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.

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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 8edactor (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)

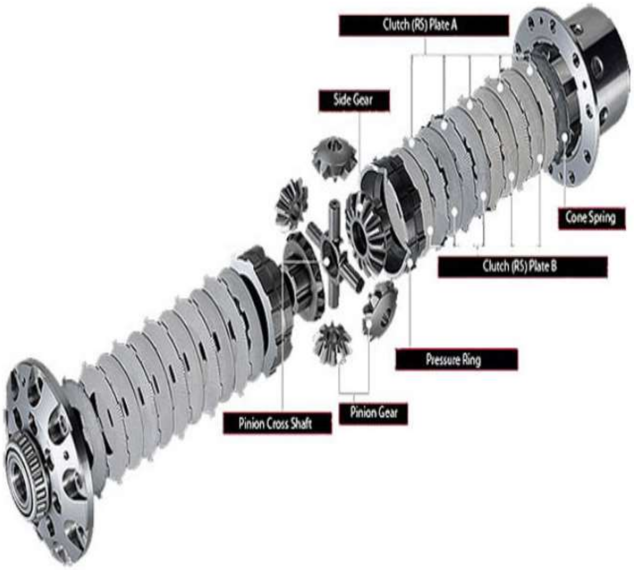


Figure 1-1: ZF differential[1]

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.

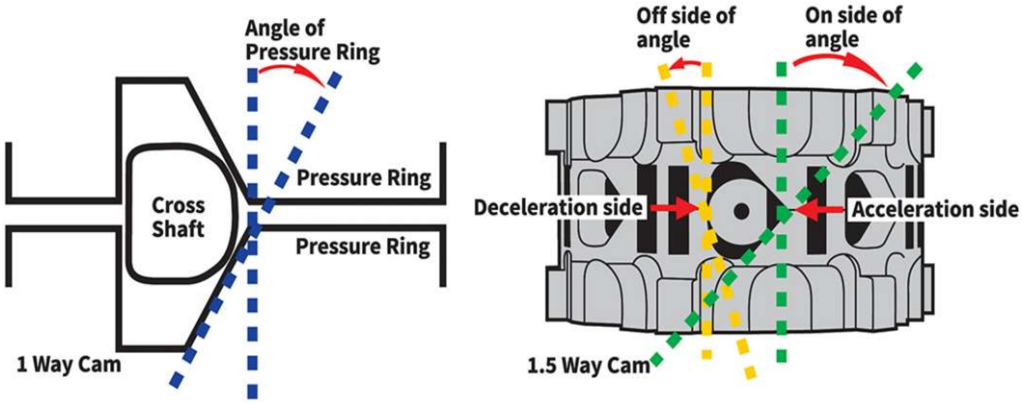


Figure 1-2: schematic diagram of ZF differential[2]

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.

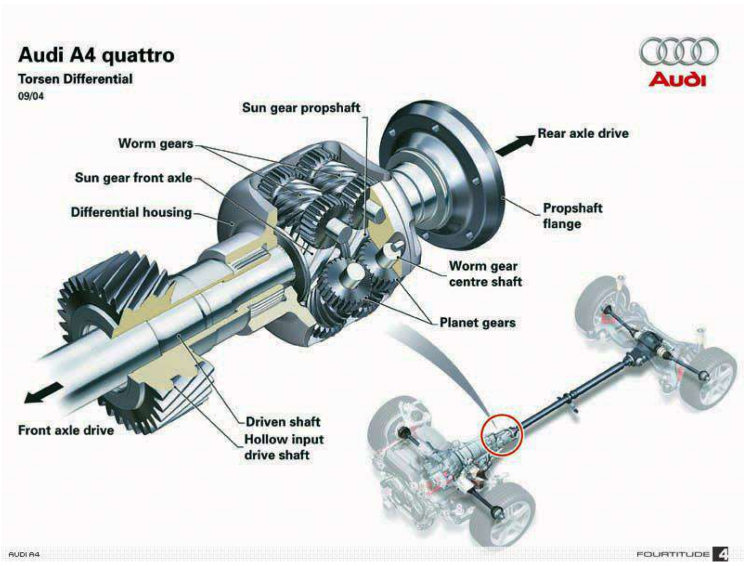


Figure 1-3: Torsen differential[3]

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.

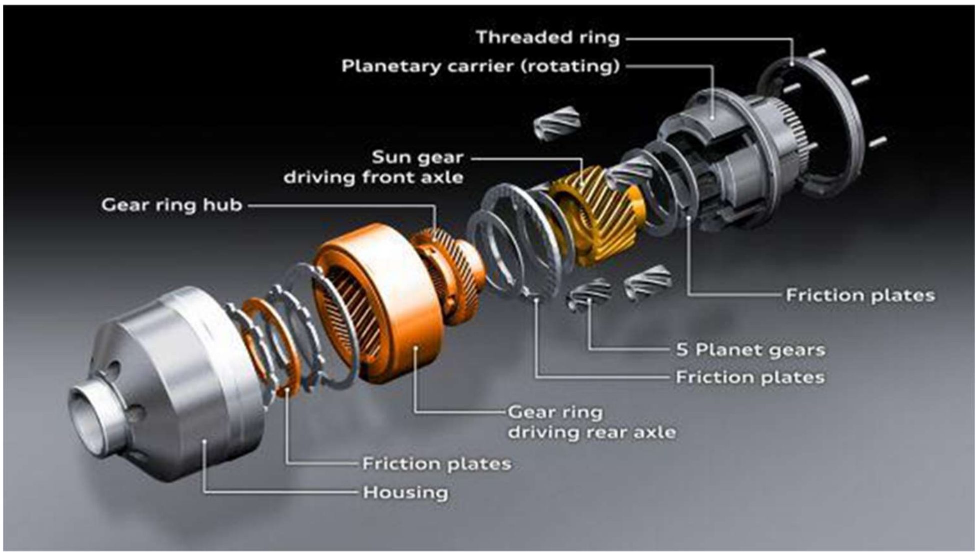


Figure 1-4: Torsen C differential[4]

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

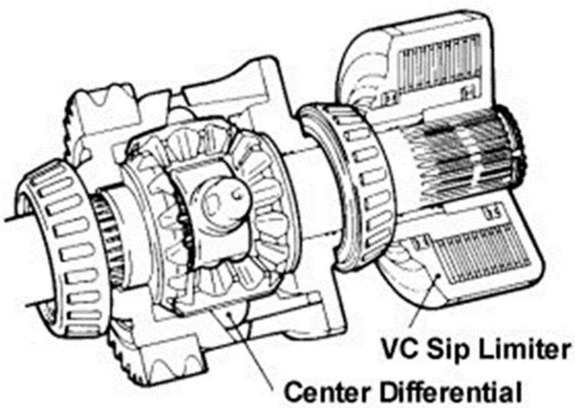


Figure 1-5: Viscous-Coupling Differential

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.[5] 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)

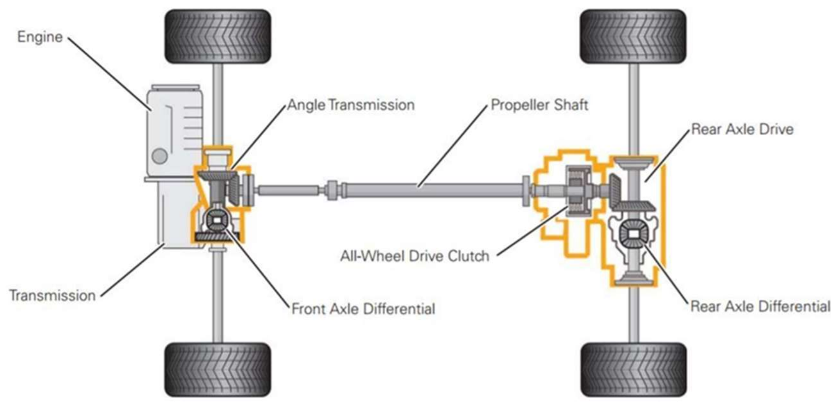


Figure1-6: driveline of Audi TT quattro[6]

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.

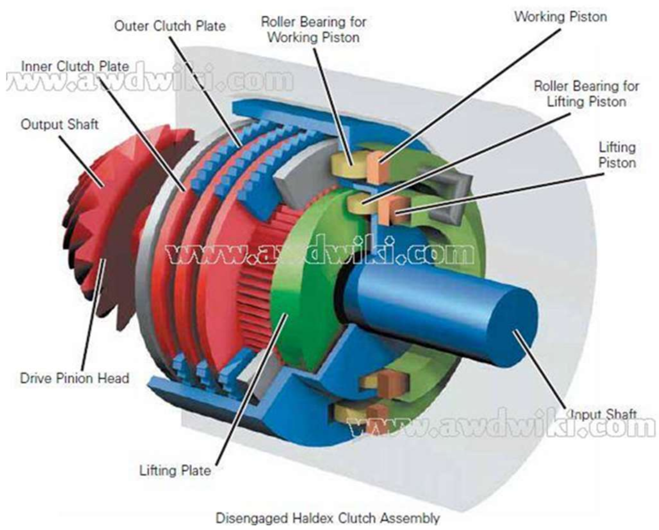


Figure 1-7: Haldex clutch

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.

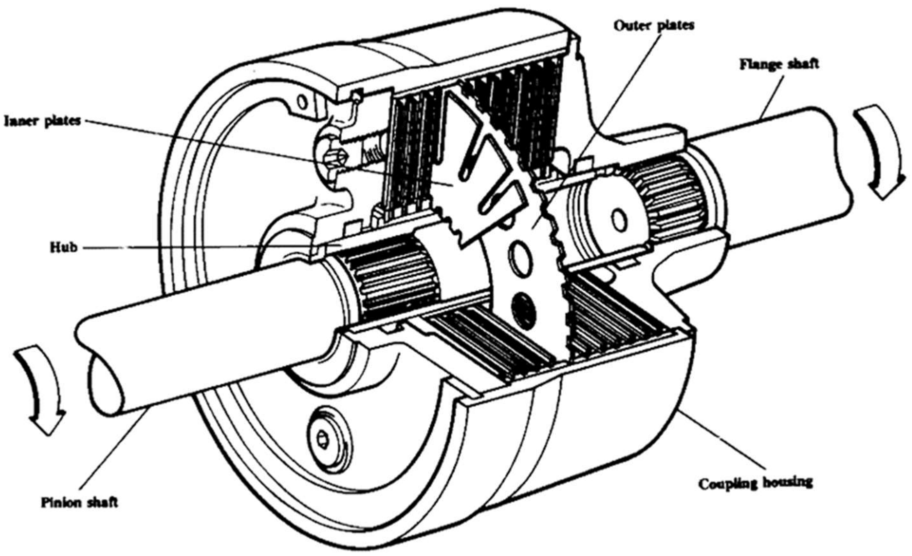


Figure 1-8: Viscous clutch

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.

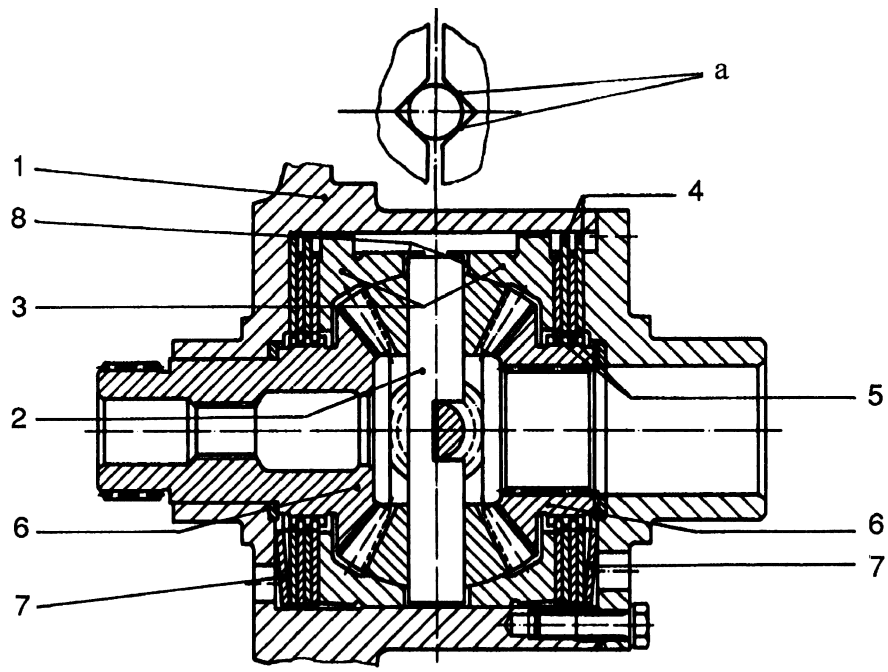
2.The layout and analysis of transfer case

In this part, the paper mainly illustrates the transfer cases for each mechanical solution of toque 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.



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

Figure 2-1: ZF differential[7]

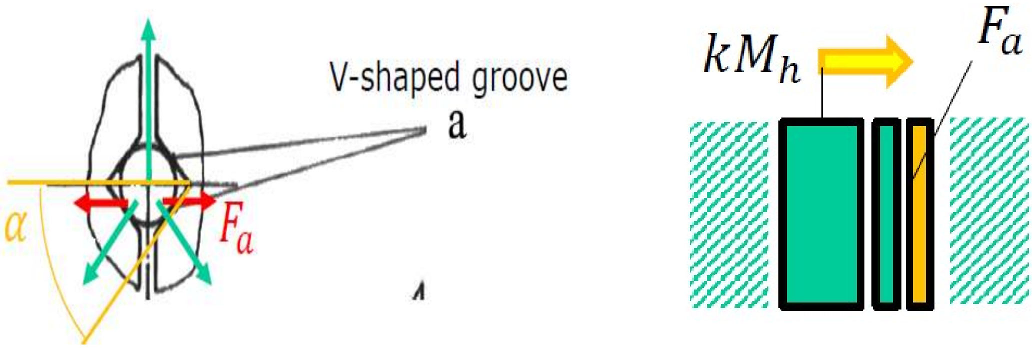


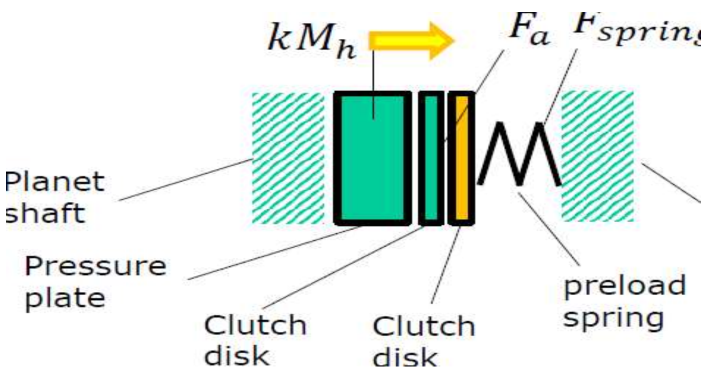
Figure 2-2: schematic diagram of upper part

Discs pressure force is about proportional to the input torque.

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

$$C_F = N_{cl}\, R_{cl}\mu\, F_a = N_{cl}\, R_{cl}\mu M_h / R_p \tan\alpha \qquad 2\text{-}1$$

$$F_a = \mathrm{k} M_h \quad \Delta M = 2C_F \qquad 2\text{-}2$$



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

Figure 2-3: schematic diagram of Lower part

$$C_F = N_{cl}\, R_{cl}\mu\, F_a \qquad 2\text{-}3$$

$$\text{If } M_h < M_h^*(F_{spring}),\ F_a = F_{spring} \qquad 2\text{-}4$$

$$\text{If } M_h > M_h^*(F_{spring}),\ F_a = \frac{N_{cl}\, R_{cl}\mu}{R_p \tan\alpha} * M_h \qquad 2\text{-}5$$

2.1.2 Torsen C differential

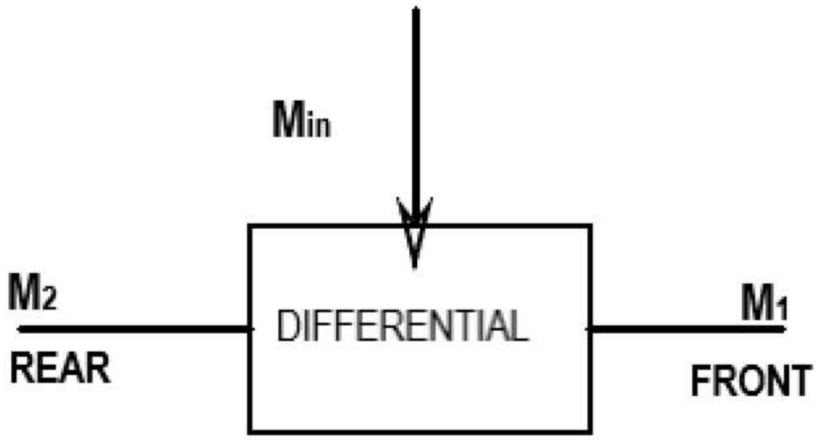


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Figure 2-4: schematic diagram of self-locking differential

Torque Bias Ratio (TBR):

$$\text{TBR}=\mathbf{M}_{max}/\mathbf{M}_{min} \qquad 2\text{-}6$$

locking coefficient b

$$b=\frac{\mathbf{M}_{max}-\mathbf{M}_{min}}{\mathbf{M}_{in}}\ ,|\Delta\mathbf{M}|=\mathbf{M}_{max}\text{-}\mathbf{M}_{min}\leq\mathbf{bM}_{in} \qquad 2\text{-}7$$

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

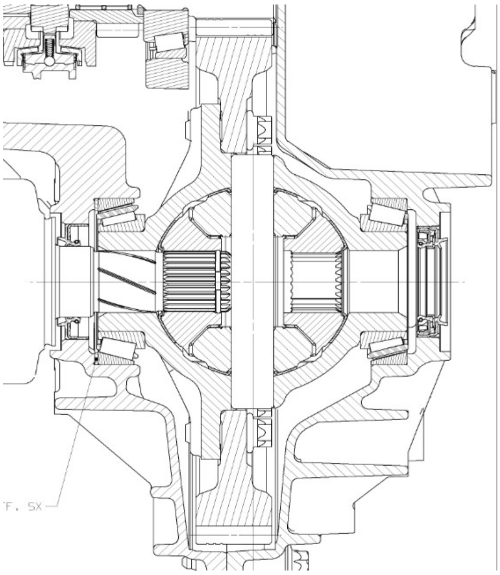
If $|\Delta M| = bM_{in}$, the differential is unlocked.

$$\mathbf{M}_F=\frac{\mathbf{M}_{in}}{2}+\frac{\mathbf{b}|\mathbf{M}_{in}|}{2}\mathbf{sin}(\boldsymbol{\omega}_R-\boldsymbol{\omega}_F) \qquad 2\text{-}8$$

$$\mathbf{M}_R=\frac{\mathbf{M}_{in}}{2}-\frac{\mathbf{b}|\mathbf{M}_{in}|}{2}\mathbf{sin}(\boldsymbol{\omega}_R-\boldsymbol{\omega}_F) \qquad 2\text{-}9$$

$$\boldsymbol{\omega}_{IN}=\frac{\boldsymbol{\omega}_F+\boldsymbol{\omega}_R}{2} \qquad 2\text{-}10$$

2.2 Open differential



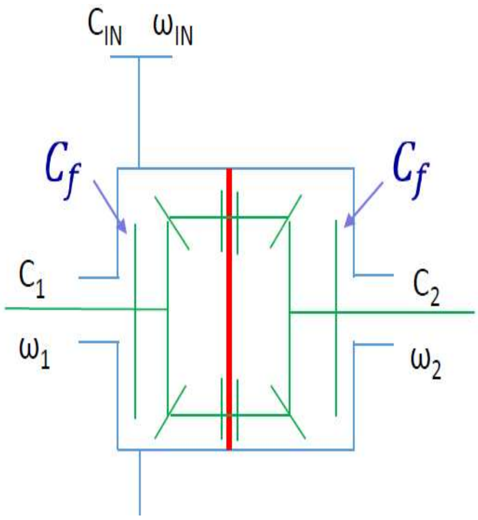


Figure 2-5: schematic diagram of open differential[8]

IN: Torque/ speed at ring gear

1: torque / speed at front shaft   2: torque / speed at rear shat

C_f—friction torque

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

$$C_1=\frac{C_{IN}}{2}\pm C_f \qquad C_2=\frac{C_{IN}}{2}\mp C_f \qquad 2\text{-}12$$

$$\omega_1>\omega_2:\ C_1=\frac{C_{IN}}{2}-C_f,C_2=\frac{C_{IN}}{2}+C_f \qquad 2\text{-}13$$

$$\omega_1<\omega_2:\ C_1=\frac{C_{IN}}{2}+C_f,C_2=\frac{C_{IN}}{2}-C_f \qquad 2\text{-}14$$

Ratio between highest and lowest torque

$$\text{TBR}=\frac{C_1}{C_2}=\frac{k_1C_{IN}}{k_2C_{IN}}=\text{const.} \qquad 2\text{-}15$$

2.3 Haldex clutch(Audi TT quattro)

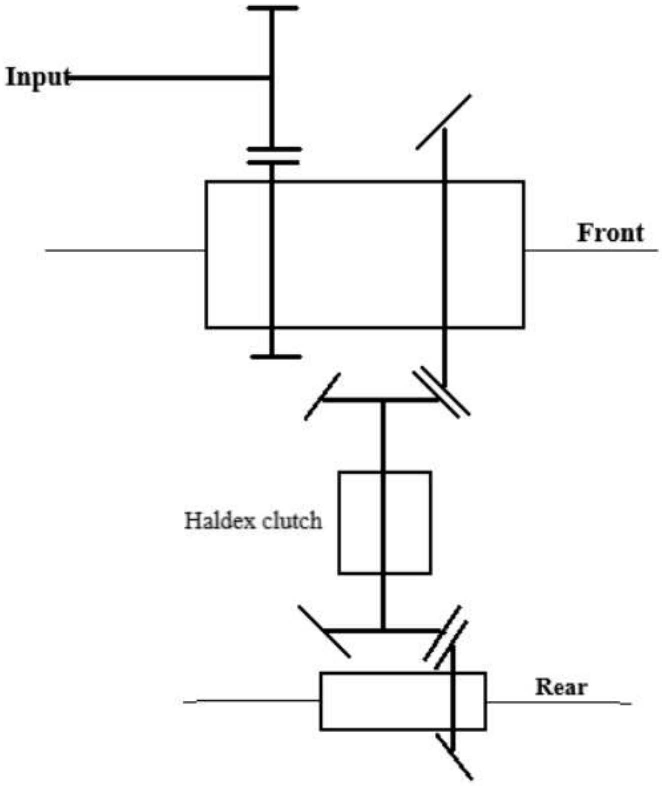


Figure2-6 stick diagram of Haldex AWD system

By applying the Newton law to the joint discs:

$$\boldsymbol{\tau} = \boldsymbol{\mu}\Delta\mathbf{V}/\mathbf{d} \qquad 2\text{-}16$$

τ= shear on disc surface element dS.

μ= dynamic viscosity coefficient.

ΔV=speed local difference.

D =gap between discs.

Force acting on the surface element:

$$\boldsymbol{\tau}\mathbf{dr^{*}r^{*}d\alpha r}=\boldsymbol{\tau}\mathbf{r^{2}drd\alpha} \qquad 2\text{-}17$$

Total braking torque of the joint:

$$\mathrm{b}M=\mathrm{n}(\boldsymbol{\pi\Delta\omega v\rho}/\mathbf{2d})\boldsymbol{r_e^4}[\mathbf{1}-(\boldsymbol{r_i}/\boldsymbol{r_e})]^{\mathbf{4}} \qquad 2\text{-}18$$

r_i, r_e=inside, outside radii of facing surfaces.

N=number of facing surfaces.

Δω=rotational speed difference.

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

2.4 Viscous coupling

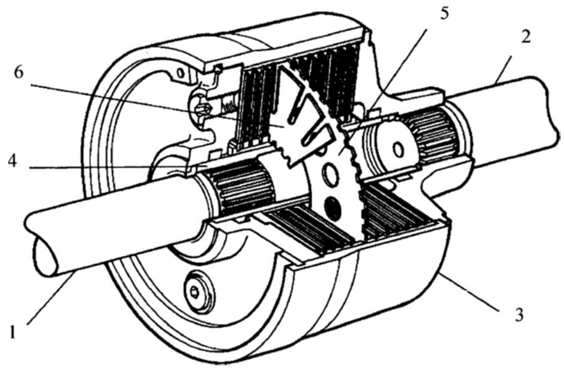


Figure 2-7: schematic diagram of Viscous coupling

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:

$$\mathrm{bM{=}n}(\boldsymbol{\pi}\Delta\boldsymbol{\omega}\boldsymbol{v}\boldsymbol{\rho}/\mathbf{2d})\boldsymbol{r}_e^4[\mathbf{1}-(\boldsymbol{r}_i/\boldsymbol{r}_e)]^4 \qquad \text{2-19}$$

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:

$$\mathbf{T}_{\boldsymbol{b}}\mathrm{=K^*}\Delta\boldsymbol{\omega} \qquad \Delta\boldsymbol{\omega}=|\boldsymbol{\omega}_{\boldsymbol{F}}-\boldsymbol{\omega}_{\boldsymbol{R}}| \qquad \text{2-20}$$

So we can add a nonlinear damper between front and rear shaft to express this equation.

2.5 Torque Vectoring Transfer Case

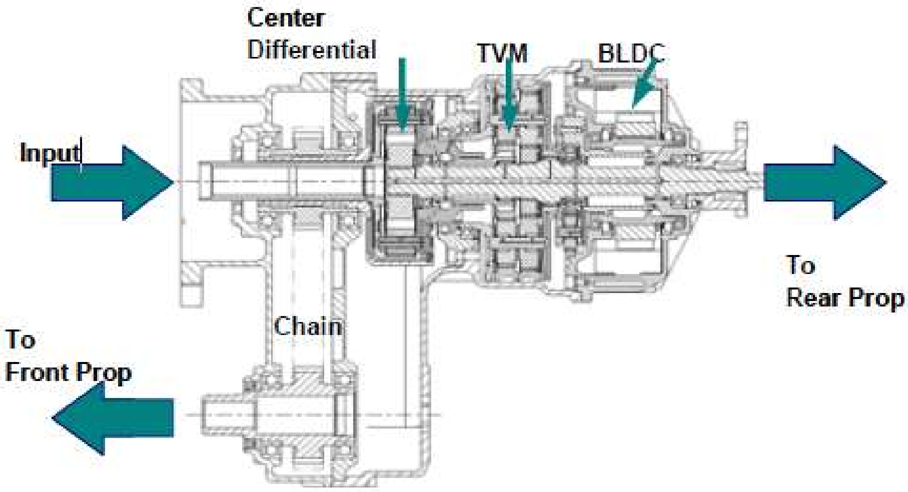


Figure 2-8: Layout of TV transfer case[9]

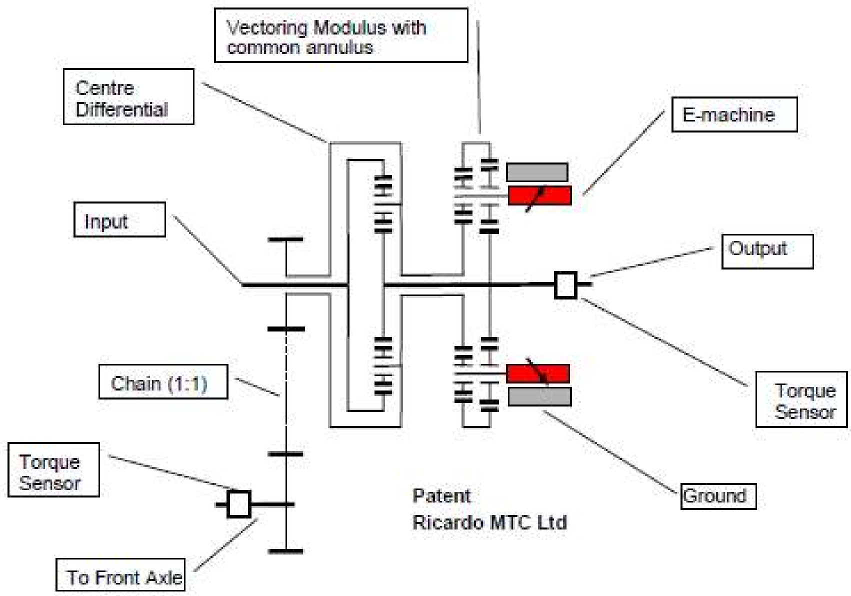


Figure 2-9: stick diagram of TV transfer case

$$T_F = T_{engine} * R_{Transmission} * (1 - R_{CD}) - T_{Actuator} * TBR \qquad 2\text{-}21$$

$$T_R = T_{engine} * R_{Transmission} * (R_{CD}) - T_{Actuator} * TBR \qquad 2\text{-}22$$

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.

$\mathbf{T}_{Actuator}$-Actuator torque. Positive when biasing to the rear axle.

2.6 Chain drive transfer case

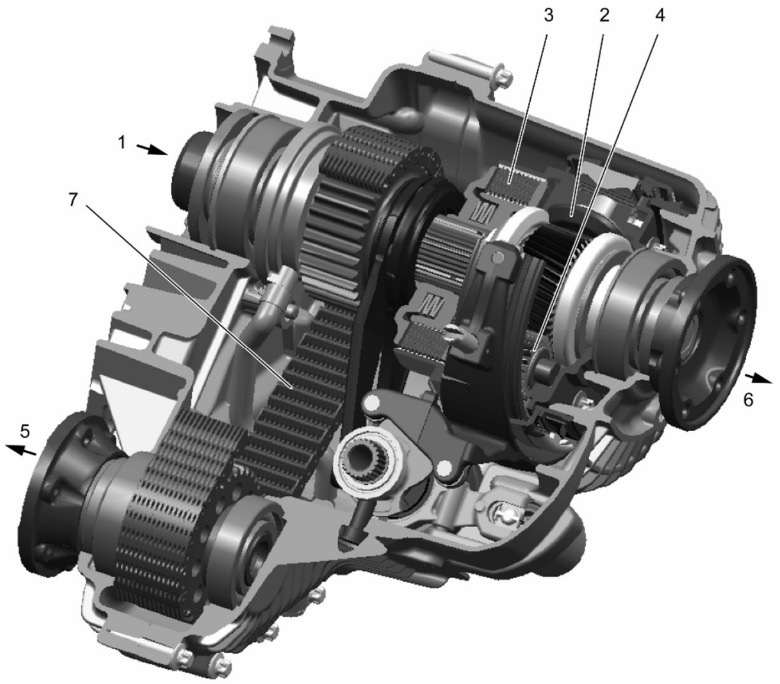


Figure 2-10: Chain drive transfer case[10]

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 toque 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.

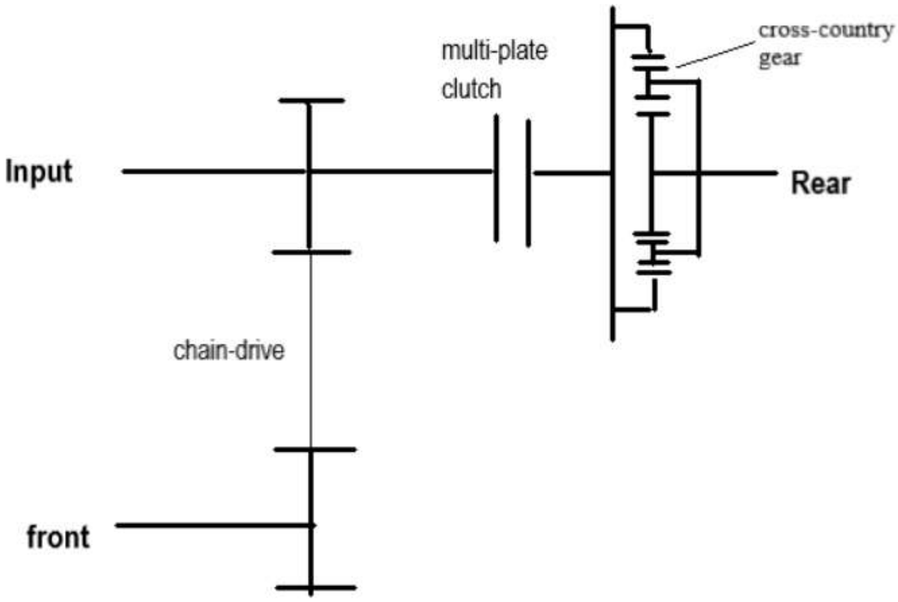


Figure 2-11: stick diagram of chain drive transfer case

MODE	CL		
FWD	OFF	$\mathbf{T}_F = \mathbf{T}_{\mathrm{IN}}$ $\mathbf{T}_R=0$	$\omega_{IN}= \omega_F$ $\forall\omega_R$
AWD	ON	$\mathbf{T}_F = \mathbf{T}_{\mathrm{IN}}$ $\mathbf{T}_R=\frac{1}{\sigma}*\mathbf{T}_{CL}-\mathbf{T}_c$ $\sigma=\frac{\mathrm{R}_R}{\mathrm{R}_S}$ $\mathbf{T}_c= \frac{1+\sigma}{\sigma}*\mathbf{T}_{CL}$	$\omega_{IN}= \omega_F$ $\omega_R = (1+\sigma)*\omega_c+\sigma*\omega_{IN}$

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

$$\mathbf{T}_{\mathrm{CL}}=\mathbf{N}_{\mathrm{d}}\mu\mathbf{F}_{\mathrm{a}}\mathbf{R}_{\mathrm{d}}*\sin(\Delta\omega),\mathbf{F}_{\mathrm{a}}=\mathrm{P}*\mathrm{A} \qquad 2\text{-}23$$

$$\Delta\omega=\omega_s-\omega_r=\omega_F-\omega_R \qquad 2\text{-}24$$

2.7 Transfer case for active torque bias System

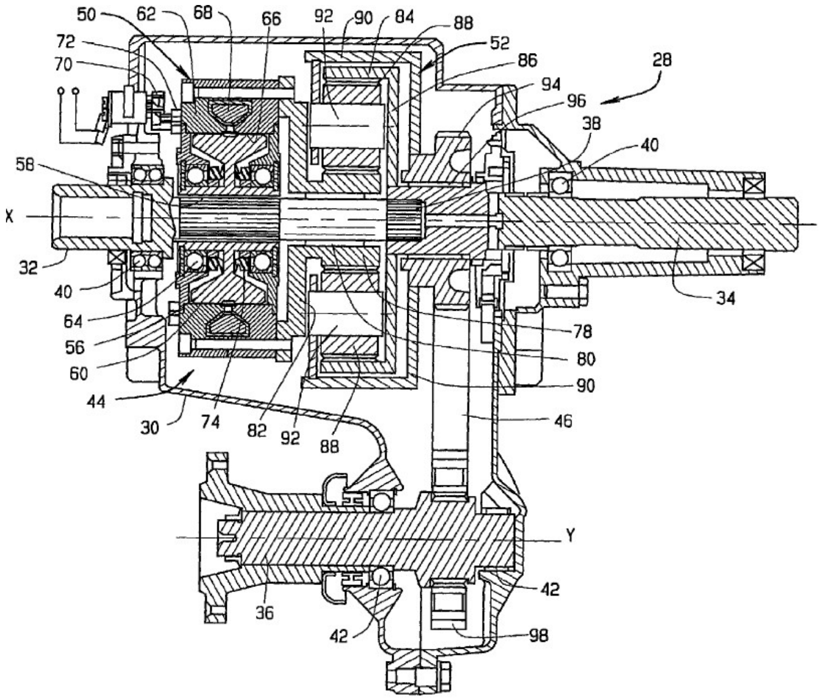


Figure 2-12: layout of Transfer case for active torque bias System[11]

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.

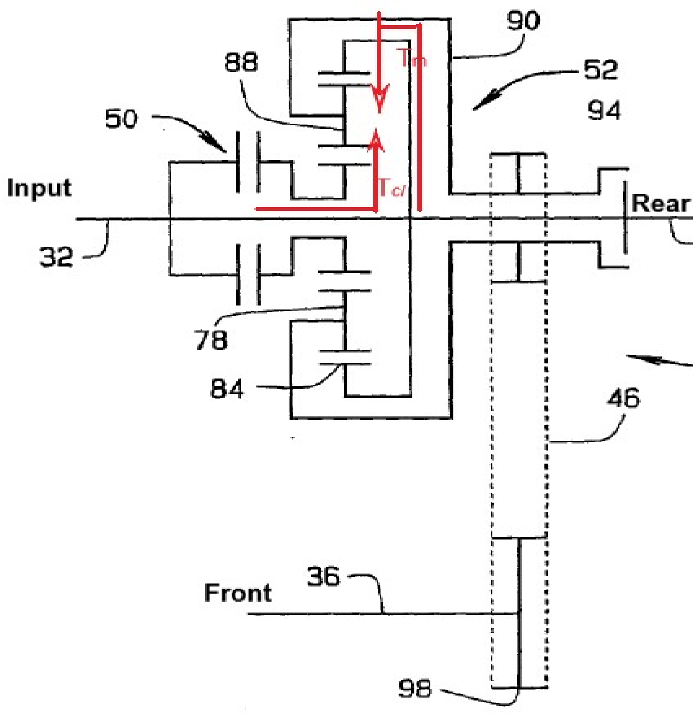


Figure 2-13: stick diagram of Transfer case for active torque bias System[12]

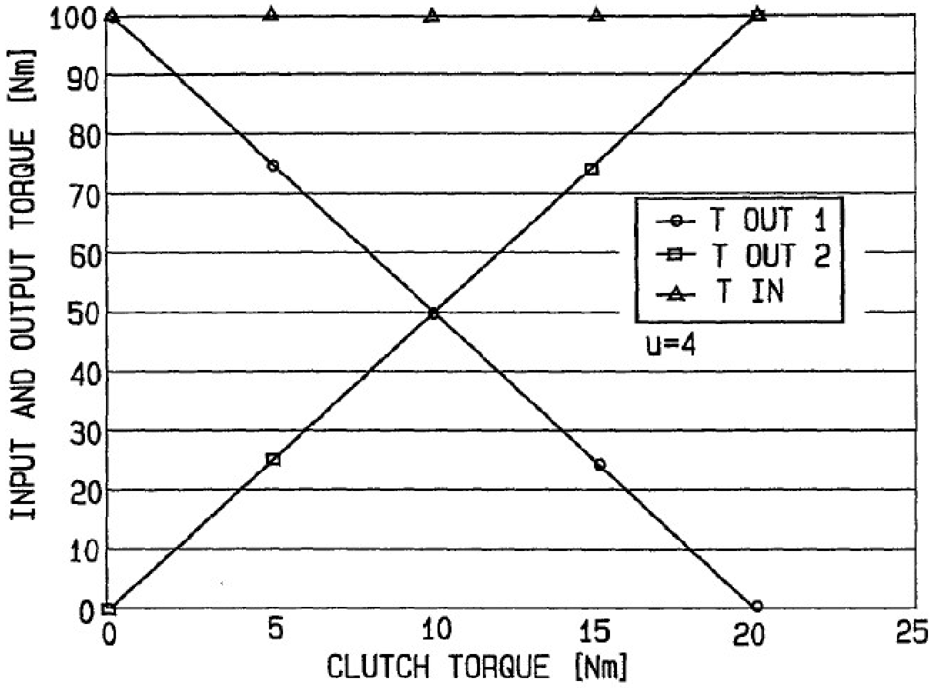


Figure 2-14: **Torque distribution versus clutch torque**

$\mathbf{T_{out\,1}}$- the torque of rear axle; $\mathbf{T_{out\,2}}$- the torque of front axle;

There are two paths to transfer torque from input shaft to the Secondary output Shaft 36.-a clutch path and a mechanical path.

$\mathbf{T}_{CL}$-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.

$\mathbf{T}_{m}$-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

$$\frac{\mathbf{T}_m}{\mathbf{T_{CL}}} = \mathrm{u} = 4 \qquad \mathbf{T}_F = \mathbf{T}_{CL} + \mathbf{T}_m \qquad 2\text{-}25$$

$\mathbf{T}_{F}$-The torque transferred into the front axles

MODE	CL		
RWD	OFF	$\mathbf{T}_R=\mathbf{T}_{IN}$ $\mathbf{T}_F=0$	$\omega_{IN}=\omega_R\ \forall\omega_F$
AWD	ON	$\mathbf{T}_F=\mathbf{T}_{CL}+\mathbf{T}_m=5\mathbf{T}_{CL}$ $\mathbf{T}_R=\mathbf{T}_{IN}-\mathbf{T}_F$	$\omega_{IN}=(1+\sigma)\,\omega_F+\sigma\omega_R$

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

$$\mathbf{T}_{CL}=\mathbf{N}_d\mu\mathbf{F}_a\mathbf{R}_d\sin(\Delta\omega) \qquad 2\text{-}26$$

$$\mathbf{F}_a=P^*A \quad \Delta\omega=\omega_F-\omega_R \qquad 2\text{-}27$$

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

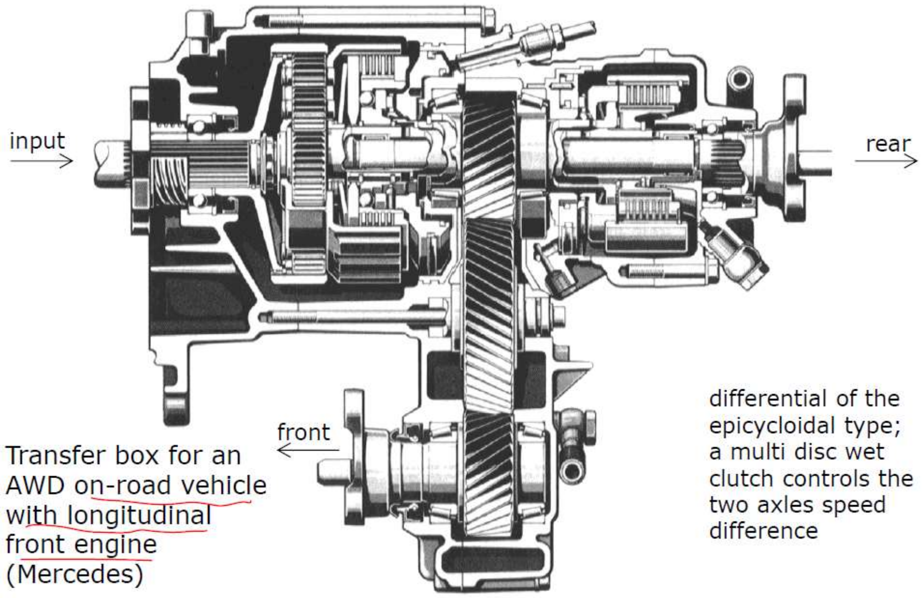


Figure 2-15 layout of transfer case AWD on-road vehicle[13]

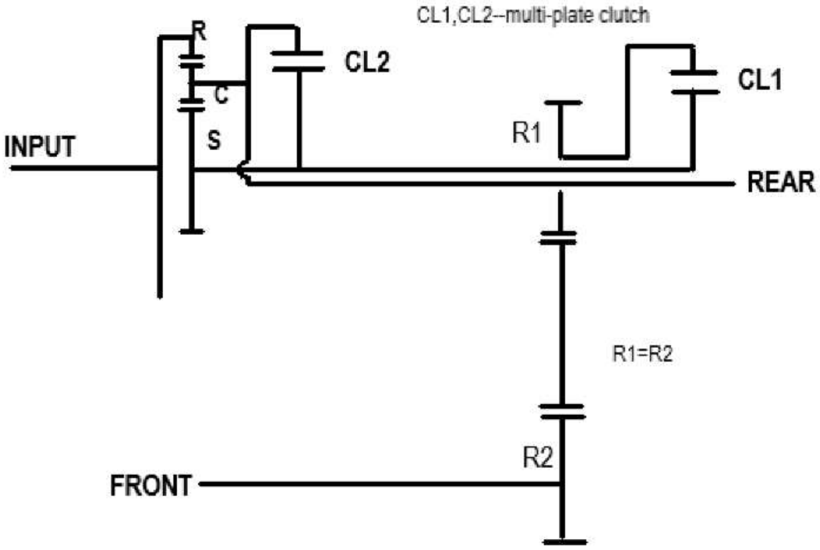


Figure 2-16 stick diagram of transfer case of AWD on-road vehicle

Mode	CL1	CL2		
RWD	OFF	ON	$T_R = T_{IN} \quad T_F=0$	$\omega_{IN}= \omega_R \quad \forall \omega_F$
AWD	ON	OFF	$\frac{T_R}{T_{IN}}=\frac{1+\sigma}{\sigma} \quad \sigma=\frac{R_R}{R_S}$ $\frac{T_F}{T_{IN}} = \frac{1}{\sigma}$	$\omega_F=(1+\sigma)*\omega_R+\sigma*\omega_{IN}$ $\omega_{CL1}= \omega_F$ $\omega_{CL2}=\omega_F$
LOCKED	ON	ON	$T_F = \frac{1}{\sigma} * T_{IN}+T_{CL}$ $T_R=\frac{1+\sigma}{\sigma}*T_{IN}-T_{CL}$	

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

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

$$\Delta\omega=\omega_s-\omega_r=\omega_F-\omega_R \qquad 2\text{-}29$$

3.Simulink model with Simscape Driveline

3.1 The model of 2WD vehicle

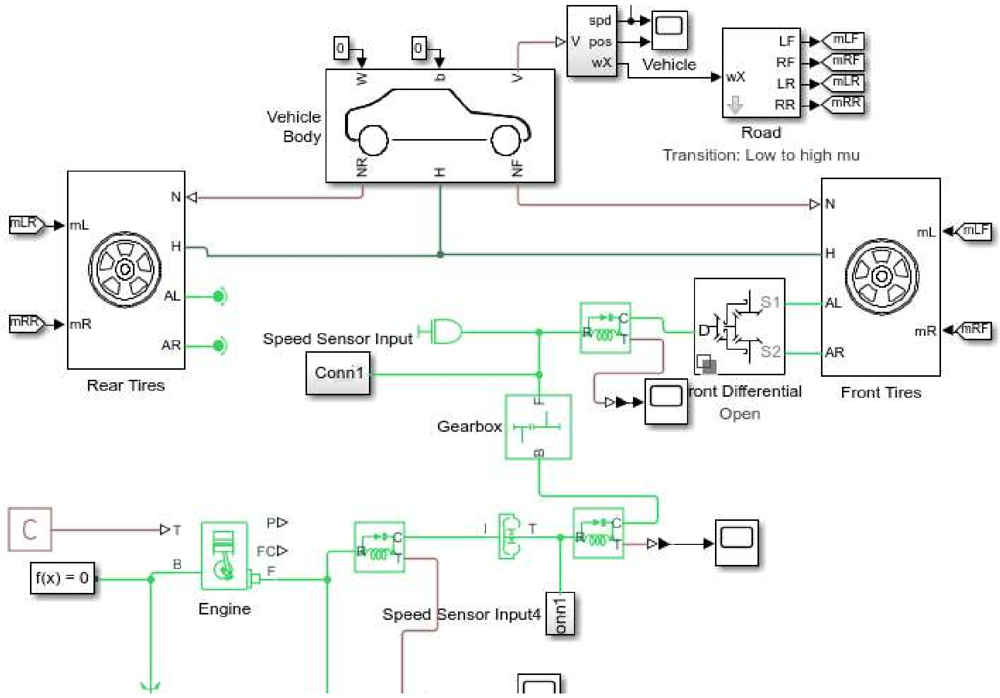


Figure 3-1: the model of FWD vehicle

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 Ω. 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 $\tau = P/\Omega$.

Port	Description
B	Rotational conserving port representing the engine block
F	Rotational Conserving port representing the engine crankshaft
T	Physical signal input port specifying the normalized engine throttle level
P	Physical signal output port reporting the instantaneous engine power
FC	Physical signal output port reporting the fuel consumption rate

Table 3-1: Description of port in engine block

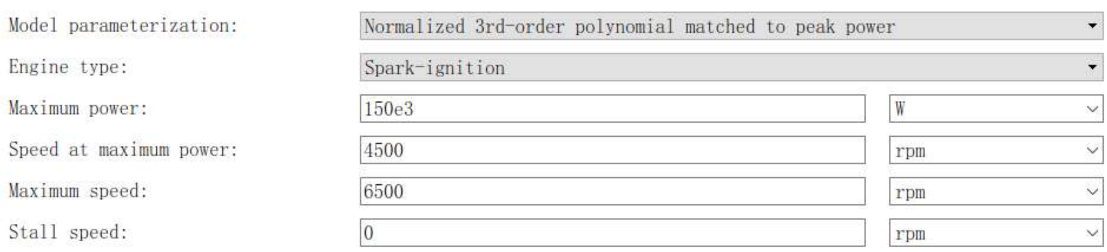


Figure 3-2: Parameters of engine torque

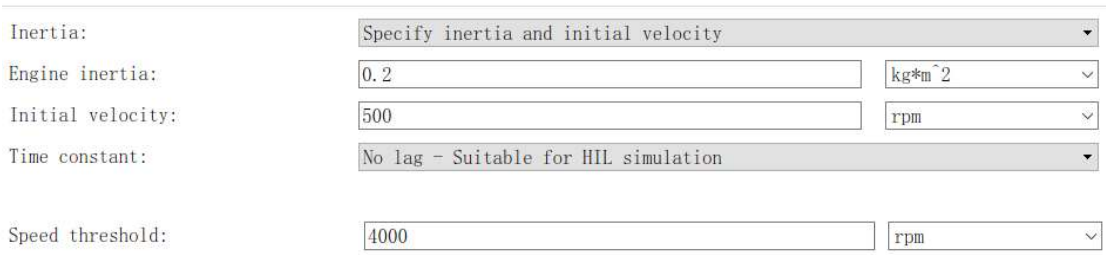


Figure 3-3: Parameters of engine dynamics

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 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

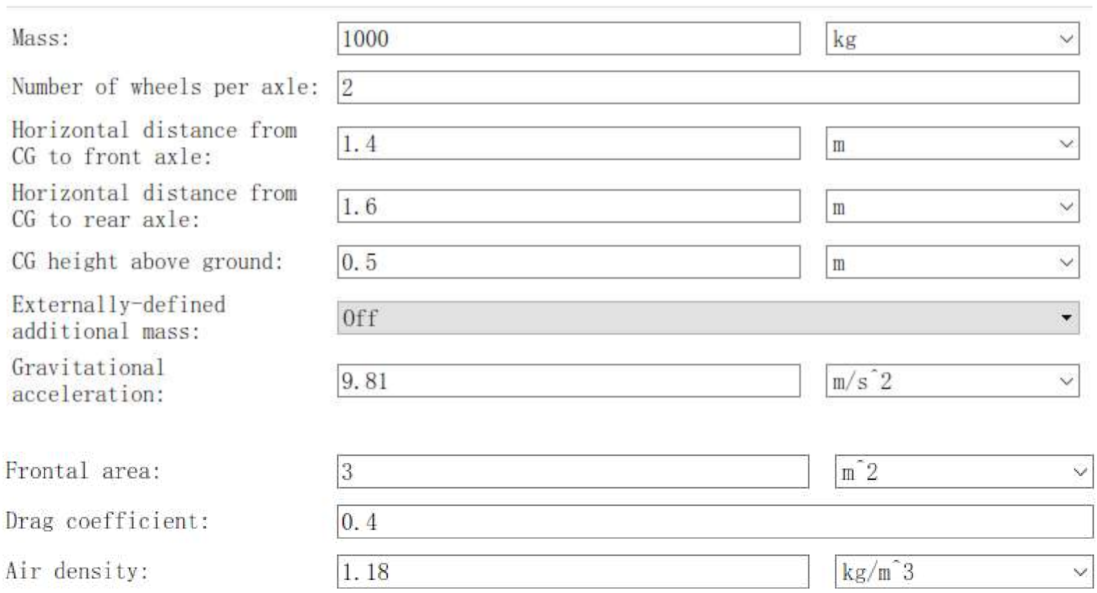
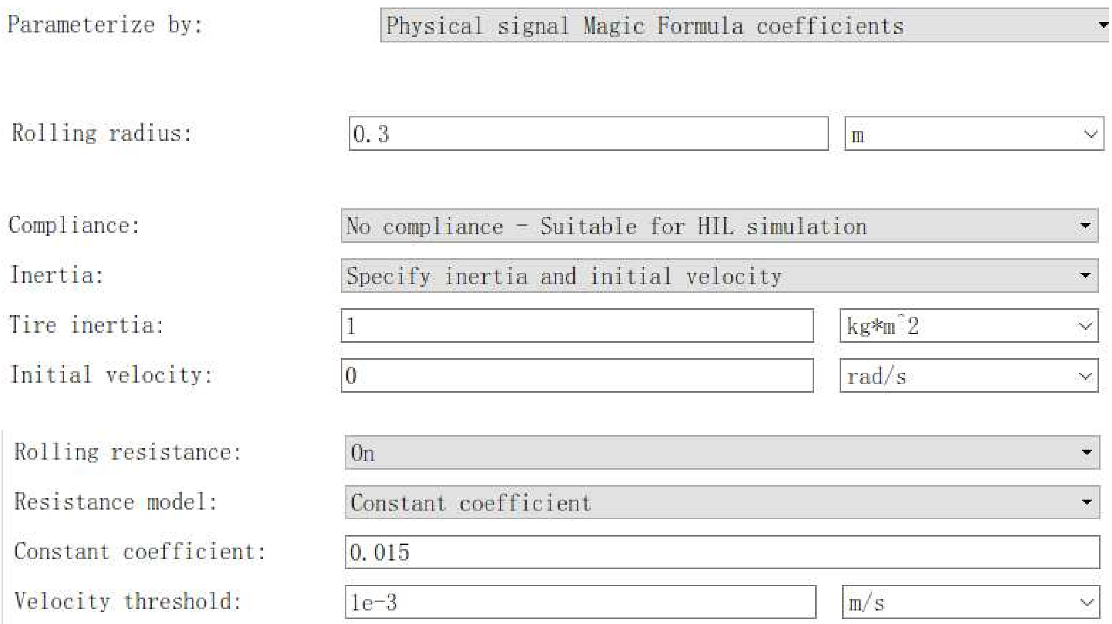


Figure 3-4: Parameters of vehicle body block

Parameters of Tires



3.2 4WD vehicle with different central differential

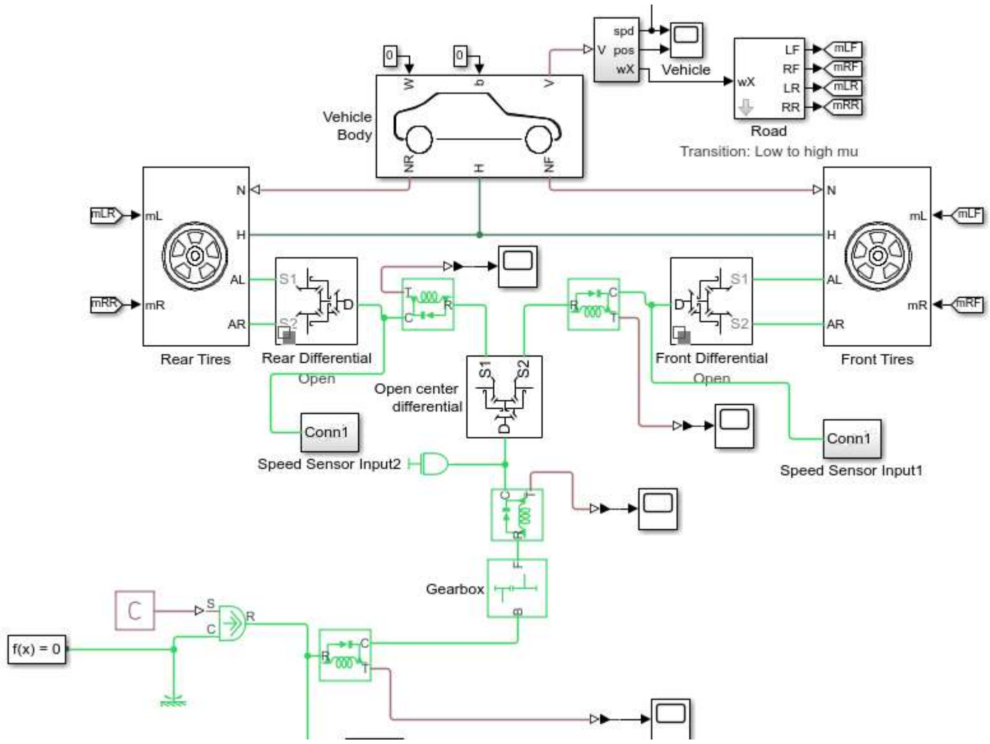


Figure 3-5: The model of 4wd vehicle

3.2.1 Open central differential

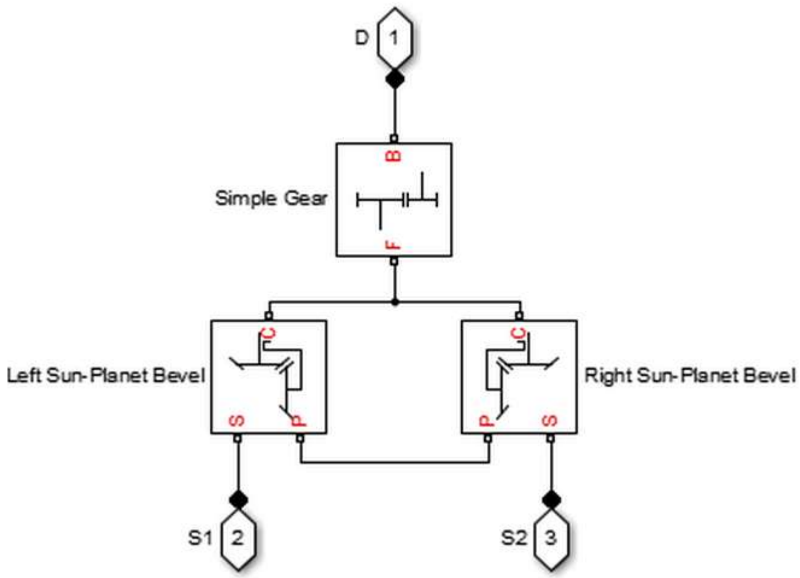


Figure 3-6: Model layout of open differential

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.

$$\boldsymbol{r}_C = \boldsymbol{r}_S \pm \boldsymbol{r}_p \qquad \boldsymbol{r}_C \boldsymbol{\omega}_C = \boldsymbol{r}_S \boldsymbol{\omega}_S \pm \boldsymbol{r}_p \boldsymbol{\omega}_p \tag{3-1}$$

r_C is the radius of the carrier gear.

ω_C is the angular velocity of the carrier gear.

r_S is the radius of the sun gear.

ω_S is the angular velocity of the sun gear.

r_P is the radius of the planet gear.

ω_P is the angular velocity of the planet gear.

The planet-sun gear ratio is defined as

$$\mathbf{g}_{PS} = \frac{\boldsymbol{r}_p}{\boldsymbol{r}_s} = \frac{\boldsymbol{N}_p}{\boldsymbol{N}_s} \tag{3-2}$$

The torque transfer is defined as

$$\tau_P = \tau_{loss} - g_{PS}\tau_S, \tag{3-3}$$

τ_{loss} is the torque loss.

τ_s is the torque for the sun gear.

τ_p is the torque for the planet gear.

Ideal open center differential

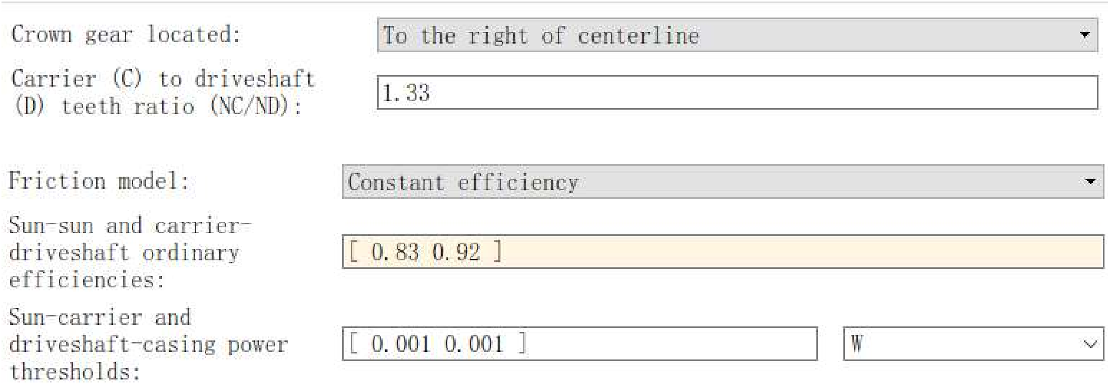
TBR=1 $\boldsymbol{\eta}_i$=100%

Settings

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

Real open center differential

TBR=1.2 η_i=83%



3.2.2 Self-locking central ZF differential

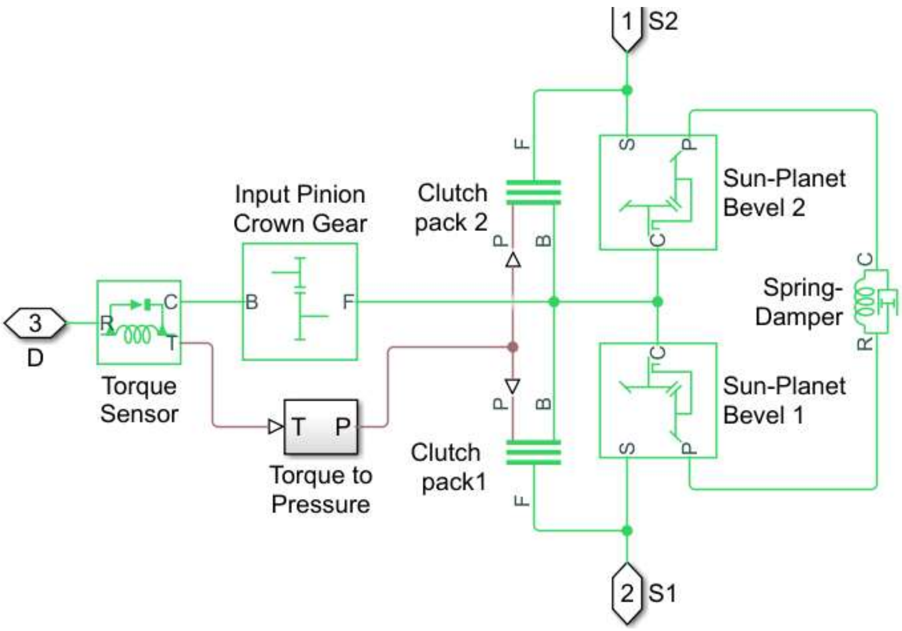


Figure 3-7: Model layout of Self-locking central ZF differential

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

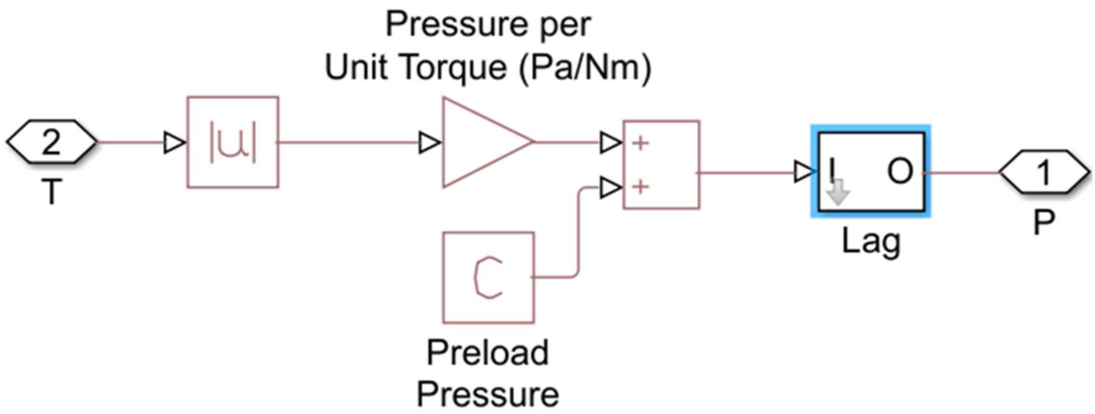
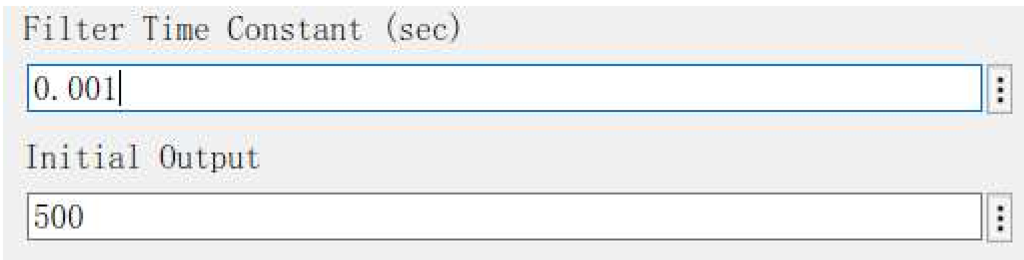
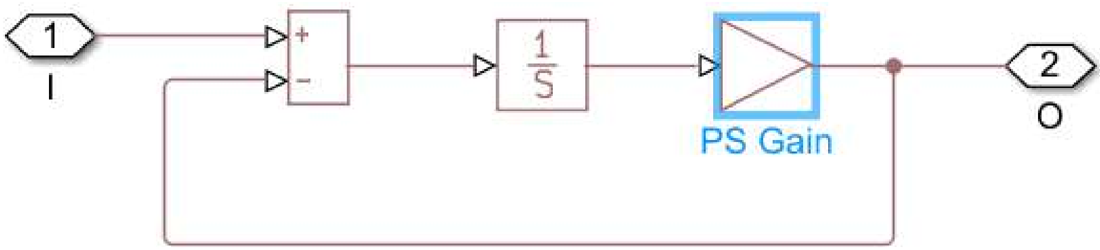


Figure 3-8: The model of Torque to pressure

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.

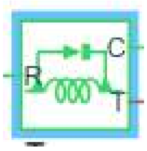
Lag





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

Block description

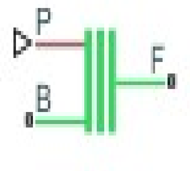


Torque sensor The Ideal Torque Sensor block represents a device that converts a variable passing through the sensor into a control signal proportional to the torque. The sensor is ideal because it does not account for inertia, friction, delays, energy consumption.

Ports T—Torque N*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:

$$\omega = \omega_F - \omega_B. \qquad 3\text{-}4$$

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

$$\tau_K = \mu\omega + \tau_{contact} \qquad 3\text{-}5$$

The contact friction torque:

$$\tau_{contact} = k_K \cdot D \cdot N \cdot r_{eff} \cdot P_{fric} \cdot A \geq 0 . \qquad 3\text{-}6$$

kinetic friction coefficientk_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}:

$$r_{eff} = \frac{2}{3}\frac{r_o{}^3 - r_i{}^3}{r_o{}^2 - r_i{}^2} \qquad 3\text{-}7$$

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

The clutch de-rating factor D:

$$D \rightarrow \frac{3}{4}\frac{(r_o + r_i)^2}{r_o^2 + r_o r_i + r_i^2} \quad 3\text{-}8$$

Settings

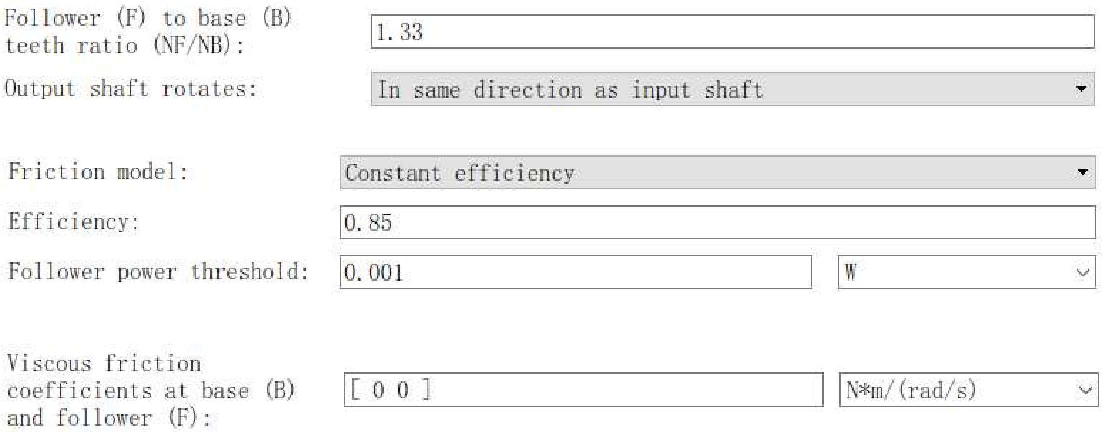


Figure 3-9: The parameters of Input Pinion Crown Gear

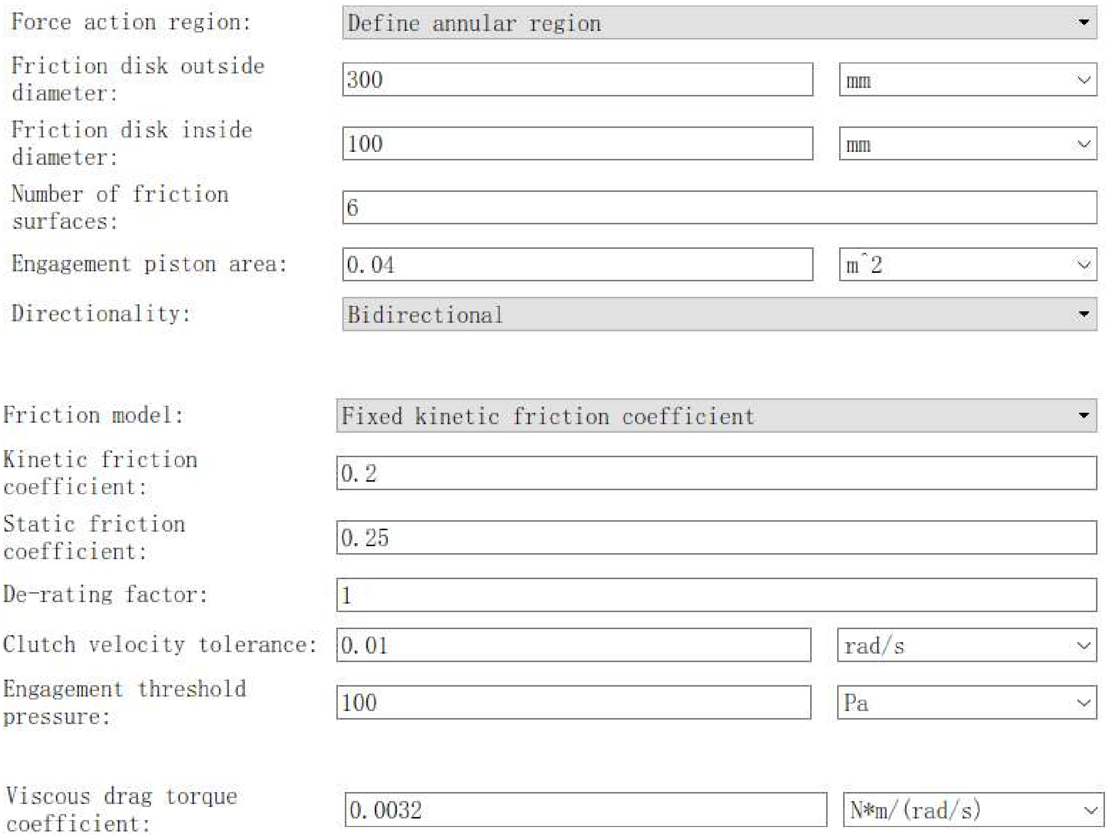


Figure 3-10: The parameters of Clutch pack

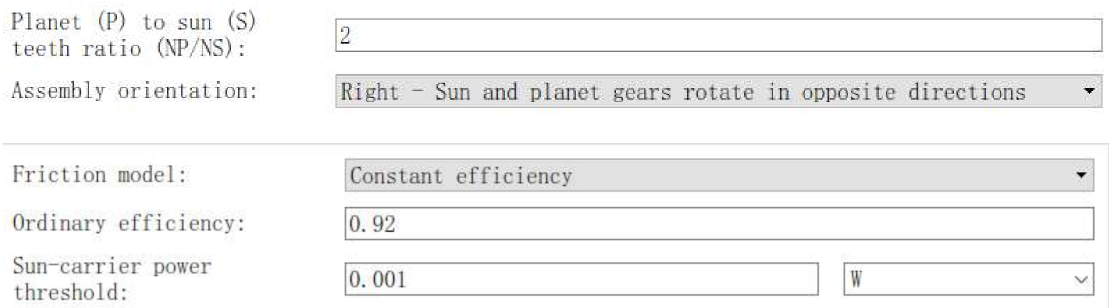


Figure 3-11: The parameters of Sun-Planet Bevel

3.2.3 Torsen center differential

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

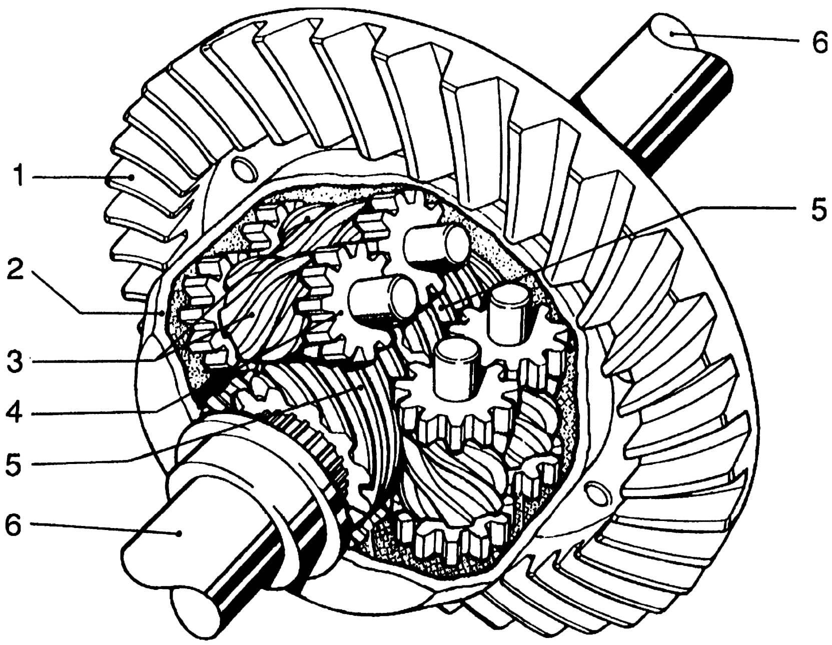


Figure 3-12: Torsen center differential[14]

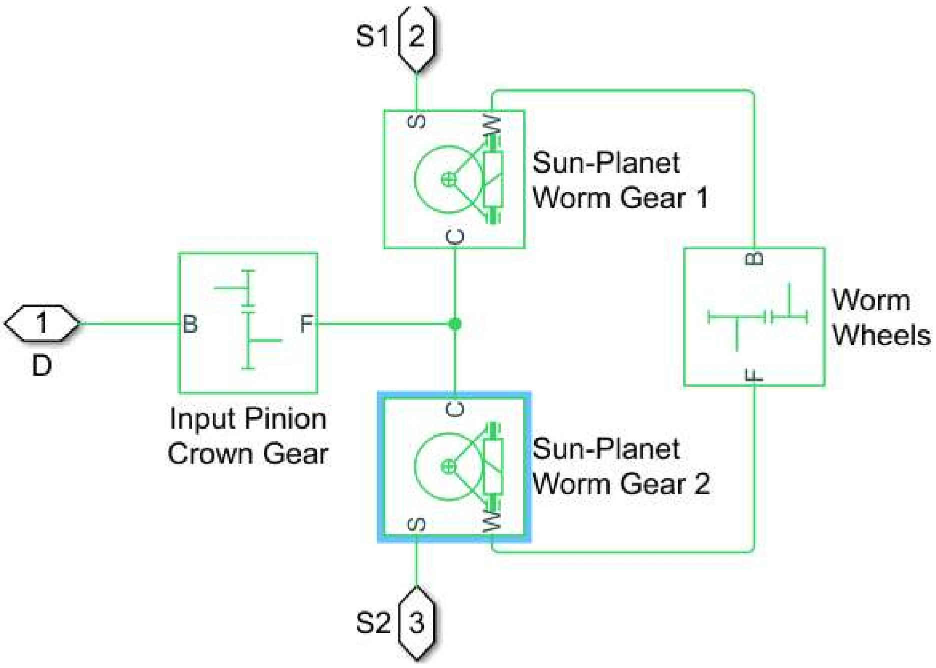


Figure 3-13: Model layout of torsen differential

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

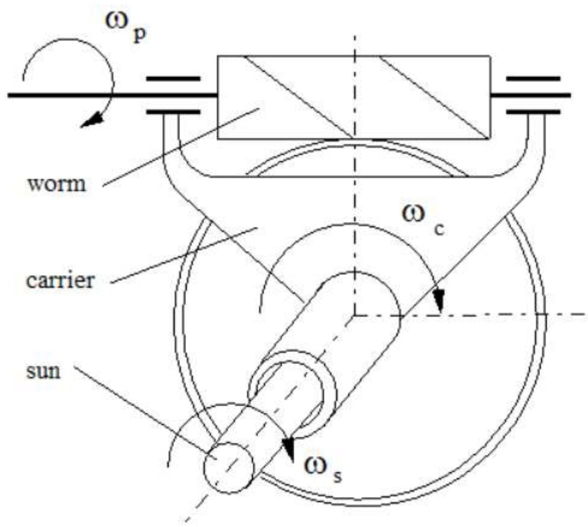


Figure 3-14 the layout of Sun-Planet Worm Gear

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 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:

$$\omega_S = \omega_p / R_{WG} + \omega_C \qquad 3\text{-}9$$

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

ω_S—Angular velocity of the sun gear ω_p—planet(Worm) angular velocity

ω_C—carrier angular velocity

The torque transfer is:

$$R_{WG}\tau_p + \tau_S - \tau_{Loss} = 0,\ \tau_C = -\tau_S \qquad 3\text{-}10$$

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

τ_p—Torque applied to the planet shaft τ_S—Torque applied to the sun shaft

τ_C—Torque applied to the carrier shaft τ_{Loss}—Torque loss due to meshing friction.

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

$$\eta_{WG} = (\cos\alpha - k*\tan\lambda)/(\cos\alpha + k/\tan\lambda) \qquad 3\text{-}11$$

$$\eta_{GW} = (\cos\alpha - k/\tan\lambda)/(\cos\alpha + k*\tan\alpha) \qquad 3\text{-}12$$

α-Normal pressure angle; λ-Worm lead angle; 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.

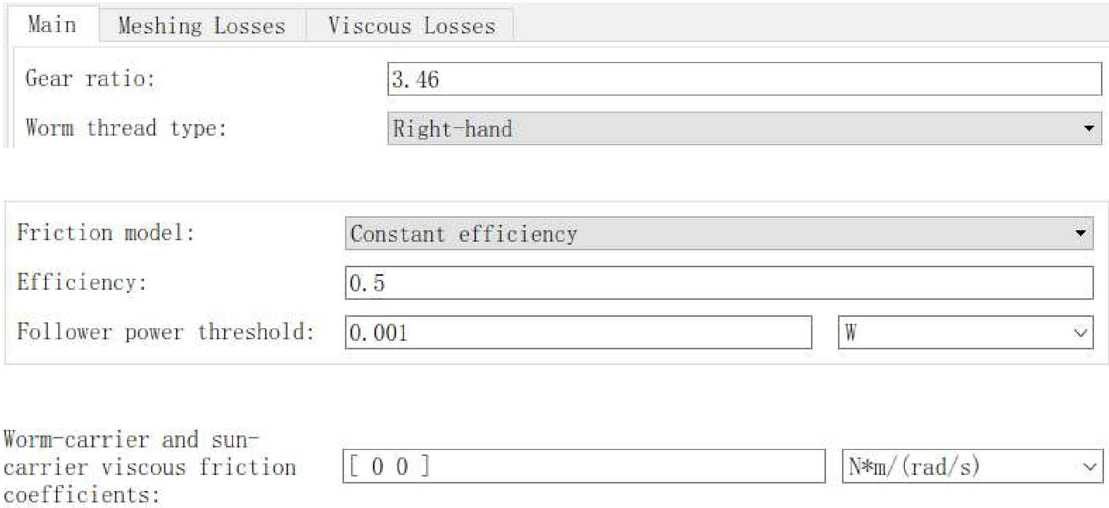


Figure 3-15: Parameters of Sun-Planet Worm Gear

3.3 The model differential Test

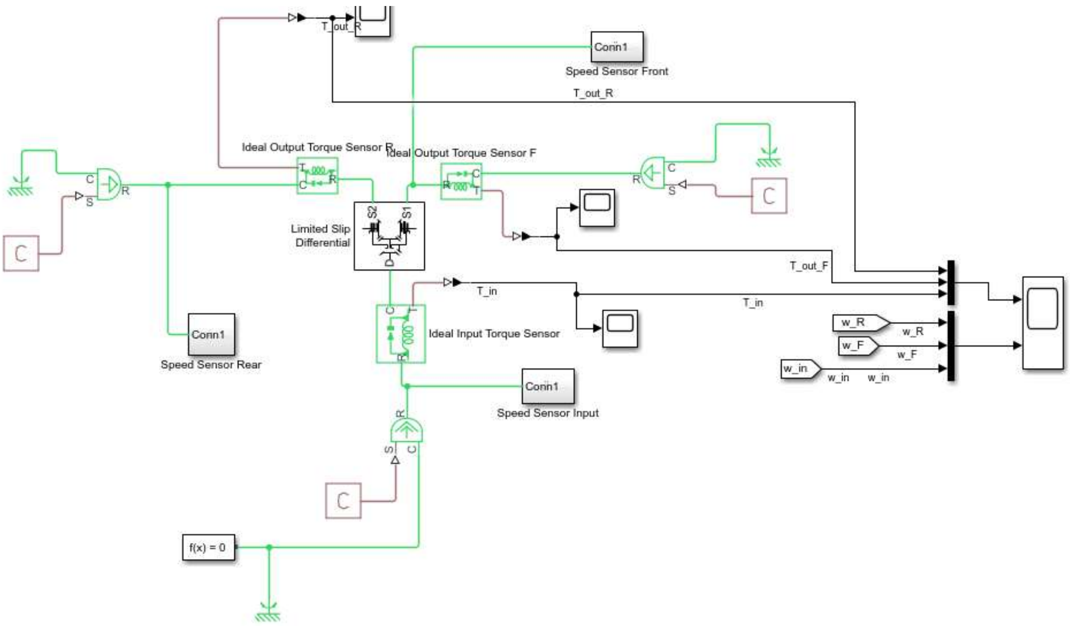


Figure 3-16: Model layout for differential testing

Before investigating that the differential works as a center coupling in AWD vehicle, I tend to test the differential working in a 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:20rad/s. I keep internal efficiency of front and rear sun-planet bevel gears in a constant value. I also set the preload pressure in 500pa, I change the pressure per unit torque in 3 values: 20 Pa/Nm (LSD1), 40Pa/Nm (LSD2), 60Pa/Nm (LSD3).

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

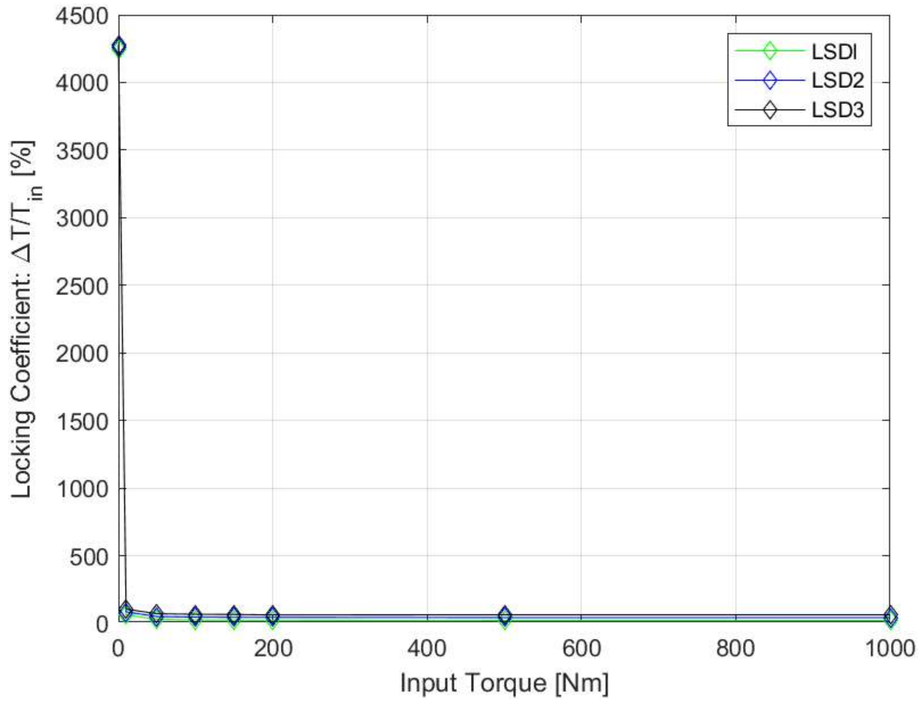


Figure 3-17: Locking coefficient versus input torque for Limited-slip differential

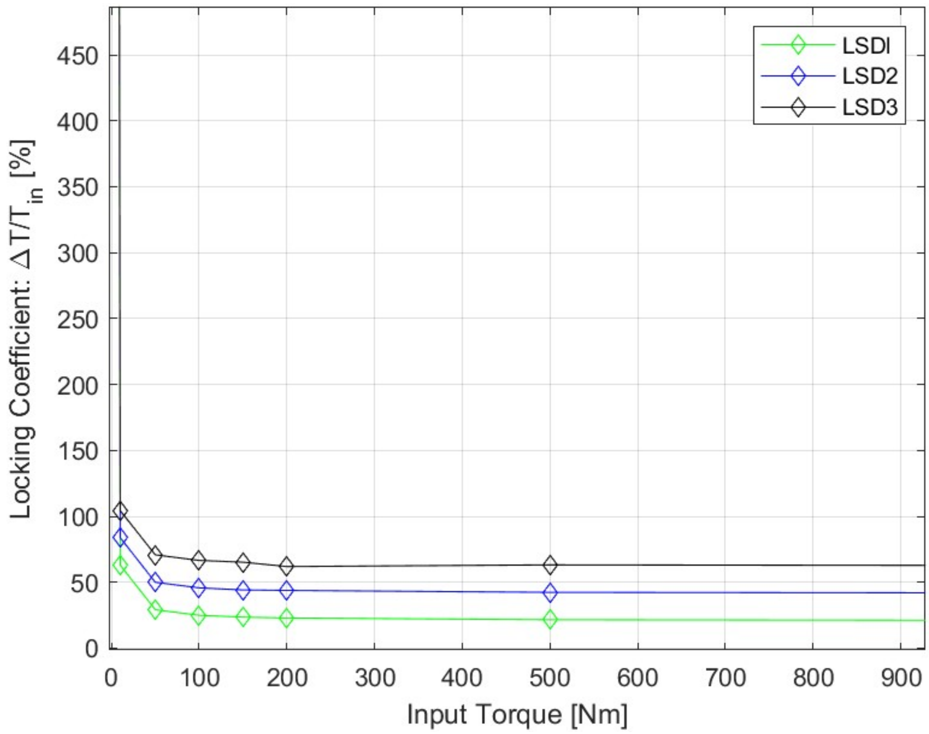


Figure 3-18: Partial enlargement of Figure 3-22

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.

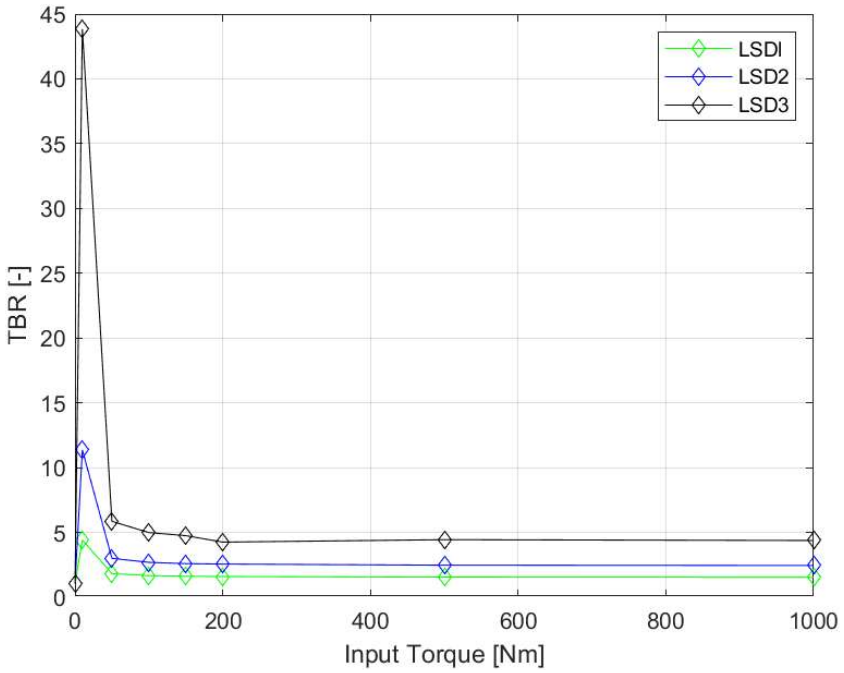


Figure 3-19: TBR versus input torque for Limited-slip differential

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.

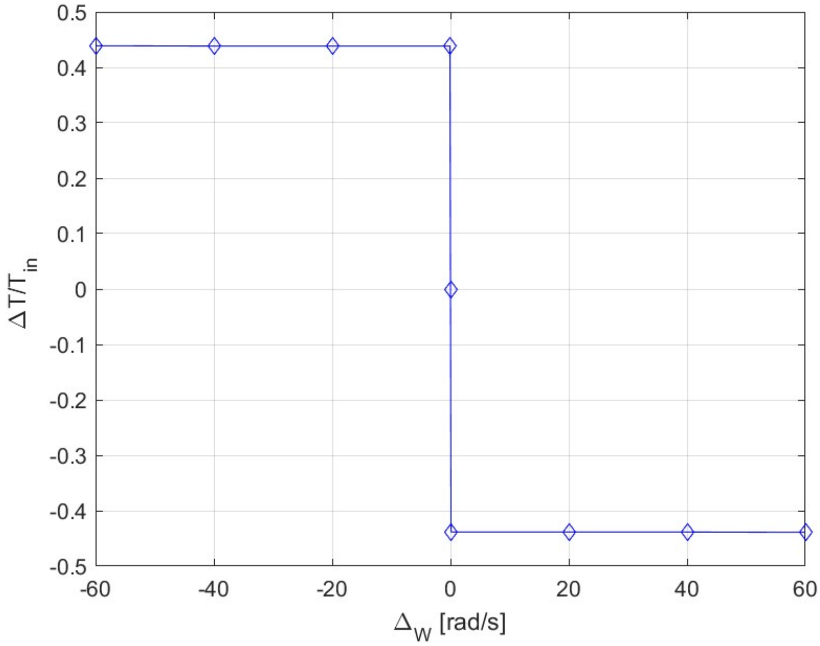


Figure 3-20: ΔT VS Δω for Limited-slip differential

Torque split ratio

To test the torque split ratio between front and rear axles, I set speed difference Δω=20rad/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: $\eta_{sun-sun}=100\%$ $\eta_{\mathrm{carrier-driveshaft}}=100\%$

Real 1: $\eta_{sun-sun}=83\%$ $\eta_{\mathrm{carrier-driveshaft}}=92\%$

Real 1: $\eta_{sun-sun}=74\%$ $\eta_{\mathrm{carrier-driveshaft}}=92\%$

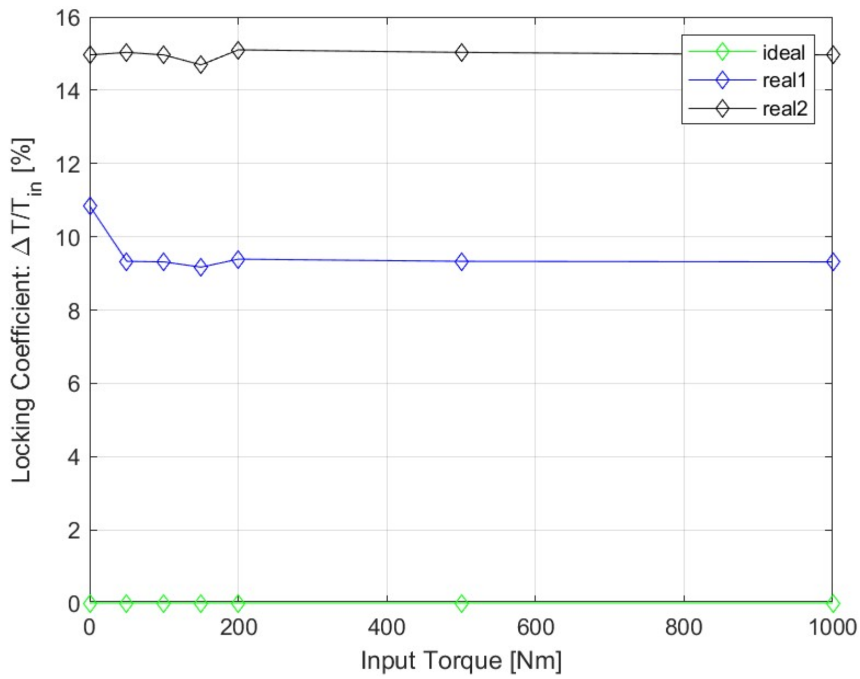


Figure 3-21: Locking coefficient versus input torque for open differential

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

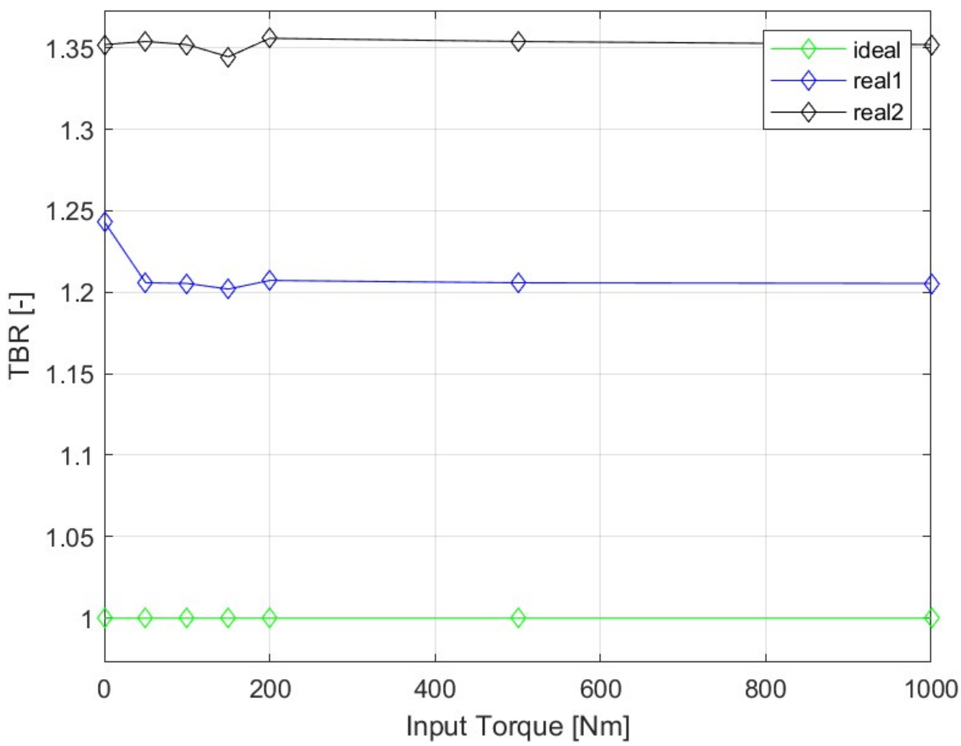


Figure 3-22: TBR versus input torque for open differential

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

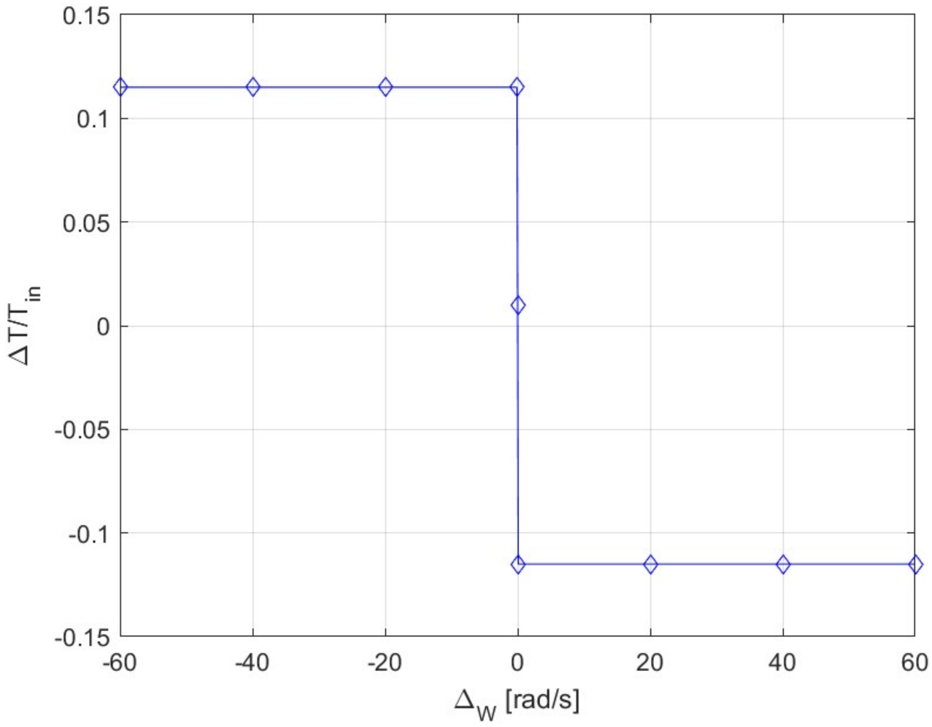


Figure 3-23: ΔT VS $\Delta\boldsymbol{\omega}$ for open differential

From the graph, I got: $\Delta\omega > 0$ ΔT=$T_F - T_R$=-23Nm $\Delta\omega = 0$ ΔT=$T_F - T_R$=0

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

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 η=100%, real1η=83% real2η=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: η_{Input}=95% η_{front}=100% η_{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(η_{worm} =50%); TCD2(η_{worm} =40%);

TCD3(η_{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.

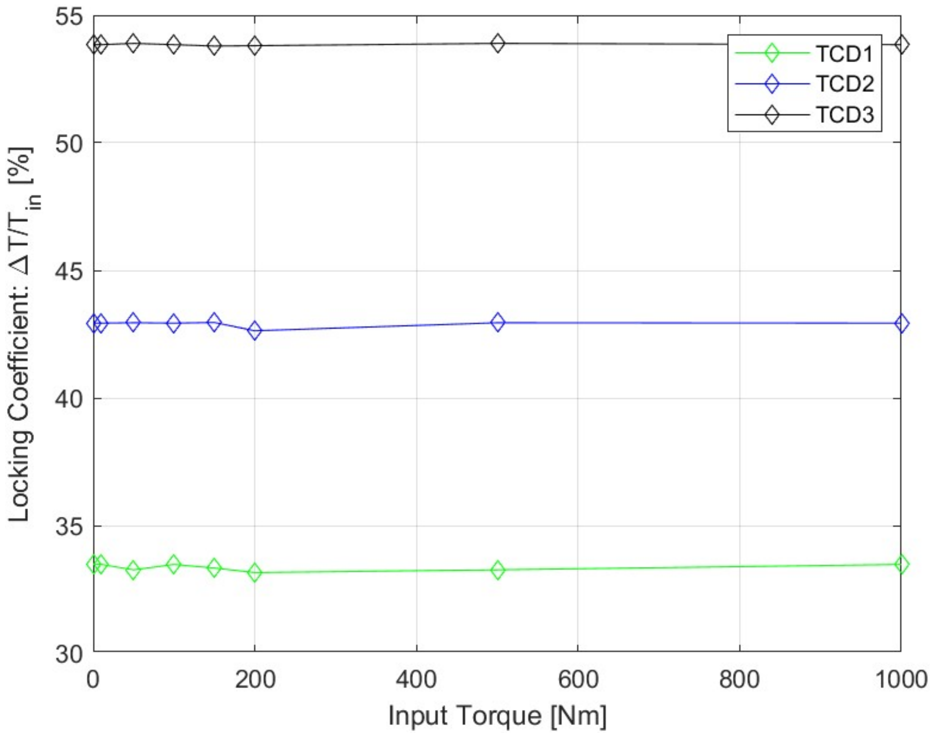


Figure 3-24: Locking coefficient versus input torque for torsen differential

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

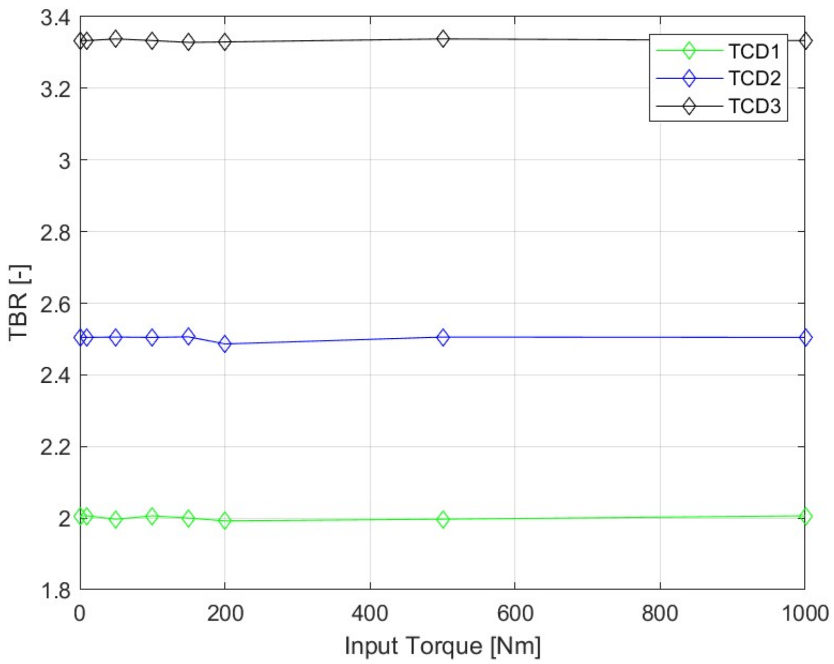


Figure 3-25: TBR versus input torque for torsen differential

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 η_{front} and η_{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

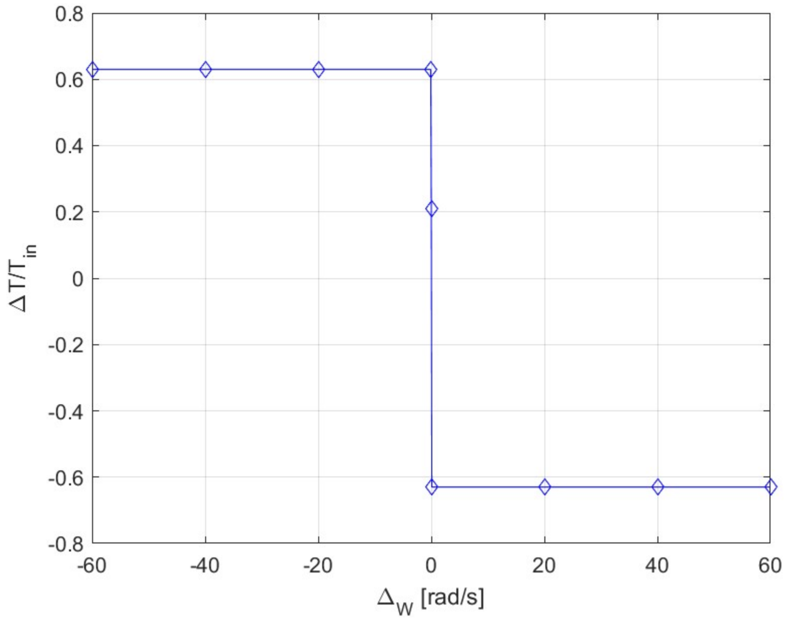


Figure 3-26: ΔT VS Δω for Torsen differential

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=T_F:T_R=33.4:66.6;

TCD2 : i=T_F:T_R=28.7:71.3;

TCD3: i=T_F:T_R=23.1:76.9

3.4 Four-wheel drive vehicle with Viscous coupling

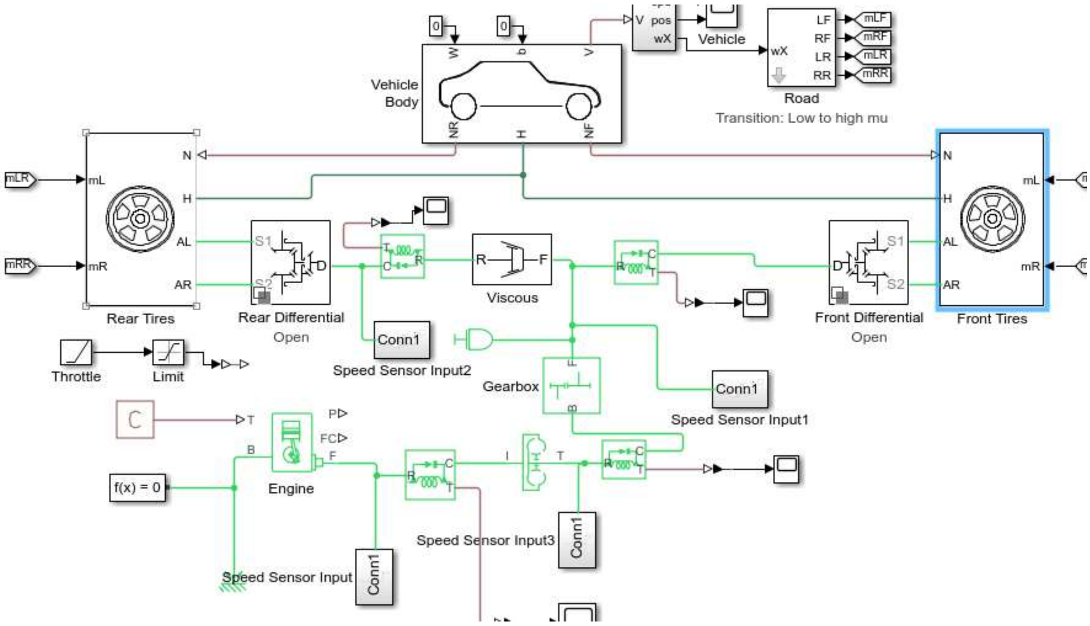


Figure 3-27: Model layout of 4WD vehicle with Viscous coupling

Nonlinear Rotational Damper

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:

$$T = b_1\omega + sign(\omega)\cdot b_2\omega^2 + b_3\omega^3 + sign(\omega)\cdot b_4\omega^4 + b_5\omega^5, \qquad \text{3-19}$$

T — Damping torque

$b_1, b_2, ..., b_5$ — Damping coefficients

ω — Relative angular velocity between ports R and C, $\omega = \omega_R - \omega_C$

ω_R — Absolute angular velocity associated with port R

ω_C — Absolute angular velocity associated with port C

Viscous coupling testing

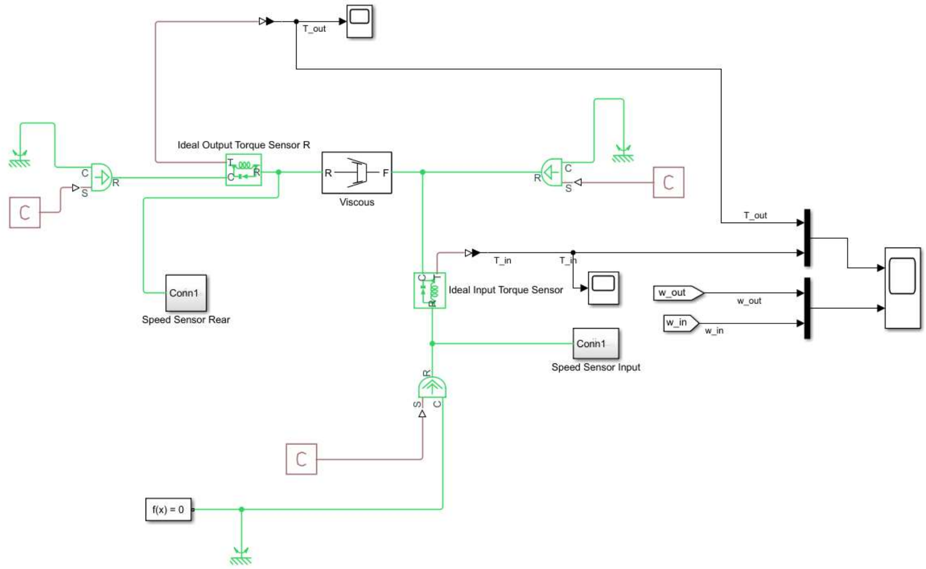


Figure 3-28: Model layout of Viscous coupling testing

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

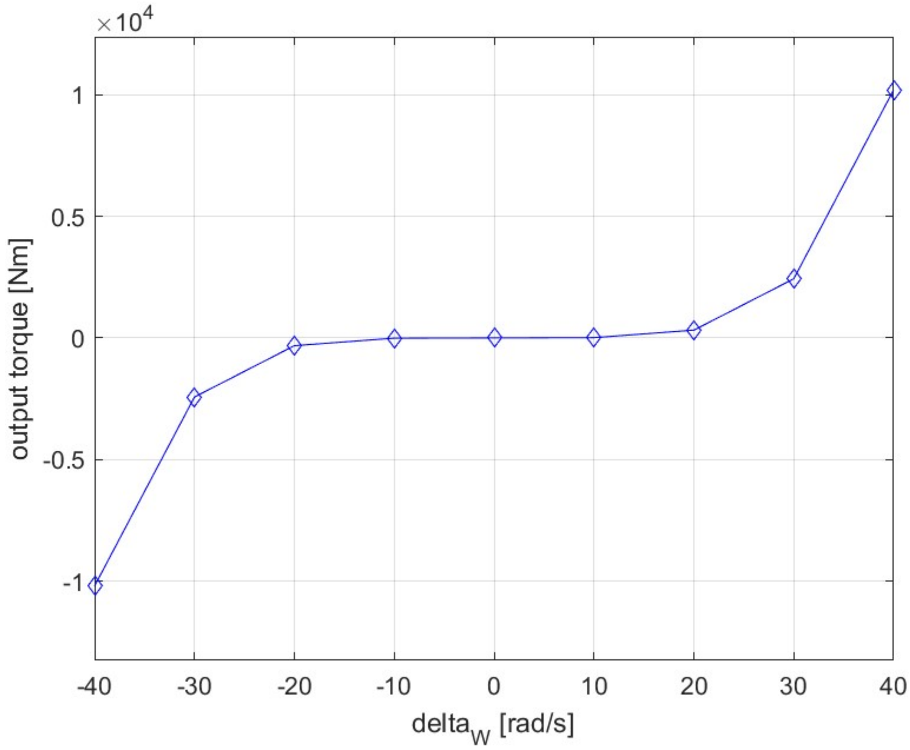


Figure 3-29: **Output torque** VS **Δω** for Viscous coupling

4.SIMULATION RESULTS

4.1 Working Condition

4.1.1 Parameters

	OPDI	OPDR	TCD	Viscous	FWD
Gearbox ratio and efficiency	2/99%	2/99%	2/99%	2.66/99%	2.66/99%
Central Differential Pinion/Crown ratio and efficiency	1.33/100%	1.33/83%	1.33/100%	--	--
Torsen Differential Sun-Planet Worm Gear and efficiency	--	--	3.46/ η_{WG}=0.84 η_{GW}=0.5	--	--
Front Differential Pinion/Crown ratio and efficiency	1.33/100%	1.33/100%	1.33/100%	1.33/100%	1.33/100%
Rear Differential Pinion/Crown ratio and efficiency	1.33/100%	1.33/100%	1.33/100%	1.33/100%	--
TBR	1	1.2	2.5	--	--
b	0	0.09	0.43	--	--

OPDI—ideal open differential; OPDR—real open differential;

TCD—torsen differential; Viscous—Viscous coupling; FWD—front wheel drive

η_{WG}, η_{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).

	B	C	D	E
Dry	10	1.3	1	0.97
Wet	12	2.3	0.82	1
Snow	5	2	0.3	1
Ice	4	2	0.1	1

Through the equation:

$$\mu=D\sin(\arctan(Bx-E[Bx-\arctan Bx])) \qquad 4\text{-}1$$

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.

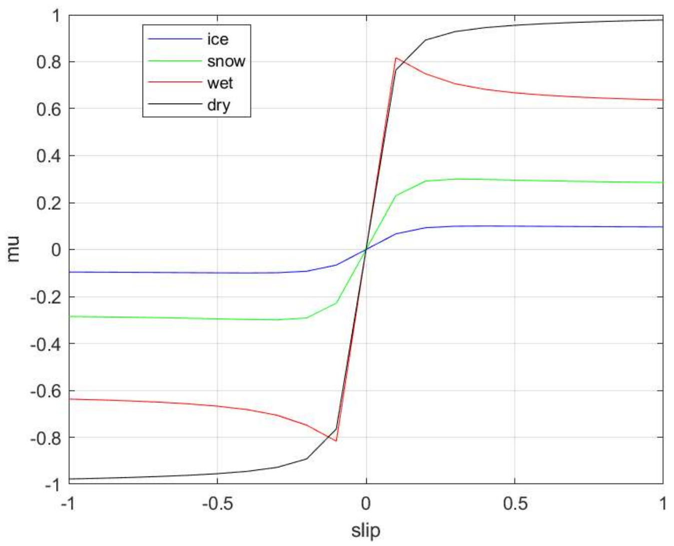


Figure 4-1: μ of different surface in different slip

From this figure, I got the max friction coefficient of each surface.

	ice	snow	wet	dry
μ_{max}	0.1	0.3	0.82	0.97

Table4-1

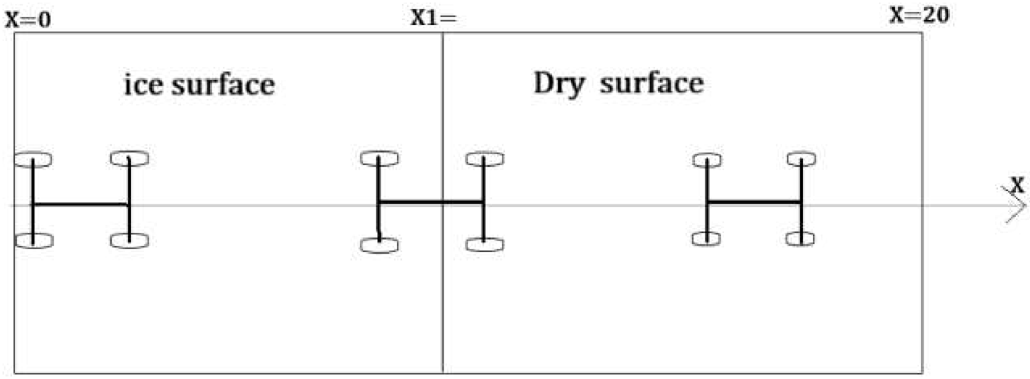


Figure 4-2: schematic diagram of Simulink test

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

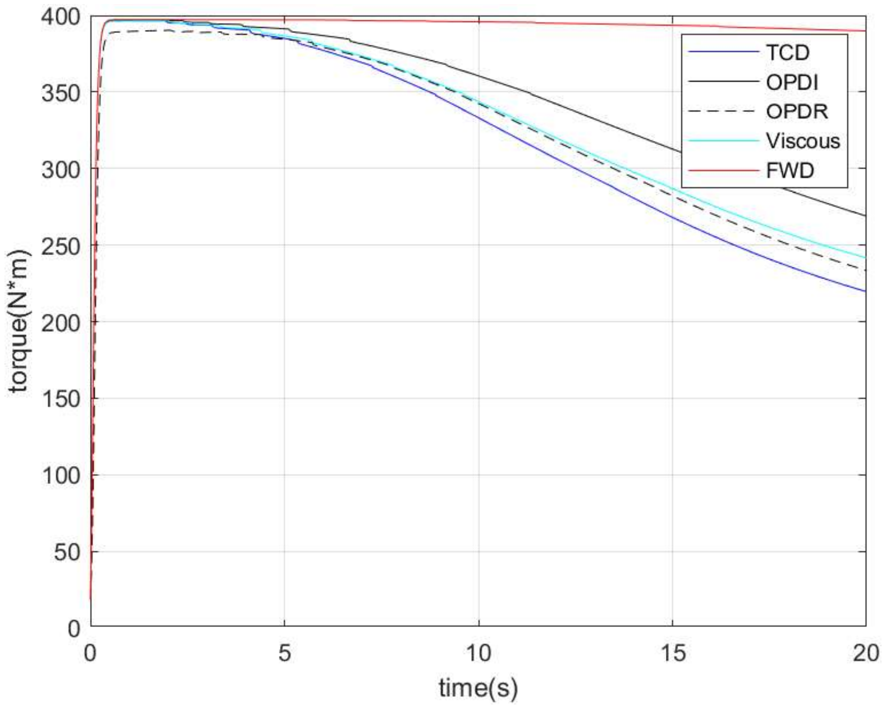


Figure 4-3: Engine torque of different transfer case

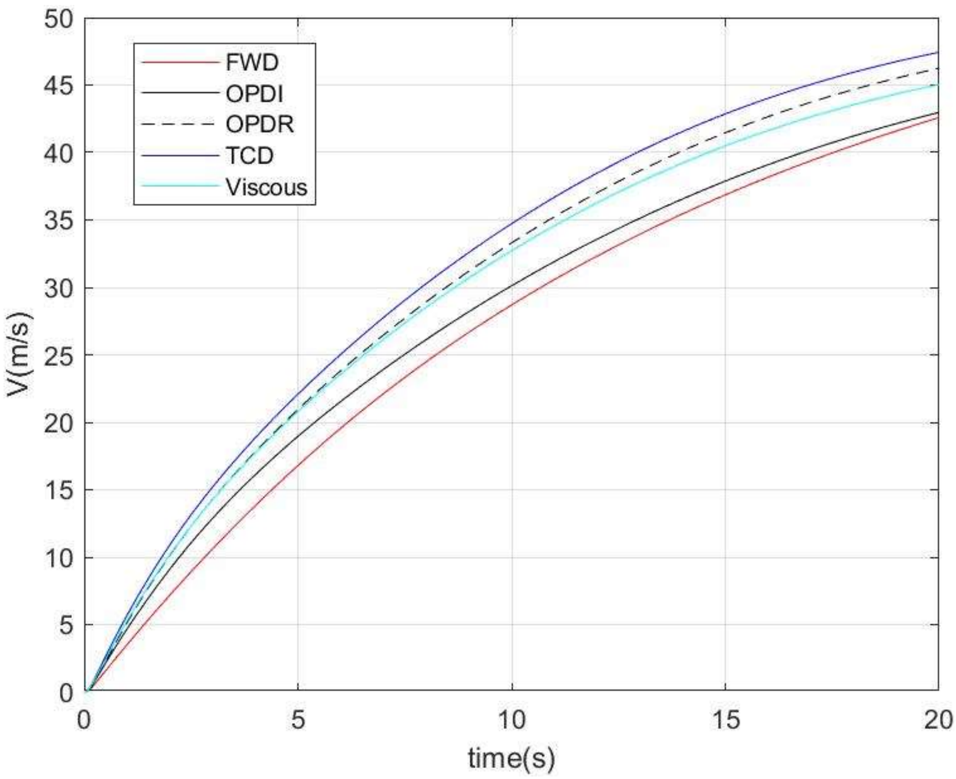


Figure 4-4: Engine speed of different transfer case

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.

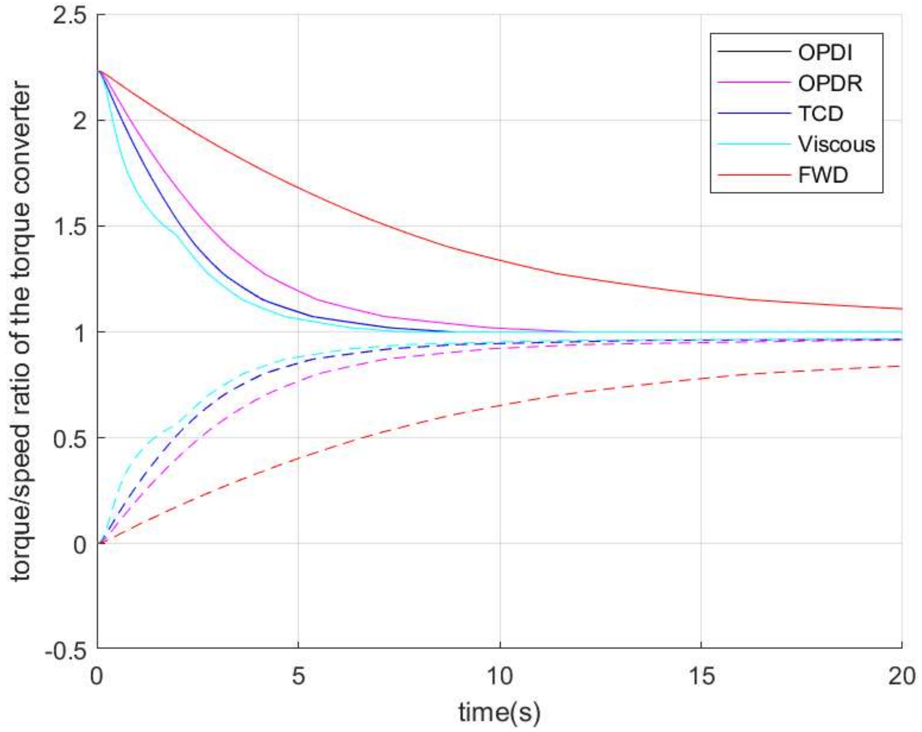


Figure 4-5: Torque/speed ratio between input and output axle

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)

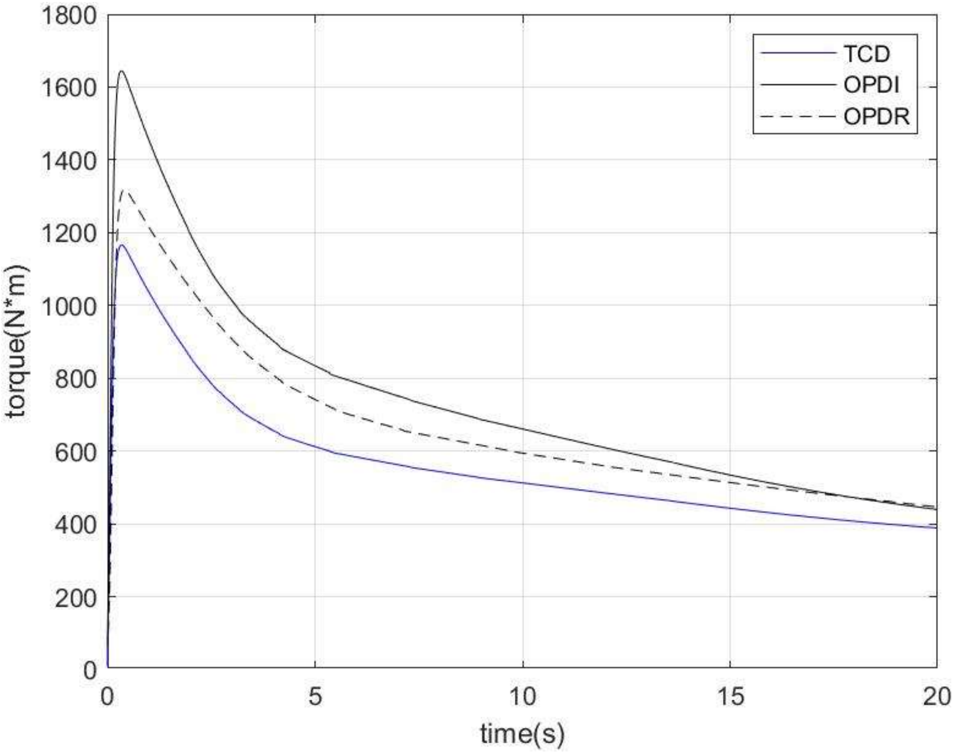


Figure 4-6: Input torque of differentials

4.2.2 Performance of coupling devices

Open differential(η=100%)

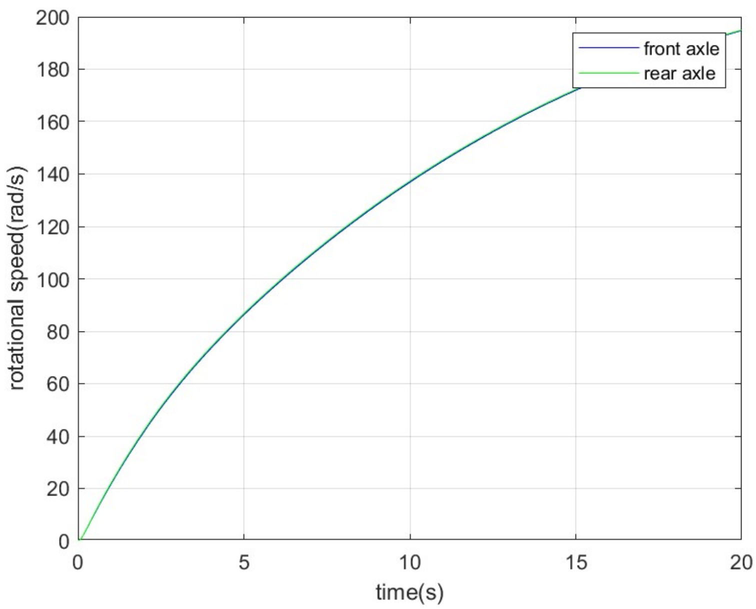


Figure 4-7: rotational speed front and rear axles

For the ideal open differential, the torque split ratiio is 50:50. And no speed difference between the front and rear axles occures in the whole process.

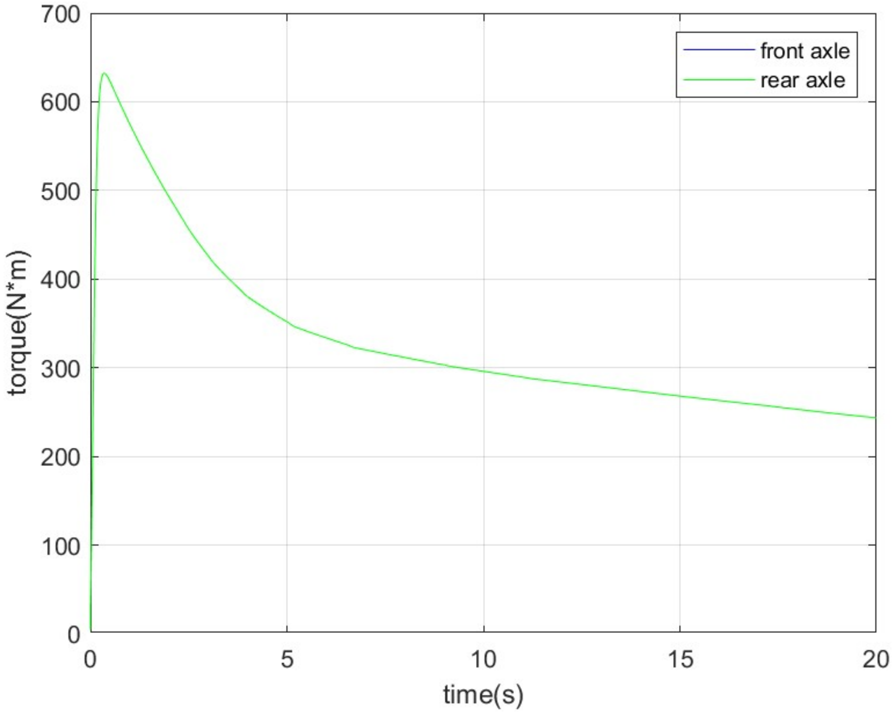


Figure 4-8: Outout torque of front and rear axles

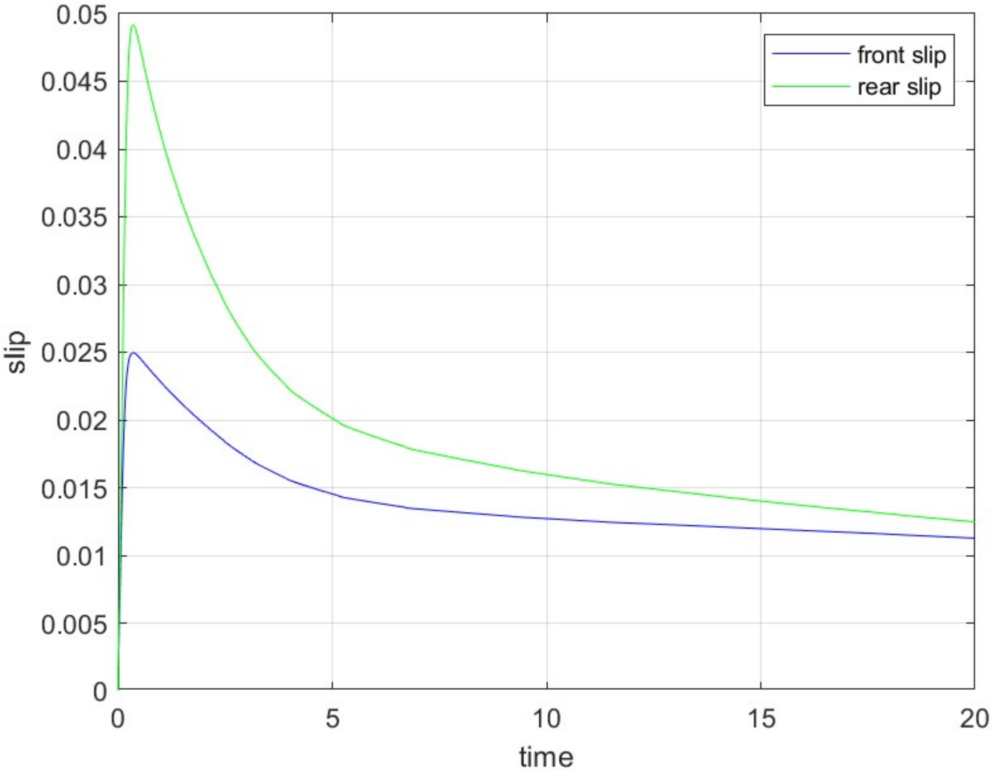


Figure4-9: Slips of front and rear wheels for ideal open differential

Open differential(η=83%)

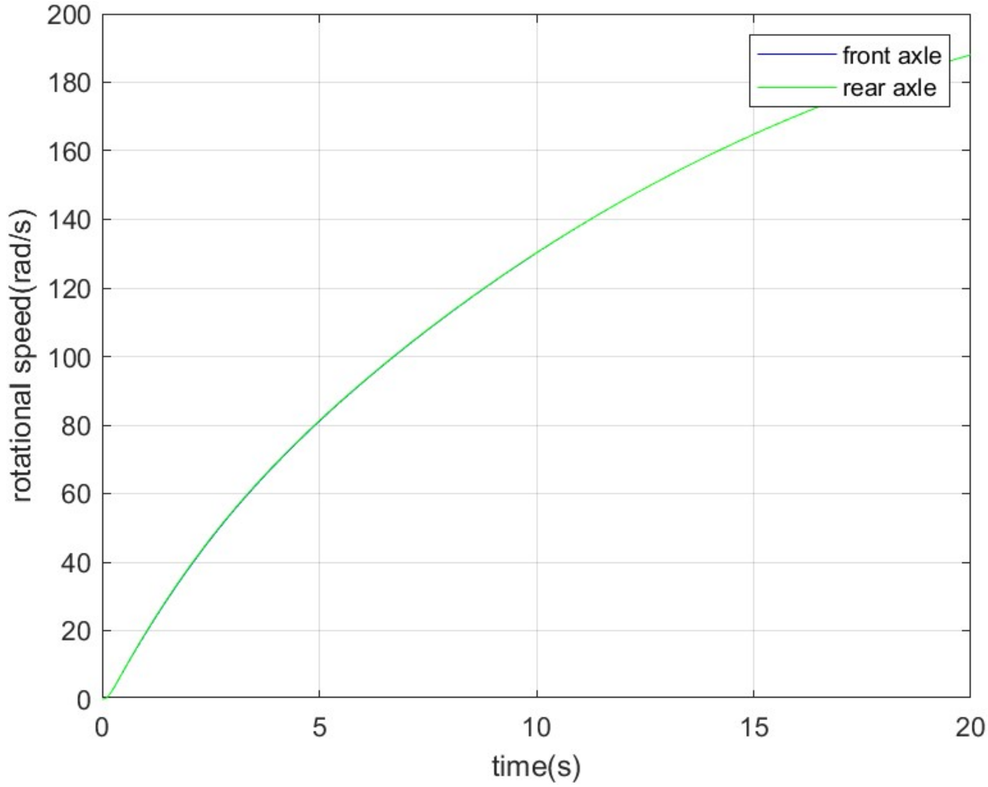


Figure 4-10: rotational speed of front and rear axles

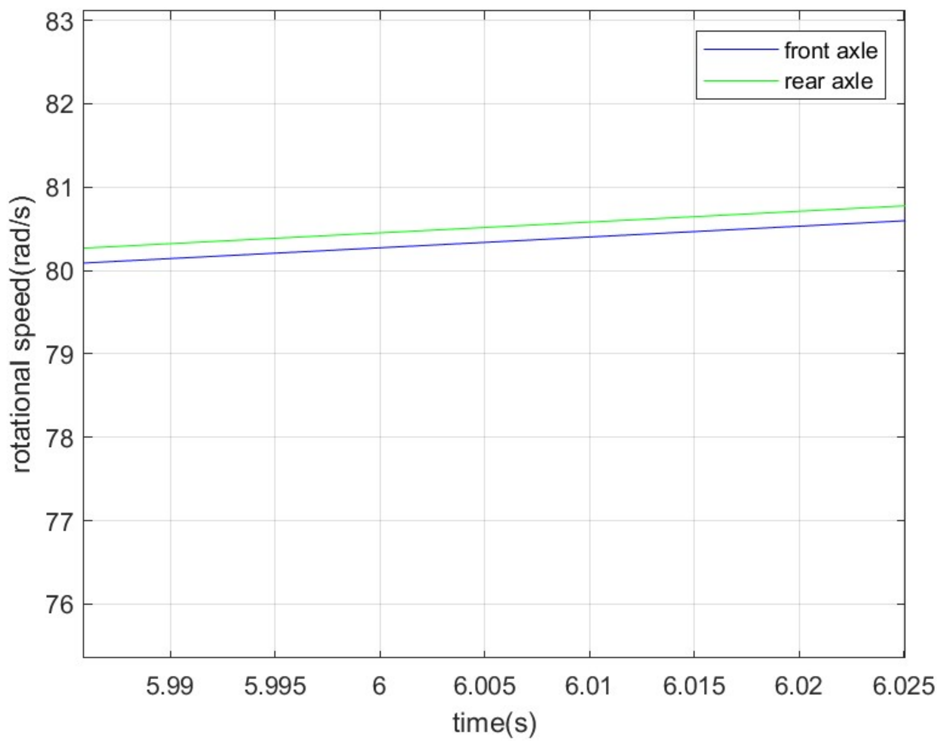


Figure 4-11: Partial enlargement of Figure 4-9

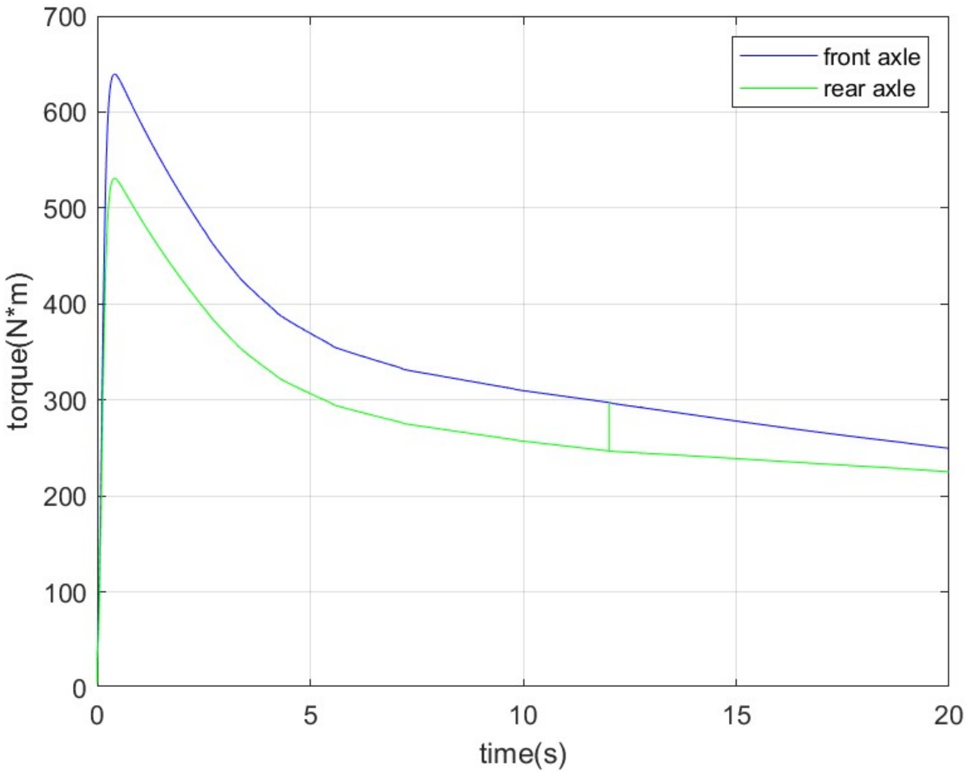


Figure 4-12: outout torque of front and rear axles

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.

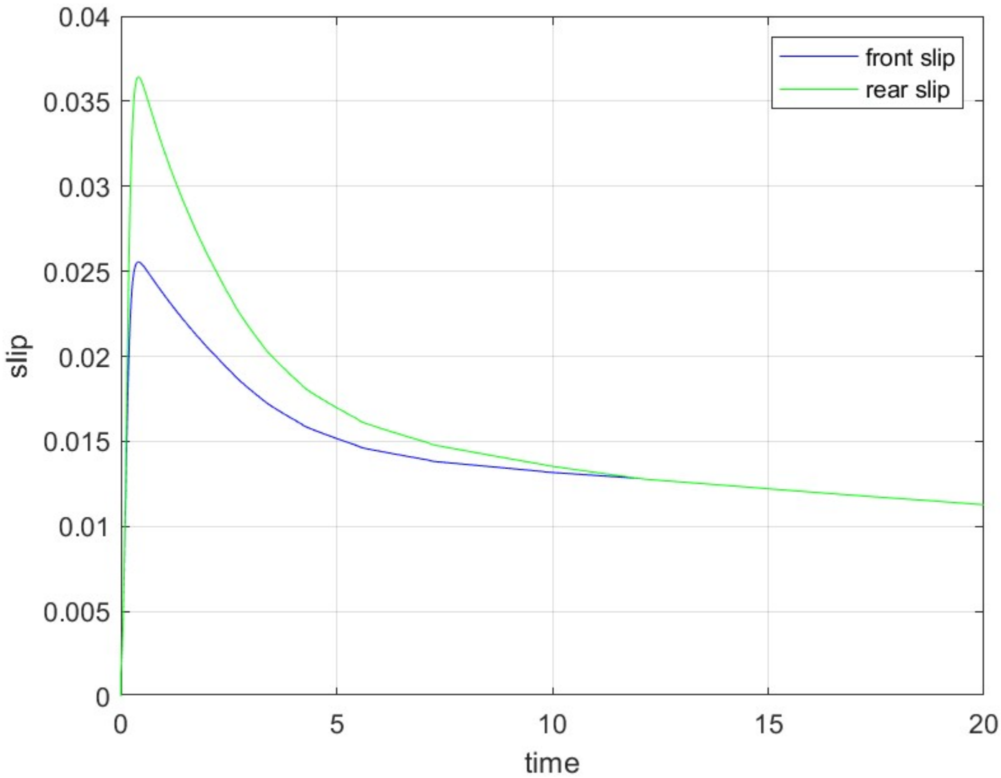


Figure4-13: Slips of front and rear wheels for real open differential

Torsen differential

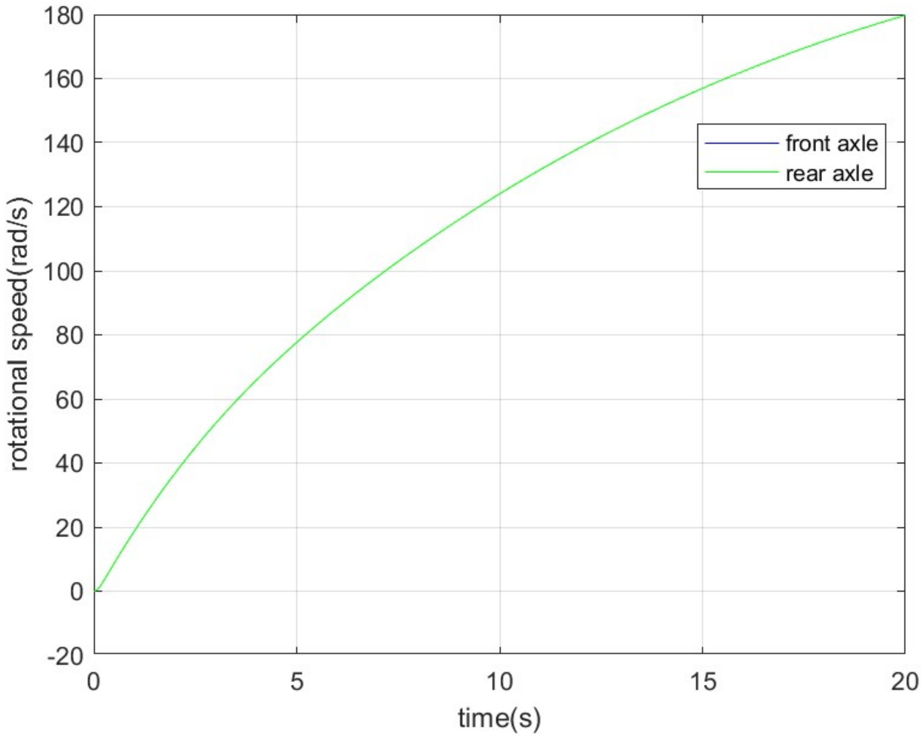


Figure 4-14: rotational speed of front and rear axles

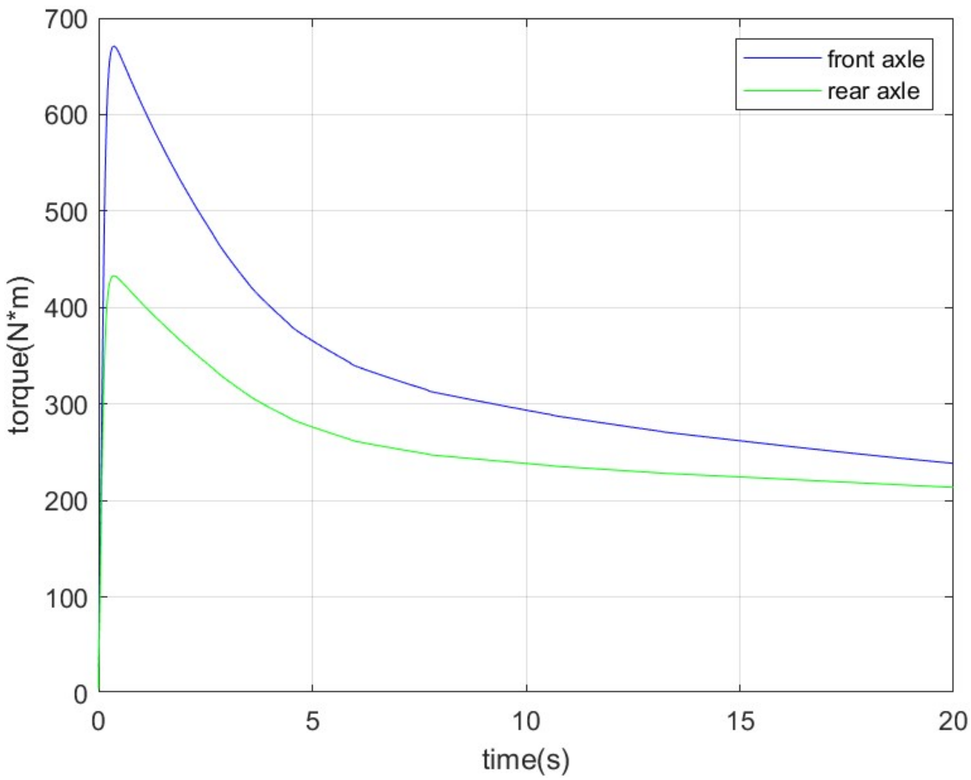


Figure 4-15: outout torque of front and rear axles

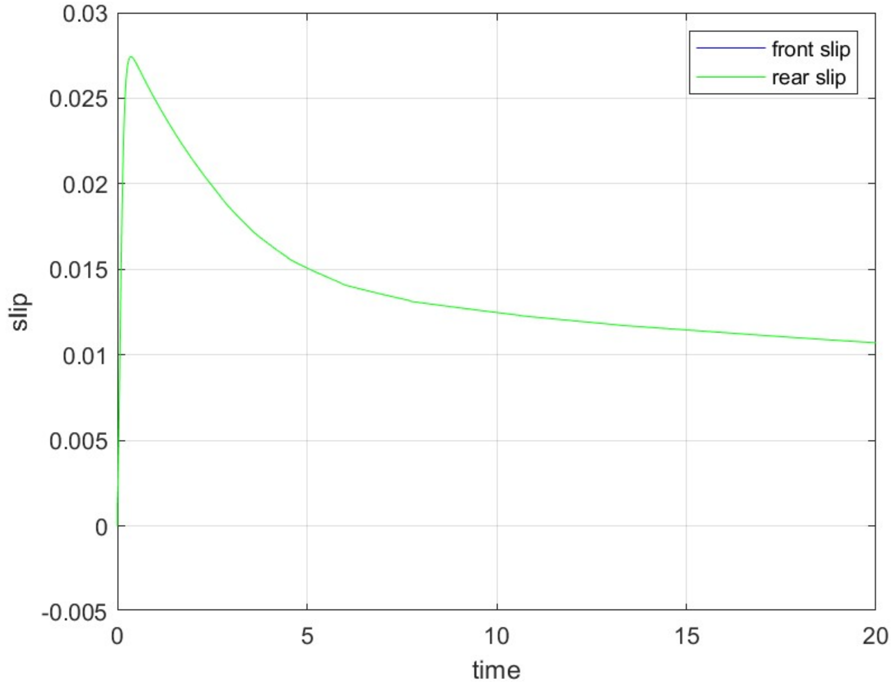


Figure4-16: Slips of front and rear wheels for TCD

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

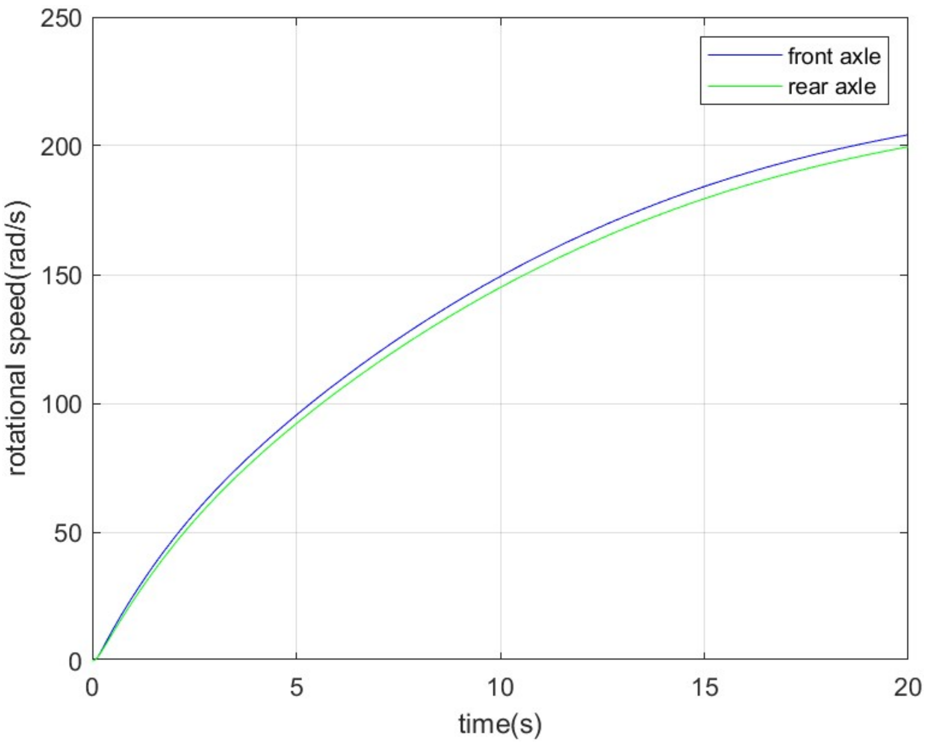


Figure 4-17: rotational speed of front and rear axles

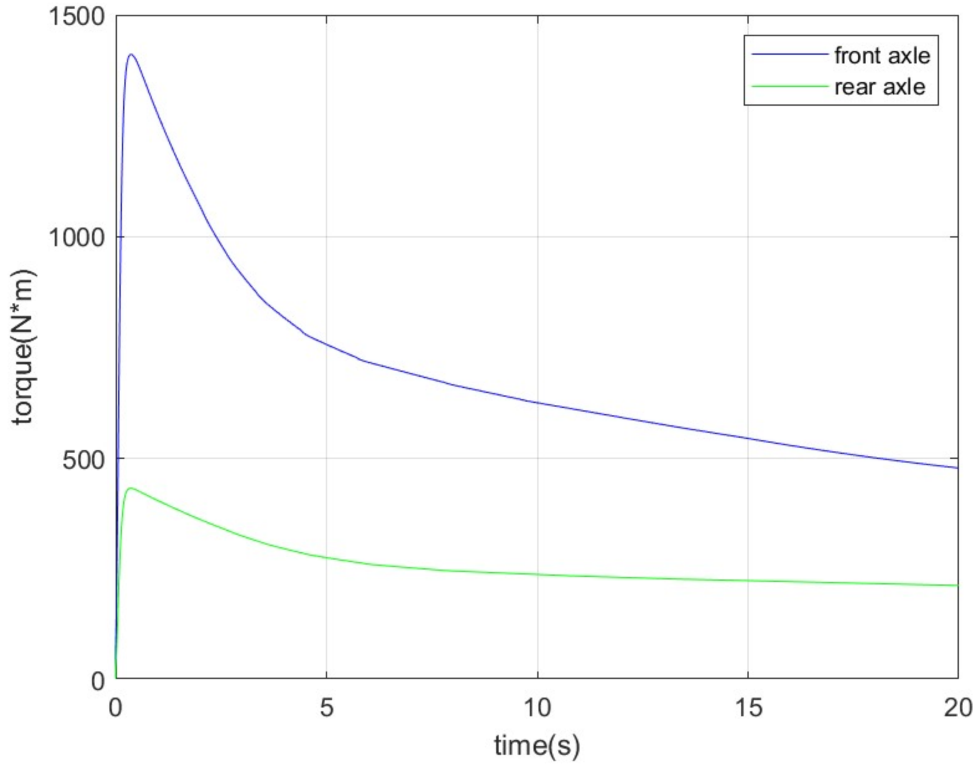


Figure 4-18: outout torque of front and rear axles

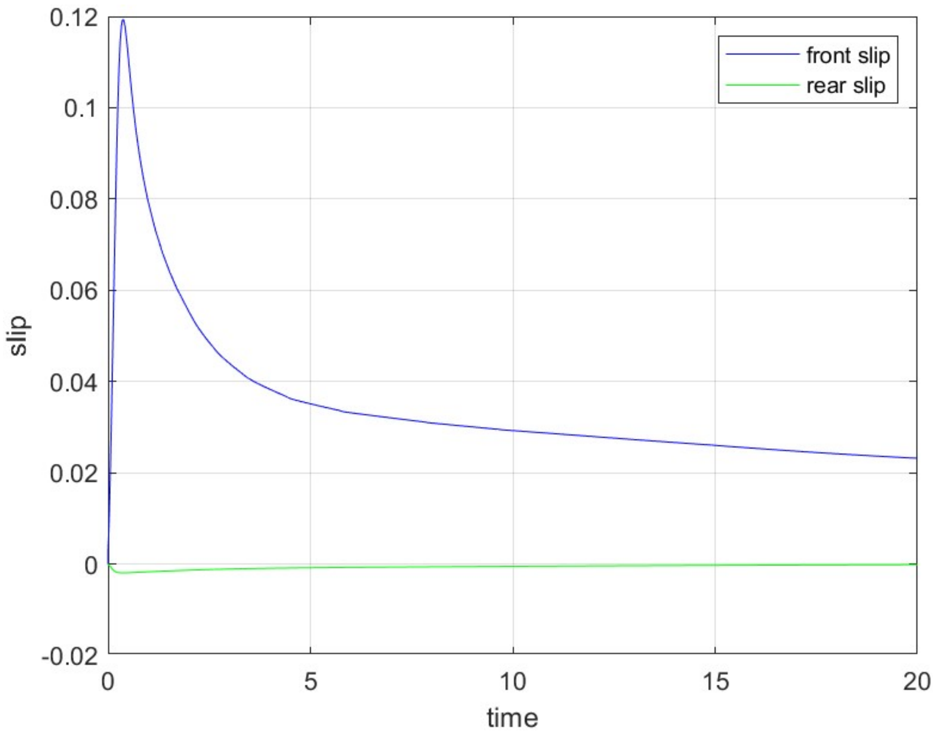


Figure4-19: Slips of front and rear wheels for viscous coupling

Tire performances

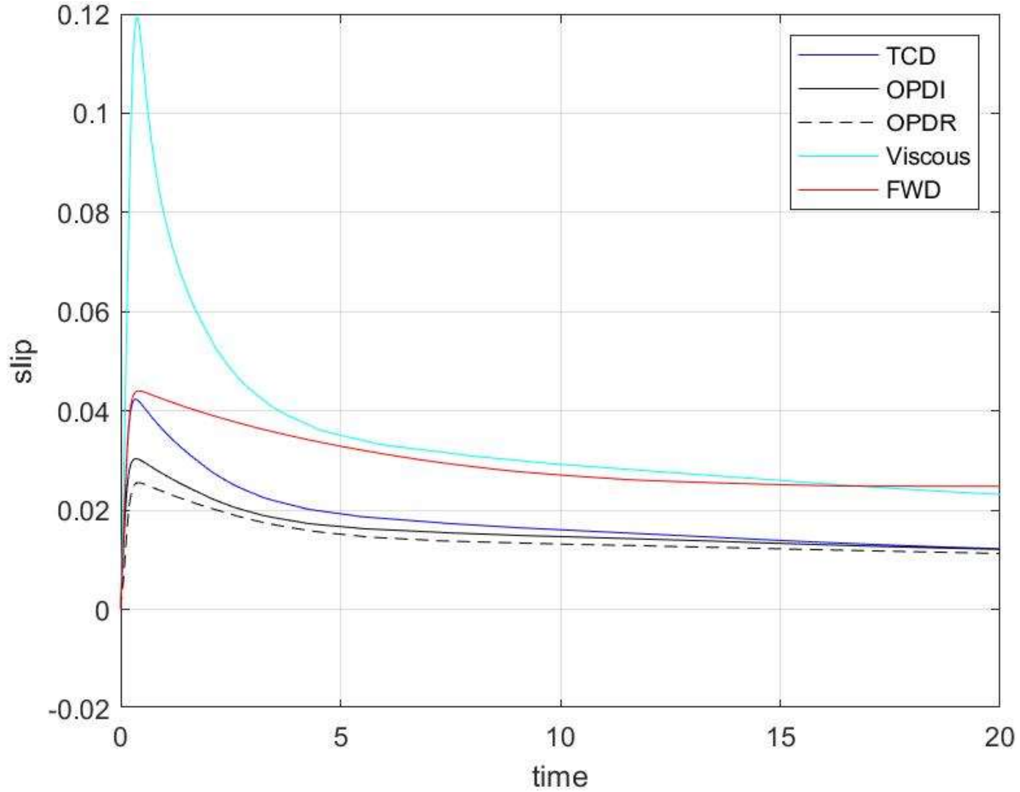


Figure4-20: Slips of front wheels

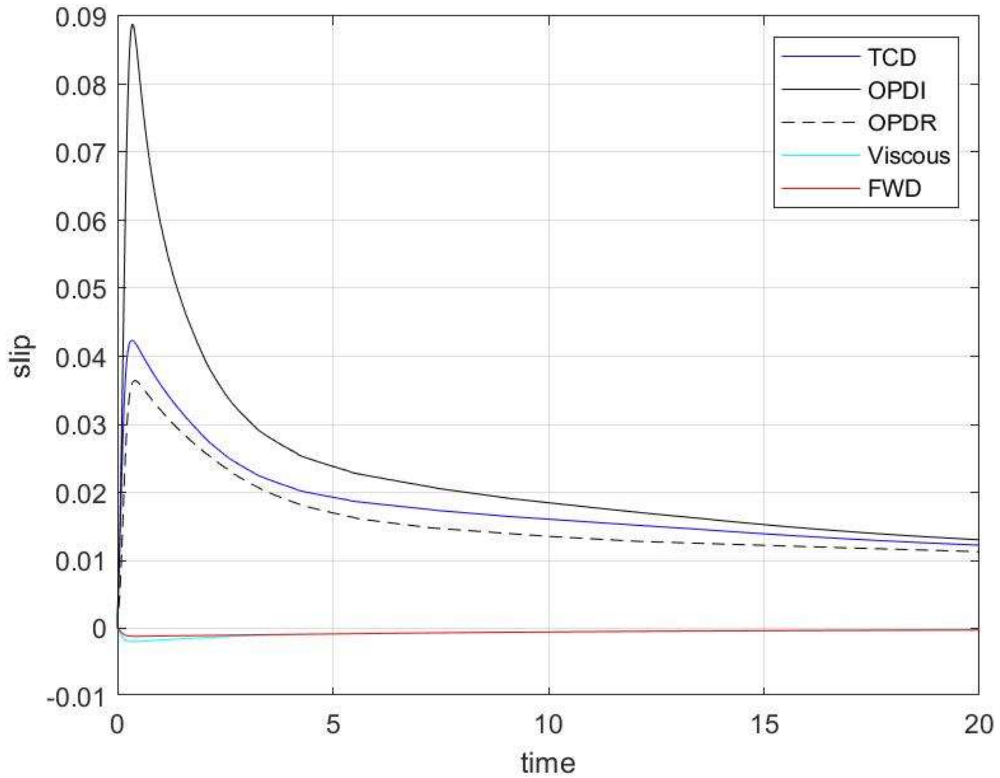


Figure4-21: Slips of rear wheels

4.2.3 VEHICLE PREFORMANCE

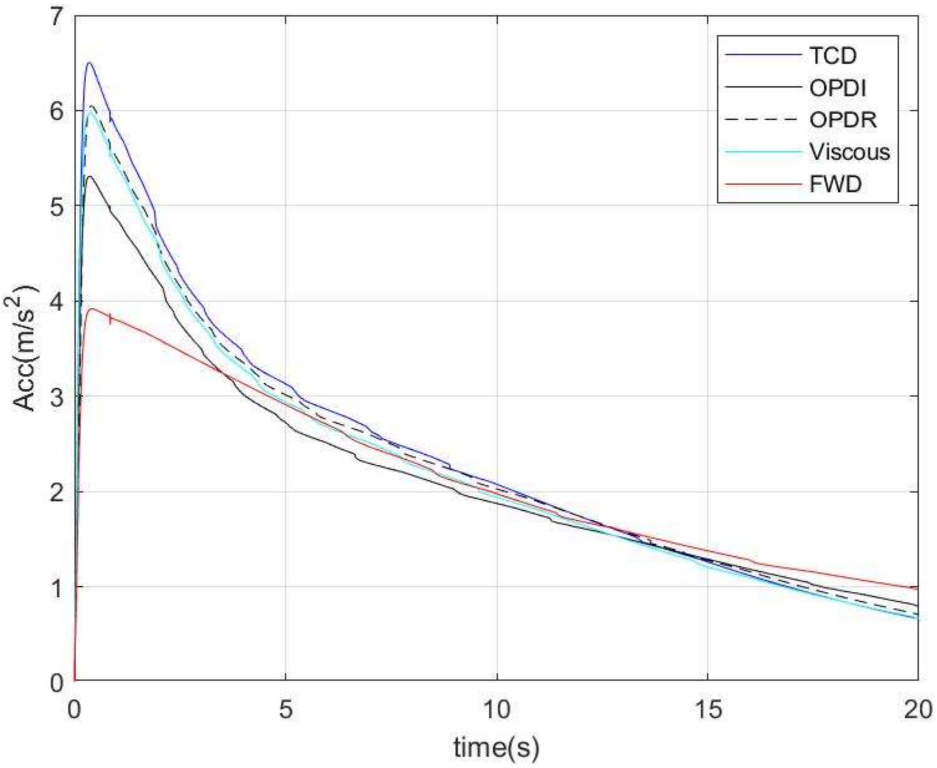


Figure 4-22: The vehicle accelrations of different transfer cases

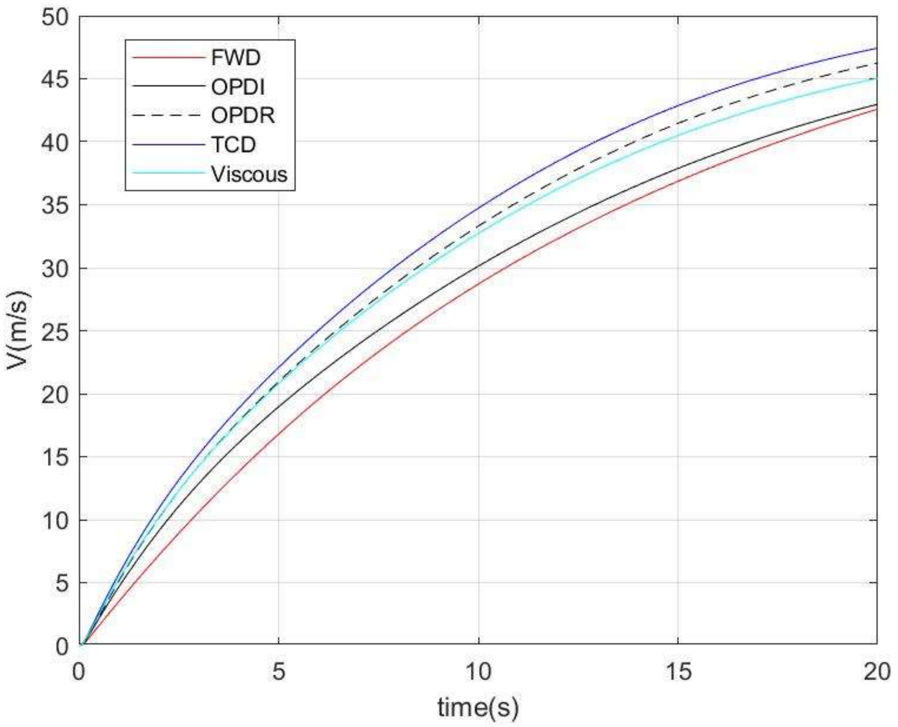


Figure 4-23: The vehicle speeds of different transfer cases

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 andtorsen 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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