be used as front/rear differential or center (inter-axle) differentials.
The Torsen differential is fully mechanical with satellite and helical gears. Their self-locking characteristics depend on the difference in torque between the front and rear axles or between the left and right wheels.

![Torsen C Differential](image)

**Figure 1-4: Torsen C differential**

This differential is configured as a planetary gear. An internal gear surrounds the sun gear; rotating between these two elements are roller-shaped planet gears which are connected to a rotating housing. They distribute the torque asymmetrically – the larger part flows to the rear through the internal gear, which has a larger diameter, and the output shaft connected to it. The smaller part is transferred to the smaller sun gear, from where it is sent to the front axle. Under normal driving condition, the torque split ratio is 40:60 between the front and rear axles.

1.1.3 viscous-type LSD

![Viscous-Coupling Differential](image)

**Figure 1-5: Viscous-Coupling Differential**

It is basically an open differential added with a viscous coupling to provide anti-slip function. The differential normally offers 50:50 torque split ratio. When slip
occurs, the viscous coupling limits the speed difference while transferring more torque to the axle with better traction. Therefore, VCDL is a permanent 4WD system, but it shares some disadvantages of VC differential, such as slow response, non-linear torque transfer and longevity issues.

In no-slip driving conditions, the torque split ratio is 50:50. When slip on the front or rear axle is detected, a locking center differential is able to transfer to 80 percent of available torque to the axle that has the best traction. The center differential uses a viscous coupling that operates without the aid of computer control and reacts to the mechanical differences in grip.

1.2 Haldex (Audi TT quattro)

The Haldex clutch uses an electro-hydraulic multi-plate clutch that is typically placed near the rear axle to engage the rear axle when more traction is required. Normally the input shaft and the output shaft rotate at the same speed, so no torque is transmitted to the rear axle even when the clutch is engaged.

Figure 1-7: Haldex clutch
When a speed difference occurs, the pumping starts to generate oil flow immediately. Therefore, the clutch can transmit torque to the rear axle with greater traction. The more the clutch is engaged, the more the torque transmitted.

The oil flows to the clutch piston, compressing the clutch assembly and the speed difference between the brake shafts. The oil is returned to the reservoir through a controllable valve that regulates the oil pressure and the force on the clutch.

In traction/high slip conditions, a high pressure is transferred; in tight curves (i.e. parking), or at high speeds, a much lower pressure is provided.

1.3 viscous-coupling system (Audi R8 sports car)

Inside the clutch, there are many circular plates that are very close to each other. Half of them connect to the front drive shaft and the rest connect to the rear drive shaft. The sealed housing is fully filled with a highly viscous fluid, which has a strong tendency to stick those plates together.

![Figure 1-8: Viscous clutch](image)

To use the viscous coupling, the torque is transferred through the viscous friction (slip-controlled). The level of coupling torque depends on the clutch characteristic. It can be influenced by the level, viscosity and temperature of the oil (using silicone oil with low viscosity change and temperature fluctuations).

In normal condition, front and rear axles rotate at the same speed, so the plates and viscous fluid rotate together without relative displacement. When one of the shafts experiences a tire slip, the alternating plates spine at different speeds, thus the viscous fluid attempts to stick them together. As a result, torque is transferred from the faster drive shaft through the fluid to the slower drive shaft.

The greater the speed difference, the more torque it transfers to the slower drive
shaft and the more it resists the faster driveshaft. In other words, it provides anti-slip function.

2. The layout and analysis of transfer case

In this part, the paper mainly illustrates the transfer cases for each mechanical solution of torque splitting between front and rear axles. For each transfer case, we mainly investigate on the torque distribution on front and rear axles and the effect on vehicle dynamics. For calculating the torque distribution more easily, I sketch the stick diagram for each transfer case.

2.1 Limited slip differential

2.1.1 ZF differential

The multi disc wet clutches apply a constant (upper version) or a variable (lower version) locking coefficient.

![ZF differential diagram]

3-pressure plate; 4, 5-outer / inner discs; 7-diaphragm spring

Figure 2-1: ZF differential
Discs pressure force is about proportional to the input torque.

The locking torque $C_F$ is proportional to the total torque and always oppose the relative rotation of the half axles.

\[
C_F = N_{cl} R_{cl} \mu F_a = N_{cl} R_{cl} \mu M_h/R_p \tan \alpha \tag{2-1}
\]

\[
F_a = k M_h \quad \Delta M = 2 C_F \tag{2-2}
\]

Discs pressure force for low input torque is constant $F_{spring}$ (due to spring preload) and then proportional to the input torque.

\[
C_F = N_{cl} R_{cl} \mu F_a \tag{2-3}
\]

\[
\text{If } M_h < M_h^*(F_{spring}) \text{, } F_a = F_{spring} \tag{2-4}
\]

\[
\text{If } M_h > M_h^*(F_{spring}) \text{, } F_a = \frac{N_{cl} R_{cl} \mu}{R_p \tan \alpha} M_h \tag{2-5}
\]
2.1.2 Torsen C differential

![Diagram of Torsen C differential]

Figure 2-4: Schematic diagram of self-locking differential

Torque Bias Ratio (TBR):

\[ TBR = \frac{M_{\text{max}}}{M_{\text{min}}} \]  \hspace{1cm} 2-6

locking coefficient \( b \)

\[ b = \frac{M_{\text{max}} - M_{\text{min}}}{M_{\text{in}}} \quad |\Delta M| = M_{\text{max}} - M_{\text{min}} \leq bM_{\text{in}} \]  \hspace{1cm} 2-7

If \( |\Delta M| < bM_{\text{in}} \), the differential is locked.

If \( |\Delta M| = bM_{\text{in}} \), the differential is unlocked.

\[ M_F = \frac{M_{\text{in}}}{2} + \frac{b|M_{\text{in}}|}{2} \sin(\omega_R - \omega_F) \]  \hspace{1cm} 2-8

\[ M_R = \frac{M_{\text{in}}}{2} b|M_{\text{in}}| \frac{1}{2} \sin(\omega_R - \omega_F) \]  \hspace{1cm} 2-9

\[ \omega_{IN} = \frac{\omega_F + \omega_R}{2} \]  \hspace{1cm} 2-10
2.2 Open differential

![Diagram of open differential](image)

Figure 2-5: schematic diagram of open differential

IN: Torque/speed at ring gear
1: torque/speed at front shaft
2: torque/speed at rear shaft

$C_f$—friction torque

$$\omega_{IN} = \frac{\omega_1 + \omega_2}{2} (\tau = -1)$$  \hspace{1cm} 2-11

$$C_1 = \frac{C_{IN}}{2} \pm C_f \hspace{1cm} C_2 = \frac{C_{IN}}{2} \mp C_f$$  \hspace{1cm} 2-12

$\omega_1 > \omega_2$:  
$$C_1 = \frac{C_{IN}}{2} - C_f \hspace{1cm} C_2 = \frac{C_{IN}}{2} + C_f$$  \hspace{1cm} 2-13

$\omega_1 < \omega_2$:  
$$C_1 = \frac{C_{IN}}{2} + C_f \hspace{1cm} C_2 = \frac{C_{IN}}{2} - C_f$$  \hspace{1cm} 2-14

Ratio between highest and lowest torque

$$TBR = \frac{C_1}{C_2} = \frac{k_1 C_{IN}}{k_2 C_{IN}} = \text{const.}$$  \hspace{1cm} 2-15
2.3 Haldex clutch (Audi TT quattro)

By applying the Newton law to the joint discs:

\[ \tau = \frac{\mu \Delta V}{d} \]  \hspace{1cm} 2-16

\( \tau \) = shear on disc surface element dS.
\( \mu \) = dynamic viscosity coefficient.
\( \Delta V \) = speed local difference.
\( D \) = gap between discs.

Force acting on the surface element:

\[ \tau dr r^2 d\alpha = \tau r^2 dr d\alpha \] \hspace{1cm} 2-17

Total braking torque of the joint:

\[ b M = n (\pi \Delta \omega \nu \rho / 2d) r_e^4 [1 - (r_i/r_e)]^4 \]  \hspace{1cm} 2-18

\( r_i, r_e \) = inside, outside radii of facing surfaces.
\( N \) = number of facing surfaces.
\( \Delta \omega \) = rotational speed difference.

The locking coefficient \( b \) is therefore depending on the relative differential speed (speed-sensing).
2.4 Viscous coupling

The friction between the two shafts 2 and 3 is made by the viscosity of the silicon oil inside the joint, working on discs 5 and 6; the friction torque depends on the relative speed between 2 and 3.

the total braking torque:

\[ b_M = n \left( \frac{\pi \Delta \omega v \rho}{2d} \right) r_e^4 \left[ 1 - \left( \frac{r_i}{r_e} \right) \right]^4 \]  \hspace{1cm} 2-19

\( r_i, r_e \) = inside, outside radii of facing surfaces.
\( n \) = number of facing surfaces.
\( \Delta \omega \) = rotational speed difference.
the locking coefficient \( b \) is therefore depending on the relative differential speed (speed-sensing).

To simplify the equation, we can get:

\[ T_b = K^* \Delta \omega \quad \Delta \omega = |\omega_F - \omega_R| \]  \hspace{1cm} 2-20

So we can add a nonlinear damper between front and rear shaft to express this equation.
2.5 Torque Vectoring Transfer Case

\[ T_F = T_{engine} \times R_{Transmission} \times (1-R_{CD}) - T_{Actuator} \times TBR \]  

\[ T_R = T_{engine} \times R_{Transmission} \times R_{CD} - T_{Actuator} \times TBR \]

\( T_{engine} \) - Engine Torque

\( R_{Transmission} \) - Gain factor of main transmission ie gear ratio and torque converter amplification

\( R_{CD} \) - Bias ratio of the centre differential expressed as a ratio with respect to the rear axle.
$T_{\text{Actuator}}$ - Actuator torque. Positive when biasing to the rear axle.

### 2.6 Chain drive transfer case

![Figure 2-10: Chain drive transfer case](image)

1 Gearbox input; 2 bevel gear interaxle differential; 3 multi-plate clutch; 4 planetary gear set for cross-country gear; 5 front axle output; 6 rear axle output; 7 chain drive

The drive torque is conducted via a manual or automatic gearbox to the transfer gearbox 1. The transfer gearbox contains a bevel gear interaxle differential 2, a multi-plate clutch 3 for locking the interaxle differential and a planetary gear set 4 for shifting the cross gear ($I = 2.93$). With the bevel gear interaxle differential, 50% of the torque is conducted to the front axle and 50% to the rear axle. The locking torque of the multi-plate lock is modulated in a processor-controlled way up to fixed all-wheel drive, with a torque distribution corresponding to the axle load distribution. If needed, the cross-country gear can be shifted during operation by means of a synchronization.
AWD/4WD transfer case for active torque split between the axles

![Diagram of chain drive transfer case](image)

**Figure 2-11: stick diagram of chain drive transfer case**

<table>
<thead>
<tr>
<th>MODE</th>
<th>CL</th>
<th>( T_F = T_{IN} )</th>
<th>( T_R = 0 )</th>
<th>( \omega_{IN} = \omega_F ) ( \forall \omega_R )</th>
</tr>
</thead>
<tbody>
<tr>
<td>FWD</td>
<td>OFF</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>AWD</td>
<td>ON</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

\[
\sigma = \frac{R_R}{R_S} \\
T_c = \frac{1+\sigma}{\sigma}T_{CL} \\
\omega_R = (1+\sigma)\omega_c + \sigma\omega_{IN}
\]

**Table 2-1: Torque and rotational speed distribution for chain drive transfer case**

\[
T_{CL} = N_d \mu F_a R_d \sin(\Delta\omega), F_a = P^*A \\
\Delta\omega = \omega_s - \omega_r = \omega_F - \omega_R
\]

2-23

2-24
2.7 Transfer case for active torque bias System

The input shaft 32 and the primary output shaft 34 of the case 28 are joined together at the Splines 38 and cannot rotate relative to each other. No slippage occurs between the shafts 32 and 34.
There are two paths to transfer torque from input shaft to the Secondary output shaft 36.- a clutch path and a mechanical path.

\( T_{CL} \)- Torque transferred through the clutch path passes from the input shaft 32, through the clutch 50, and then to the Sun gear 78 which transfers it to the planet gears 88. The planet gears 88 in turn transfer it to the carrier 90 at the pins 92 which are in the planet gears 88.

\( T_m \)- The torque transferred through the mechanical path passes from the input shaft 32 through the end plate 86 to the ring gear 84. The ring gear 84 transfers it to the planet gears 88 where it combines at the pins 92 with torque transmitted through the clutch path.

The relationship between the torque transmitted through the clutch path of the torque bias coupling 44 and the torque delivered to the primary output shaft 34 and the secondary output shaft 36 in which the ratio \( u \) between the teeth on the Sun gear 78 and the teeth on the ring gear 84 is 4

\[
\frac{T_m}{T_{CL}} = u = 4 \quad T_F = T_{CL} + T_m \tag{2-25}
\]

\( T_F \)- The torque transferred into the front axles
AWD/4WD transfer case for active torque split between the axles

<table>
<thead>
<tr>
<th>MODE</th>
<th>CL</th>
<th>CL</th>
<th>CL</th>
<th>CL</th>
</tr>
</thead>
<tbody>
<tr>
<td>RWD</td>
<td>OFF</td>
<td>TR = TI</td>
<td>TF = 0</td>
<td>ωIN = ωR ∀ ωF</td>
</tr>
<tr>
<td>AWD</td>
<td>ON</td>
<td>TF = TCL + Tm = 5TCL</td>
<td>TR = TI - TF</td>
<td>ωIN = (1 + σ) ωF + σωR</td>
</tr>
</tbody>
</table>

Table 2-1: Torque and rotational speed distribution for Transfer case with active torque bias

System

\[ T_{CL} = N_d \mu F_a R_d \sin(\Delta \omega) \quad 2-26 \]

\[ F_a = P * A \Delta \omega = \omega_F - \omega_R \quad 2-27 \]

2.8 Transfer case for AWD on-road vehicle (Mercedes)

Figure 2-15 layout of transfer case AWD on-road vehicle\(^{13}\)
AWD/4WD transfer case for active torque split between the axles

Figure 2-16 stick diagram of transfer case of AWD on-road vehicle

<table>
<thead>
<tr>
<th>Mode</th>
<th>CL1</th>
<th>CL2</th>
<th>Equation</th>
<th>Condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>RWD</td>
<td>OFF</td>
<td>ON</td>
<td>$T_R = T_{IN}$, $T_F = 0$</td>
<td>$\omega_{IN} = \omega_R \forall \omega_F$</td>
</tr>
<tr>
<td>AWD</td>
<td>ON</td>
<td>OFF</td>
<td>$\frac{T_R}{T_{IN}} = \frac{1+\sigma}{\sigma}$, $\frac{T_F}{T_{IN}} = \frac{1}{\sigma}$</td>
<td>$\omega_F = (1 + \sigma) \omega_R + \sigma \omega_{IN}$, $\omega_{CL1} = \omega_F$</td>
</tr>
<tr>
<td>LOCKED</td>
<td>ON</td>
<td>ON</td>
<td>$T_F = \frac{1}{\sigma} \cdot T_{IN} + T_{CL}$</td>
<td>$\omega_{CL2} = \omega_F$</td>
</tr>
</tbody>
</table>

Table 2-2: Torque and rotational speed distribution for transfer case of AWD on-road vehicle

$$T_{CL} = N_d \mu F_a R_d \sin(\Delta \omega), F_a = P*A$$ \hspace{1cm} 2-28

$$\Delta \omega = \omega_s - \omega_r = \omega_F - \omega_R$$ \hspace{1cm} 2-29
3. Simulink model with Simscape Driveline

3.1 The model of 2WD vehicle

The block engine and torque converter are added to simulate the powertrain of vehicle. In this model, we consider the gearbox as a simple gearset, because the main object of this model is studied on the torque split between rear and front shaft. The block throttle is used to control the input torque of engine and the Block limit is to set a limit value of input torque.

**Engine Block**

The Generic Engine block represents a general internal combustion engine. Engine types include spark-ignition and diesel. Speed-power and speed-torque parameterizations are provided. A throttle physical signal input specifies the normalized engine torque. Optional dynamic parameters include crankshaft inertia and response time lag. A physical signal port outputs engine fuel consumption rate based on choice of fuel consumption model. Optional speed and redline controllers prevent engine stall and enable cruise control.

**Engine Speed, Throttle, Power, and Torque**

The engine model is specified by an engine power demand function $g(\Omega)$. The function provides the maximum power available for a given engine speed $\Omega$. The
block parameters (maximum power, speed at maximum power, and maximum speed) normalize this function to physical maximum torque and speed values.

The normalized throttle input signal $T$ specifies the actual engine power. The power is delivered as a fraction of the maximum power possible in a steady state at a fixed engine speed. It modulates the actual power delivered, $P$, from the engine: $P(\Omega,T) = T \cdot g(\Omega)$. The engine torque is $\tau = P/\Omega$.

<table>
<thead>
<tr>
<th>Port</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>B</td>
<td>Rotational conserving port representing the engine block</td>
</tr>
<tr>
<td>F</td>
<td>Rotational Conserving port representing the engine crankshaft</td>
</tr>
<tr>
<td>T</td>
<td>Physical signal input port specifying the normalized engine throttle level</td>
</tr>
<tr>
<td>P</td>
<td>Physical signal output port reporting the instantaneous engine power</td>
</tr>
<tr>
<td>FC</td>
<td>Physical signal output port reporting the fuel consumption rate</td>
</tr>
</tbody>
</table>

**Table 3-1: Description of port in engine block**

**Figure 3-2: Parameters of engine torque**

**Figure 3-3: Parameters of engine dynamics**

**Vehicle body Block**

The Vehicle Body block represents a two-axle vehicle body in longitudinal motion. The vehicle can have the same or a different number of wheels on each axle. The vehicle wheels are assumed identical in size. The vehicle can also have a center of gravity (CG) that is at or below the plane of travel.

The block accounts for body mass, aerodynamic drag, road incline, and weight distribution between axles due to acceleration and road profile. Optionally
AWD/4WD transfer case for active torque split between the axles

include pitch and suspension dynamics. The vehicle does not move vertically relative to the ground.

The vehicle axles are parallel and form a plane. The longitudinal, x, direction lies in this plane and perpendicular to the axles. If the vehicle is traveling on an incline slope, β, the normal, z, direction is not parallel to gravity but is always perpendicular to the axle-longitudinal plane.

**Parameters**

![Figure 3-4: Parameters of vehicle body block](image)

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass</td>
<td>1000</td>
<td>kg</td>
</tr>
<tr>
<td>Number of wheels per axle</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>Horizontal distance from CG to front axle</td>
<td>1.4</td>
<td>m</td>
</tr>
<tr>
<td>Horizontal distance from CG to rear axle</td>
<td>1.6</td>
<td>m</td>
</tr>
<tr>
<td>CG height above ground</td>
<td>0.5</td>
<td>m</td>
</tr>
<tr>
<td>Externally-defined additional mass</td>
<td>OFF</td>
<td></td>
</tr>
<tr>
<td>Gravitational acceleration</td>
<td>9.81</td>
<td>m/s²</td>
</tr>
<tr>
<td>Frontal area</td>
<td>3</td>
<td>m²</td>
</tr>
<tr>
<td>Drag coefficient</td>
<td>0.4</td>
<td></td>
</tr>
<tr>
<td>Air density</td>
<td>1.18</td>
<td>kg/m³</td>
</tr>
</tbody>
</table>

**Parameters of Tires**

![Parameterize by: Physical signal Magic Formula coefficients](image)

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rolling radius</td>
<td>0.3</td>
<td>m</td>
</tr>
<tr>
<td>Compliance</td>
<td>No compliance - Suitable for HIL simulation</td>
<td></td>
</tr>
<tr>
<td>Inertia</td>
<td>Specify inertia and initial velocity</td>
<td></td>
</tr>
<tr>
<td>Tire inertia</td>
<td>1</td>
<td>kg·m²</td>
</tr>
<tr>
<td>Initial velocity</td>
<td>0</td>
<td>rad/s</td>
</tr>
<tr>
<td>Rolling resistance</td>
<td>On</td>
<td></td>
</tr>
<tr>
<td>Resistance model</td>
<td>Constant coefficient</td>
<td></td>
</tr>
<tr>
<td>Constant coefficient</td>
<td>0.015</td>
<td></td>
</tr>
<tr>
<td>Velocity threshold</td>
<td>1e-3</td>
<td>m/s</td>
</tr>
</tbody>
</table>
3.2 4WD vehicle with different central differential

Figure 3-5: The model of 4wd vehicle

3.2.1 Open central differential

Figure 3-6: Model layout of open differential
**Sun-Planet Bevel**

The Sun-Planet Bevel gear block represents a set of carrier, planet, and sun gear wheels. The planet is connected to and rotates with respect to the carrier. The planet and sun corotate with a fixed gear ratio. A sun-planet and a ring-planet gear are basic elements of a planetary gear set.

Sun-Planet Bevel imposes one kinematic and one geometric constraint on the three connected axes.

\[
\mathbf{r}_c = \mathbf{r}_s \pm \mathbf{r}_p \\
\mathbf{r}_c \omega_c = \mathbf{r}_s \omega_s = \pm \mathbf{r}_p \omega_p
\]

\( \mathbf{r}_c \) is the radius of the carrier gear.  
\( \omega_c \) is the angular velocity of the carrier gear.  
\( \mathbf{r}_s \) is the radius of the sun gear.  
\( \omega_s \) is the angular velocity of the sun gear.  
\( \mathbf{r}_p \) is the radius of the planet gear.  
\( \omega_p \) is the angular velocity of the planet gear

The planet-sun gear ratio is defined as

\[
g_{ps} = \frac{r_p}{r_s} = \frac{N_p}{N_s}
\]

The torque transfer is defined as

\[
t_p = t_{loss} - g_{PS} \tau_s
\]

\( t_{loss} \) is the torque loss.  
\( \tau_s \) is the torque for the sun gear.  
\( \tau_p \) is the torque for the planet gear.

### Ideal open center differential

**TBR=1**  
**\( \eta_t = 100\% \)**

**Settings**

<table>
<thead>
<tr>
<th>Crown gear located:</th>
<th>To the right of centerline</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carrier (C) to driveshaft (D) teeth ratio (NC/ND):</td>
<td>1.33</td>
</tr>
<tr>
<td>Friction model:</td>
<td>No meshing losses - Suitable for HIL simulation</td>
</tr>
</tbody>
</table>

30
Real open center differential

TBR=1.2  \( \eta_i=83\% \)

<table>
<thead>
<tr>
<th>Crown gear located:</th>
<th>To the right of centerline</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carrier (C) to driveshaft (D) teeth ratio (NC/ND):</td>
<td>1.33</td>
</tr>
<tr>
<td>Friction model:</td>
<td>Constant efficiency</td>
</tr>
<tr>
<td>Sun-sun and carrier-driveshaft ordinary efficiencies:</td>
<td>[ 0.83 0.92 ]</td>
</tr>
<tr>
<td>Sun-carrier and driveshaft-casing power thresholds:</td>
<td>[ 0.001 0.001 ] W</td>
</tr>
</tbody>
</table>

3.2.2 Self-locking central ZF differential

![Figure 3-7: Model layout of Self-locking central ZF differential](image)

The limited slip differential is modeled using components from the Gears library and Clutches library in Simscape Driveline. Wheel slip is limited by clutches that engage when the torque applied to the input of the differential exceeds a threshold. The clutches lock the differential so that the output shafts of the differential spin at the same speed.

The spring-damper connecting the sun-planet-bevel gears introduces a small amount of compliance to allow for when both clutches are locked at the same time.
Torque to pressure

![Diagram of Torque to pressure](image)

**Figure 3-8: The model of Torque to pressure**

This subsystem is modeled as the effect of diagram preload spring. The block C represents preload pressure, the block PS gain represents the pressure of input torque. For the value of Pressure per Unit Torque (Pa/Nm), I set it in 40. And the value of Preload pressure of spring is 500.

Lag

![Filter Time Constant and Initial Output](image)

The block Lag filter of form \(1/(b*s+1)\) where \(s\) is the Laplace operator.

Block description

**Torque sensor** The Ideal Torque Sensor block represents a device that converts a variable passing through the sensor into a control signal proportional to the torque. The sensor is ideal because it does not account for inertia, friction, delays, energy consumption.
AWD/4WD transfer case for active torque split between the axles

Ports T—Torque N\(\times\)m. Physical signal output port for torque.

Ports R—Rod (positive probe). Mechanical rotational conserving port associated with the sensor positive probe.

Ports C—case (reference probe). Mechanical rotational conserving port associated with the sensor negative (reference) probe.

Disk Friction Clutch

This block represents a friction clutch with two flat friction plate sets that come into contact to engage. The clutch engages when the applied plate pressure exceeds an engagement threshold pressure. Once engaged, the plates experience frictional torques that enable them to transmit power between the base and follower driveshafts.

The block defines the slip velocity as the Relative angular velocity:

\[ \omega = \omega_F - \omega_B. \quad 3-4 \]

The kinetic friction torque is the positive sum of viscous drag and surface contact friction torques:

\[ \tau_K = \mu \omega + \tau_{contact} \quad 3-5 \]

The contact friction torque:

\[ \tau_{contact} = k_K \cdot D \cdot N \cdot r_{eff} \cdot P_{fric} \cdot A \geq 0. \quad 3-6 \]

kinetic friction coefficient \(k_K\) is a constant.

The clutch applies a normal force from its piston as the product of the clutch friction capacity \(P_{fric}\) and engagement surface area \(A\) on each of \(N\) friction surfaces. The pressure signal \(P\) should be nonnegative. If \(P\) is less than \(P_{th}\), the clutch applies no friction at all.

The effective torque radius \(r_{eff}\):

\[ r_{eff} = \frac{2}{3} \frac{r_o^3 - r_i^3}{r_o^2 - r_i^2} \quad 3-7 \]

\(ro\) and \(ri\) are the outer and inner radius of the friction surface.

The clutch de-rating factor \(D\):
AWD/4WD transfer case for active torque split between the axles

\[ D = \frac{3}{4} \left( \frac{r_o + r_i}{r_o^2 + r_o r_i + r_i^2} \right) \]

3-8

**Settings**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Follower (F) to base (B) teeth ratio (NF/NB):</td>
<td>1.33</td>
</tr>
<tr>
<td>Output shaft rotates:</td>
<td>In same direction as input shaft</td>
</tr>
<tr>
<td>Friction model:</td>
<td>Constant efficiency</td>
</tr>
<tr>
<td>Efficiency:</td>
<td>0.85</td>
</tr>
<tr>
<td>Follower power threshold:</td>
<td>0.001 W</td>
</tr>
<tr>
<td>Viscous friction coefficients at base (B) and follower (F):</td>
<td>[ 0 0 ] Nm/(rad/s)</td>
</tr>
</tbody>
</table>

**Figure 3-9:** The parameters of Input Pinion Crown Gear

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Force action region:</td>
<td>Define annular region</td>
</tr>
<tr>
<td>Friction disk outside diameter:</td>
<td>300 mm</td>
</tr>
<tr>
<td>Friction disk inside diameter:</td>
<td>100 mm</td>
</tr>
<tr>
<td>Number of friction surfaces:</td>
<td>6</td>
</tr>
<tr>
<td>Engagement piston area:</td>
<td>0.04 m²</td>
</tr>
<tr>
<td>Directionality:</td>
<td>Bidirectional</td>
</tr>
<tr>
<td>Friction model:</td>
<td>Fixed kinetic friction coefficient</td>
</tr>
<tr>
<td>Kinetic friction coefficient:</td>
<td>0.2</td>
</tr>
<tr>
<td>Static friction coefficient:</td>
<td>0.25</td>
</tr>
<tr>
<td>De-rating factor:</td>
<td>1</td>
</tr>
<tr>
<td>Clutch velocity tolerance:</td>
<td>0.01 rad/s</td>
</tr>
<tr>
<td>Engagement threshold pressure:</td>
<td>100 Pa</td>
</tr>
<tr>
<td>Viscous drag torque coefficient:</td>
<td>0.0032 Nm/(rad/s)</td>
</tr>
</tbody>
</table>

**Figure 3-10:** The parameters of Clutch pack
AWD/4WD transfer case for active torque split between the axles

| Planet (P) to sun (S) teeth ratio (NP/NS): | 2 |
| Assembly orientation: | Right – Sun and planet gears rotate in opposite directions |
| Friction model: | Constant efficiency |
| Ordinary efficiency: | 0.92 |
| Sun-carrier power threshold: | 0.001 |

Figure 3-11: The parameters of Sun-Planet Bevel

3.2.3 Torsen center differential

The locking torque is due to the high friction existing between the worm gear planet wheels and satellites.

Figure 3-12: Torsen center differential

Figure 3-12: Torsen center differential
In the torsen differential subsystem, the input gear indicates the gear 1 in the figure 3-17 and the sun-planet worm gear indicates the gearset consisted of components 3,4,5 in figure 3-17. And the port S1 is connected to the real axle while the port S2 is connected to the front axle.

**Sun-Planet Worm Gear**

The Sun-Planet Worm Gear and planet gears. The sun and planet gears are crossed helical spur gears arranged as a worm-gear transmission, in which the planet gear is a worm. Such transmissions are used in the Torsen type 1.
AWD/4WD transfer case for active torque split between the axles

differential. When transmitting power, the sun gear can be independently rotated by the worm (planet) gear, or by the carrier, or both.

Sun-planet worm gear imposes one kinematic constraint on the three connected axes:

$$\omega_S = \omega_p/R_{WG} + \omega_c$$  \hspace{1cm} 3-9

$R_{WG}$—Gear, or transmission, ratio determined as the ratio of the worm angular velocity to the gear angular velocity.

$\omega_S$—Angular velocity of the sun gear \hspace{1cm} $\omega_p$—planet(Worm) angular velocity

$\omega_c$—carrier angular velocity

The torque transfer is:

$$R_{WG}\tau_p + \tau_S - \tau_{Loss} = 0,$$  \hspace{0.5cm} 3-10

with $\tau_{loss} = 0$ in the ideal case.

$\tau_p$—Torque applied to the planet shaft \hspace{1cm} $\tau_s$—Torque applied to the sun shaft

$\tau_c$—Torque applied to the carrier shaft \hspace{1cm} $\tau_{loss}$—Torque loss due to meshing friction.

The efficiency of Sun-Planet Worm Gear in direct and inverse way:

$$\eta_{WG} = (\cos\alpha - k\tan\lambda)/(\cos\alpha + k\tan\lambda)$$  \hspace{1cm} 3-11

$$\eta_{GW} = (\cos\alpha - k\tan\lambda)/(\cos\alpha + k^*\tan\alpha)$$  \hspace{1cm} 3-12

$\alpha$—Normal pressure angle; \hspace{0.5cm} $\lambda$—Worm lead angle; \hspace{0.5cm} $k$—Friction coefficient;

If you set efficiency for the reverse power flow to a negative value, the differential exhibits self-locking. Power cannot be transmitted from sun gear to worm and from carrier to worm unless some torque is applied to the worm to release the train. In this case, the absolute value of the efficiency specifies the ratio at which the train is released. The smaller the train lead angle, the smaller the reverse efficiency.

**Settings**

Input gear ratio F/B=2
I set the efficiency of Sun-Planet Worm Gears in 0.5.

<table>
<thead>
<tr>
<th>Main</th>
<th>Meshing Losses</th>
<th>Viscous Losses</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gear ratio:</td>
<td>3.46</td>
<td></td>
</tr>
<tr>
<td>Worm thread type:</td>
<td>Right-hand</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Friction model:</th>
<th>Constant efficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td>Efficiency:</td>
<td>0.5</td>
</tr>
<tr>
<td>Follower power threshold:</td>
<td>0.001</td>
</tr>
</tbody>
</table>

Worm-carrier and sun-carrier viscous friction coefficients: [0 0] \(\text{Nm/(rad/s)}\)

**Figure 3-15: Parameters of Sun-Planet Worm Gear**

### 3.3 The model differential Test

Before investigating that the differential works as a center coupling in AWD vehicle, I tend to test the differential working in an ideal condition. So I add a constant input torque instead of powertrain system.

This model is to simulate the differential works in ideal condition. In this part, I assume the input torque grow in linear trend. In this simulation, we test several differentials to figure out the characteristics curve of differentials by changing
the input torque. In this passage, I mainly consider the locking coefficient $b$ and Torque Bias Ratio (TBR).

We define as locking coefficient $b$: the maximum difference of torque (referred to the total axle torque) between the two output shafts that can be sustained with no difference in speed; the torque difference is called locking torque. The locking coefficient $b$ can be constant or can depend upon speed difference or input torque values ($b=0$ for open differential).

TBR means ratio between highest and lowest torque.

**Limited-slip differential**

For the limited-slip differential, I vary the input torque to test the locking coefficient $b$ and the TBR of differential. At first, I set the speed difference between front and rear axles: 20 rad/s. I keep internal efficiency of front and rear sun-planet bevel gears in a constant value. I also set the preload pressure in 500 pa, I change the pressure per unit torque in 3 values: 20 Pa/Nm (LSD1), 40 Pa/Nm (LSD2), 60 Pa/Nm (LSD3).

In this way, I can get the diagram of $b$ and TBR for the limited-slip differential with different $\eta$.

![Figure 3-17: Locking coefficient versus input torque for Limited-slip differential](image)
AWD/4WD transfer case for active torque split between the axles

From this graph, we can find the locking coefficient $b$ reduce linear with input torque first, and it will keep constant. And the $b$ increase as the pressure per unit torque reduces at same input torque.

Besides the $b$ and TBR, I also test the torque difference and rotational speed differences between front and rear axles for the limited-slip differential.
Through testing the LSD2 differential at the input torque=200 Nm, I can get the graph as follow.

**Figure 3-20: $\Delta T$ VS $\Delta \omega$ for Limited-slip differential**

**Torque split ratio**

To test the torque split ratio between front and rear axles, I set speed difference $\Delta \omega=20$rad/s and the input torque=200Nm. In this way, I test the differential with different efficiency of front

LSD1: $i=T_F:T_R=38.5:61.5$
LSD2: $i=T_F:T_R=28:72$
LSD3: $i=T_F:T_R=19:80$

**Open center differential**

For the open center differential, I vary the rear sun gear-front sun gear driveshaft ordinary efficiencies and the carrier-driveshaft ordinary efficiencies. And I set the speed difference between front and rear axles:20rad/s. I assume the differential in 3 conditions:

Ideal: $\eta_{sun-sun}=100\%$ $\eta_{carrier-driveshaft}=100\%$
Real 1: $\eta_{sun-sun}=83\%$ $\eta_{carrier-driveshaft}=92\%$
Real 1: $\eta_{sun-sun}=74\%$ $\eta_{carrier-driveshaft}=92\%$
AWD/4WD transfer case for active torque split between the axles

Figure 3-21: Locking coefficient versus input torque for open differential

From the graph, I can find that the locking coefficient nearly keep constant as input torque raises for the fixed $\eta$. If I set a constant input torque, the $b$ increases as $\eta$ reduces. In the simulation, the $b$ of ideal differential equal to 0.

Figure 3-22: TBR versus input torque for open differential

Besides the $b$ and TBR, I also test the torque difference and rotational speed differences between front and rear axles of open differential. Through testing the
AWD/4WD transfer case for active torque split between the axles

differential in the real 1 condition at input torque=200 Nm, I can get the graph as follow.

![Graph](image)

Figure 3-23: $\Delta T$ VS $\Delta \omega$ for open differential

From the graph, I got:

- $\Delta \omega > 0$  $\Delta T=T_F - T_R=-23$Nm
- $\Delta \omega = 0$  $\Delta T=T_F - T_R=0$
- $\Delta \omega < 0$  $\Delta T=T_F - T_R=23$Nm

Torque split ratio

To test the torque split ratio between front and rear axles, I set the input torque=200Nm. In this way, I test the differential with different rear-front driveshaft ordinary efficiencies. ( Ideal $\eta=100\%$, real1$\eta=83\%$ real2$\eta=74\%$).

Ideal differential:  $i=T_F:T_R=50:50$;  
real1  $i=T_F:T_R=45:55$  
real2  $i=T_F:T_R=42:58$

Torsen center differential

For the torsen differential, I set the efficiencies of input gears, rear and Sun-Planet Worm Gears in a constant value: $\eta_{input}=95\%$, $\eta_{front}=100\%$, $\eta_{rear}=100\%$. I also set the speed difference between front and rear axles in 20rad/s. And I will vary the efficiencies of the worm-gear connected to the front and rear axles. In this way, I assume 3 differentials: TCD1($\eta_{worm}=50\%$); TCD2($\eta_{worm}=40\%$);
TCD3(\eta_{worm}=30\%). And I have tied set a negative value of worm-gear efficiency. In this situation, the differential is locked and the simscape model can't run.

Figure 3-24: Locking coefficient versus input torque for torsen differential

From this graph, we can find the locking coefficient almost keep constant with increasing of input torque. And locking coefficient of the ideal torsen differential equal to 0. For real differential, the locking coefficient reduces as \eta increases.

Figure 3-25: TBR versus input torque for torsen differential
In this graph, the relationship between TBR and input torque is similar to the Limited-slip differential. For the ideal differential, the TBR equal to 0. For real situation, the TBR reduces as difference between $\eta_{front}$ and $\eta_{rear}$ increases.

Besides the b and TBR, I also test the torque difference and rotational speed differences between front and rear axles of torsen differential. Through testing the TCD1 differential in the input torque=200 Nm, I can get the graph as follow

![Graph](image)

**Figure 3-26: $\Delta T$ VS $\Delta \omega$ for Torsen differential**

**Torque split ratio**

To test the torque split ratio between front and rear axles, I set the input torque=200Nm. In this way, I test the differential in different condition

TCD1: $i=\frac{T_F}{T_R}=33.4:66.6$

TCD2: $i=\frac{T_F}{T_R}=28.7:71.3$

TCD3: $i=\frac{T_F}{T_R}=23.1:76.9$
3.4 Four-wheel drive vehicle with Viscous coupling

The block represents a nonlinear rotational damper. Polynomial and table lookup parameterizations define the nonlinear relationship between damping torque and relative angular velocity. The damping torque can be symmetric or asymmetric about the 0 velocity point. The block applies equal and opposite damping torques on the two rotational conserving ports.

The symmetric polynomial parameterization defines the damping torque for both positive and negative relative velocities according to the expression:

\[ T = b_1 \omega + \text{sign}(\omega) \cdot b_2 \omega^2 + b_3 \omega^3 + \text{sign}(\omega) \cdot b_4 \omega^4 + b_5 \omega^5, \]

3-19

\( T \) — Damping torque
\( b_1, b_2, \ldots, b_5 \) — Damping coefficients
\( \omega \) — Relative angular velocity between ports R and C, \( \omega = \omega_R - \omega_C \)
\( \omega_R \) — Absolute angular velocity associated with port R
\( \omega_C \) — Absolute angular velocity associated with port C
Viscous coupling testing

Figure 3-28: Model layout of Viscous coupling testing

This model is built for plotting the output torque versus $\Delta\omega$. In this model, I set the input torque as 200Nm. I vary the $\Delta\omega$ from -60 to 60 rad/s. In this way, I get the chart as follow.

Figure 3-29: Output torque VS $\Delta\omega$ for Viscous coupling
### 4. SIMULATION RESULTS

#### 4.1 Working Condition

##### 4.1.1 Parameters

<table>
<thead>
<tr>
<th></th>
<th>OPDI</th>
<th>OPDR</th>
<th>TCD</th>
<th>Viscous</th>
<th>FWD</th>
</tr>
</thead>
<tbody>
<tr>
<td>Central Differential Pinion/Crown ratio and efficiency</td>
<td>1.33/100%</td>
<td>1.33/83%</td>
<td>1.33/100%</td>
<td>--</td>
<td>--</td>
</tr>
<tr>
<td>Torsen Differential Sun-Planet Worm Gear and efficiency</td>
<td>--</td>
<td>--</td>
<td>3.46/ $\eta_{WG}=0.84$ $\eta_{GW}=0.5$</td>
<td>--</td>
<td>--</td>
</tr>
<tr>
<td>Front Differential Pinion/Crown ratio and efficiency</td>
<td>1.33/100%</td>
<td>1.33/100%</td>
<td>1.33/100%</td>
<td>1.33/100%</td>
<td>1.33/100%</td>
</tr>
<tr>
<td>Rear Differential Pinion/Crown ratio and efficiency</td>
<td>1.33/100%</td>
<td>1.33/100%</td>
<td>1.33/100%</td>
<td>1.33/100%</td>
<td>--</td>
</tr>
<tr>
<td>TBR</td>
<td>1</td>
<td>1.2</td>
<td>2.5</td>
<td>--</td>
<td>--</td>
</tr>
<tr>
<td>b</td>
<td>0</td>
<td>0.09</td>
<td>0.43</td>
<td>--</td>
<td>--</td>
</tr>
</tbody>
</table>
OPDI—ideal open differential; OPDR—real open differential; 
TCD—torsen differential; Viscous—Viscous coupling; FWD—front wheel drive

\( \eta_{WG}, \eta_{GW} \)--The efficiency of Sun-Planet Worm Gear in direct and inverse way (Figure 3-14)

4.1.2 Road surface.

In this test, there are 4 types of surface: ice, snow, wet, dry. In the model, we use magic formula tire. So the input parameters of surfaces are Physical signal Magic Formula coefficients: B- Rated vertical load, C- Peak longitudinal force at rated load, D E- Slip at peak force at rated load (percent).

<table>
<thead>
<tr>
<th></th>
<th>B</th>
<th>C</th>
<th>D</th>
<th>E</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dry</td>
<td>10</td>
<td>1.3</td>
<td>1</td>
<td>0.97</td>
</tr>
<tr>
<td>Wet</td>
<td>12</td>
<td>2.3</td>
<td>0.82</td>
<td>1</td>
</tr>
<tr>
<td>Snow</td>
<td>5</td>
<td>2</td>
<td>0.3</td>
<td>1</td>
</tr>
<tr>
<td>Ice</td>
<td>4</td>
<td>2</td>
<td>0.1</td>
<td>1</td>
</tr>
</tbody>
</table>

Through the equation:

\[
\mu = D \sin(\arctan(Bx - E[Bx - \arctan Bx]))
\]

I can calculate the friction coefficient. The x is the longitudinal slip. And I assume the x ∈ [-1, 1] in this calculation. In this way, I obtain the graph about friction coefficient of each surface.

Figure 4-1: \( \mu \) of different surface in different slip
From this figure, I got the max friction coefficient of each surface.

<table>
<thead>
<tr>
<th></th>
<th>ice</th>
<th>snow</th>
<th>wet</th>
<th>dry</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\mu_{\text{max}}$</td>
<td>0.1</td>
<td>0.3</td>
<td>0.82</td>
<td>0.97</td>
</tr>
</tbody>
</table>

Table 4-1

Figure 4-2: schematic diagram of Simulink test

In the graph, the axle $x$ represents longitudinal position in the vehicle trace. In the test, I assume the trace is long enough to test the characteristics of transfer case. In the next, I set the friction coefficient from low to high (surfaces ice→dry) in the trace.
4.2 Results

4.2.1 Engine Performances

![Figure 4-3: Engine torque of different transfer case](image)

![Figure 4-4: Engine speed of different transfer case](image)
From these 2 graphs, we can find the engine performances of each transfer case is different. And the trend of engine torque is opposite to the trend of engine speed. So the power of each transfer case are similar. The differences of engine torque and speed between the transfer cases are mainly caused by the different transfer efficiencies of coupling devices connected the front and rear axles.

**Torque converter**

The solid line represents torque ratio and the dash line represents the speed ratio.

![Figure 4-5: Torque/speed ratio between input and output axle](image)

From the graph, we can find the torque ratio and speed ratio are symmetric with respect to axle x=1.
Input torque (only for differentials)

![Input torque graph](image1)

Figure 4-6: Input torque of differentials

4.2.2 Performance of coupling devices

Open differential ($\eta=100\%$)

![Rotational speed graph](image2)

Figure 4-7: Rotational speed front and rear axles
For the ideal open differential, the torque split ratio is 50:50. And no speed difference between the front and rear axles occurs in the whole process.

Figure 4-8: Output torque of front and rear axles

Figure 4-9: Slips of front and rear wheels for ideal open differential
Open differential ($\eta = 83\%$)

Figure 4-10: rotational speed of front and rear axles

Figure 4-11: Partial enlargement of Figure 4-9
AWD/4WD transfer case for active torque split between the axles

Figure 4-12: output torque of front and rear axles

For the real open differential, there are speed difference and torque difference in the process. It’s mainly caused by the meshing losses of gearset in differential.

Figure 4-13: Slips of front and rear wheels for real open differential
Torsen differential

Figure 4-14: Rotational speed of front and rear axles

Figure 4-15: Output torque of front and rear axles
For the torsen differential, I set efficiencies of Sun-Planet Worm Gear in 50%. So it causes a big difference between the front and rear torque which lead to a speed difference.

Viscous coupling
AWD/4WD transfer case for active torque split between the axles

Figure 4-18: Output torque of front and rear axles

Figure 4-19: Slips of front and rear wheels for viscous coupling
Tire performances

Figure 4-20: Slips of front wheels

Figure 4-21: Slips of rear wheels
4.2.3 VEHICLE PERFORMANCE

Figure 4-22: The vehicle accelerations of different transfer cases

Figure 4-23: The vehicle speeds of different transfer cases
CONCLUSION

According to the figure of front and rear slips, I find that the transfer case with ideal open differential causes a highest rear wheel slip in the moment of road surface changing while viscous coupling lead to a highest front wheel slip. The torsen differential and real open differential have a positive effect on limiting slip.

For the slip difference between the front and rear wheels, the vehicle with limited-slip differential (torsen) has a better performance in eliminating the slip difference when vehicle move into the road surface with a higher friction coefficient. For the torsen differential, there is no slip difference in the whole process. Besides it, the real open differential (TBR=1.2) also has effect on eliminating the slip differences although it needs a long time to work. For the ideal open differential (TBR=1) and viscous coupling, the slip differences between the front and rear wheels exit in the whole process. In a word, the real open differentials and torsen differentials have a beneficial effect on the vehicle mobility.

In this thesis, I mainly study on the mechanical way for the torque split between the axles. In present, there are many transfer cases combine the self-locking differential or viscous clutch with the ECU which can make vehicle has a better performance in mobility. This trend represents the direction of the development of the AWD/4WD vehicle in the future.
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