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Наш знак:	СПИС-382	28.04.2025
Ваш знак:		

Искане за предварително проучване за новост на
изобретение с наименование
„Универсална автономна електро-гравитационна система за превантивно
механо-динамично противодействие на центробежната сила на ремаркета в завои“

Патентното ведомство Ви уведомява, че във връзка с Ваше искане № СПИС-382 от 17.04.2025г. е извършено проучване въз основа на:

- ☒ описанието, претенциите
- ☒ приложените фигури

Класове по МПК: B60T7/20; G06F7/00; B62D63/06; B60D1/01; B60D1/06
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Бази-данни: IPAS; Esp@cenet,

При проучването са намерени следните документи, посочени в приложения доклад:

Патентни документи:

US2024010036 (A1), US2023356707 (A1), US6424907

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СТАНОВИЩЕ

ОТНОСНО: предварително проучване за новост на
изобретение с наименование

„Универсална автономна електро-гравитационна система за превантивно
механо-динамично противодействие на центробежната сила на ремаркета в завои“

вх. № СпИС-382/17.04.2025 г.

Проучването е направено въз основа на представените 31 патентни претенции
описание и приложенияте към него фигури.

Обект на настоящото проучване е универсална автономна електро-
гравитационна система за превантивно механо-динамично противодействие на
центробежната сила на ремаркета в завои.

Съгласно независима патентна претенция 1, универсалната автономна
електро-гравитационна система за превантивно механо-динамично противодействие
на центробежната сила на ремаркета в завои, включва: теглещо превозно средство,
притежаващо: задна броня (1), механизъм за теглич на ремарке (2) с топка за теглич на
ремарке (3); ремарке (4), теглич на ремарке (5), лапа на теглича (6), система от
датчици, безжично свързани с интелигентен управленски компютърен блок (7), като
има баластен акумулатор (8), пързаящ се, възвратно постъпателно, напречно на
надлъжната ос на ремаркетото на ролкови колелца (9), движещи се по двустранно
наклонящи се успоредни релси (10), едностранно кинематично свързани с наклонящ ги
електрически механизъм (11), чийто електрически захранващ входен кабел е свързан,
през първо изпълнително устройство (12), с първи извод от баластния акумулатор (8),
втори електрически извод от баластния акумулатор (8), през второ изпълнително
устройство (13), е свързан към електрически вход на кордов реверсивен електрически
мотор (14), първи броневи механичен захват (15) е кинематично свързан с първа корда
(16), която през първа обратна кордова ролка (17) е кинематично свързана към
първата страна на баластния акумулатор (8), втори броневи механичен захват (18) е
кинематично свързан с втора корда (19), която през втора обратна кордова ролка (20),
е кинематично свързана към втората страна на баластния акумулатор (8), първи 15
команден извод от интелигентния управленски компютърен блок (7) е кабелно свързан
с първото изпълнително устройство (12), втори команден извод от интелигентния
управленски компютърен блок (7) е кабелно свързан с второто изпълнително
устройство (13), при което първи безжичен сигнален вход на интелигентния
управленски компютърен блок (7) е свързан със сигнален извод от диференциален

дистанционен датчик (21), втори безжичен сигнален вход на интелигентния управленски компютърен блок (7) е свързан със сигнален извод от скоростен датчик (22), трети безжичен сигнален вход на интелигентния управленски компютърен блок (7) е свързан със сигнален извод от инерционен датчик (23), а зареждащ вход на баластния акумулатор (8), през зареждащ контролер (24), е кабелно свързан с общ извод от фотоволтаични панели (25,26), монтирани на покривите, съответно на ремаркетото (4) и на теглещо превозно средство, като към надлъжните ръбове на всеки фотоволтаик, успоредни, съответно на надлъжните оси на ремаркетото (4) и на теглещо превозно средство, са шарнирно свързани огледални рефлектори (27), а интелигентният управленски компютърен блок (7) е конфигуриран за програмиране със самообучаващ се изкуствен интелект и е интегриран с хардуерен блок за архивна памет.

Съгласно зависима претенция 2, баластният акумулатор (8) е монтиран в пързяляща се количка (28) на ролкови колелца (9) по наклонени релси (10).

Съгласно зависима претенция 3, баластният акумулатор (8) е със странични механични захвати (29), за които шарнирно са свързани краищата на съответните първа корда (16) и втора корда (19).

Съгласно зависима претенция 4, всяко ролково колелце (9) е с вътрешен страничен борд (30).

Съгласно зависима претенция 5, характеризираща се с това, че ремаркетото (4) е с повече от една ос (31).

Съгласно зависима претенция 6, осите (31) са със зависимо окачване към пода на (4).

Съгласно зависима претенция 7, осите (31) са с независимо окачване към пода на ремаркетото (4).

Съгласно зависима претенция 8, кабелните връзки (32) между зареждащия контролер (24) и баластния акумулатор (8) са гъвкави.

Съгласно зависима претенция 9, електрическият механизъм (11) и кордовият реверсивен електрически мотор (14), са безчеткови електросилови машини.

Съгласно зависима претенция 10, фотоволтаичните панели (25,26), монтирани на покривите, съответно на ремаркетото (4) и на теглещо превозно средство, са на антивибрационни тампони.

Съгласно зависима претенция 11, шарнирно свързаните огледални рефлектори (27) към фотоволтаичните панели (25,26) са с регулируеми наклони, спрямо плоскостите на шарнирно свързаните огледални рефлектори (27), които са сгъваеми.

Съгласно зависима претенция 12, шарнирно свързаните огледални рефлектори (27) към фотоволтаичните панели (25,26) са рециклируеми полимерни огледала.

Съгласно зависима претенция 13, наклонените релси (10) са подвижно монтирани върху теглича на ремаркетото (5).

Съгласно зависима претенция 14, първата корда (16) и втората корда (19) са от нееластичен гъвкав материал.

Съгласно зависима претенция 15, колелата на ремаркетото (4) са с широки протектори.

Съгласно зависима претенция 16, колелата на ремаркетото (4) са самонапомпващи се надуваеми.

Съгласно зависима претенция 17, колелата на ремаркетото (4) са от плътни еластомери.

Съгласно зависима претенция 18, механизмът за теглича на ремаркетото 18 (2) е подвижно монтиран към задната част на шасито на теглещо превозно средство.

Съгласно зависима претенция 19, задната броня (1) е монтирана към задната част на шасито на теглещото превозно средство, чрез хоризонтални амортизатори (34).

Съгласно зависима претенция 20, наклонящият електрически механизъм (11) включва кинематично необратим редуктор.

Съгласно зависима претенция 21, кордовият реверсивен електрически мотор (14) включва кинематично обратим редуктор.

Съгласно зависима претенция 22, на теглича на ремаркетото (4) е монтиран поне един шарнир (35), към който е фиксиран горният край на един еднораменен лост (36), на чийто срещуположен край лагерува едно колело (37) на хоризонтален вал.

Съгласно зависима претенция 23, между едно раменния лост (36) и теглича на ремаркетото (4) е монтиран поне един пневматичен амортизатор с вграден еластичен елемент (38).

Съгласно зависима претенция 24, между еднораменния лост (36) теглича на ремаркетото (4) е монтиран поне един торсионен гумен амортизатор (38).

Съгласно зависима претенция 25, размерите на колела (37, 39, 40, 41 и 42) са еднакви.

Съгласно зависима претенция 26, всеки шарнир (35) лагерува около един наклонен вал (43).

Съгласно зависима претенция 27, колелите на всички оси на ремаркетото (4) са равни помежду си.

Съгласно зависима претенция 28, колелата на предната ос на ремаркетото (4) е по-широка от останалите колела.

Съгласно зависима претенция 29, колелата на задната ос (39 и 42) на ремаркетото (4) са двигателни.

Съгласно зависима претенция 30, колелата на предната ос на ремаркетото (4) са завиващи.

Съгласно зависима претенция 31, колелата на всички оси на ремаркетото (4) са завиващи.

Обект на настоящото проучване е да се обезпечи универсална автономна електрогравитационна система за превантивно механо-динамично противодействие на центробежната сила на ремаркета в завои, която радикално да преодолее недостатъците на разгледаните системи и ефективно да стабилизира безопасното движение на ремаркетата, при което е задължително ремаркетото да може надеждно и ефективно да балансира в завои по чисто кинематичен начин, като ползва стандартния кормилен усилвател на теглещото превозно средство.

В патентен документ US2024010036 (A1) е разкрита системата за контрол на гравитацията на ремаркетото за превозно средство включваща: задвижващ механизъм за предно ляво активно окачване, свързан с предно ляво колело на превозното средство; преден десен задвижващ механизъм за активно окачване, свързан с предно дясно колело на превозното средство; активатор на задно ляво активно окачване, свързан със задно ляво колело на превозното средство; активатор на задно дясно активно окачване, свързан със задно дясно колело на превозното средство; модул за управление на задвижващия механизъм, който е конфигуриран: въз основа на силите

на теглича при теглича на ремаркетото на превозното средство в надлъжна, странична и вертикална посока, да определя целевите вертикални сили за активатора на предното ляво активно окачване, предното дясно активно окачване задвижващ механизъм, задвижващият механизъм на задното ляво активно окачване и съответно задвижващият механизъм на задното дясно активно окачване; и селективно да регулира на предния ляв и десен задвижващ механизъм на активното окачване, задния ляв и десен задвижващ механизъм на активното окачване въз основа съответно на целевите вертикални сили; модул за управление на задвижващия механизъм, конфигуриран да определя целевите вертикални сили на базата на минимизиране на разликите между силата на теглича и референтната сила на теглича; модул за еталонни сили на теглича, конфигуриран да задава референтните стойности на силата на теглича; модул за управление на задвижващия механизъм, конфигуриран да определя целевите вертикални сили допълнително въз основа на скоростта на отклонение на превозното средство и е конфигуриран да определя целевите вертикални сили на базата на минимизиране на разликата между скоростта на отклонение и референтната скорост на отклонение; модул за еталонна скорост на завъртане, конфигуриран да задава еталонната скорост на завъртане на базата на ъгъл на завиване на превозното средство и да задава еталонната скорост на завъртане допълнително въз основа на скоростта на превозното средство; преден ляв електрически мотор, конфигуриран да извежда въртящ момент към предното ляво колело на превозното средство; преден десен електрически мотор, конфигуриран да извежда въртящ момент към предното дясно колело на превозното средство; заден ляв електрически мотор, конфигуриран да извежда въртящ момент към задното ляво колело на превозното средство; заден десен електрически мотор, конфигуриран да извежда въртящ момент към задното дясно колело на превозното средство; и модул за управление на колело, конфигуриран да: въз основа на първата сила на захващане, втората сила на захващане и третата сила на захващане, определя целевите изходни стойности на въртящия момент съответно на предния ляв електрически мотор, предния десен електрически мотор, задния ляв електрически мотор и задния десен електродвигател; и селективно регулиране на предния ляв електродвигател, предния десен електромотор, задния ляв електромотор и задния десен електромотор въз основа съответно на целевия въртящ момент; модулът за управление на колелото е конфигуриран да определя целевите изходящи въртящи моменти допълнително въз основа на целевите вертикални сили; модулът за управление на задвижващия механизъм е конфигуриран да определя целевите вертикални сили допълнително въз основа на целевия изходен момент на въртящия момент.

В патентен документ US2023356707 (A1) са разкрити метод и система, като системата включва: превозно средство, което има електрическо задвижвано ремарке, теглено към него; контролер, програмиран да задвижва превозното средство с ремаркетото, теглено към него, който управлява задвижването на превозното средство и на ремаркетото. Контролерът е допълнително програмиран да управлява компонентите на силовото предаване на превозното средство и на ремаркетото, да получава данни за загуба на компоненти на задвижването на превозното средство и на ремаркетото; картата на ефективността, свързана с факторите на задвижването на превозното средство в данните за загубите на компонентите на задвижването на превозното средство и картата на ефективността, свързана с факторите на задвижването на ремаркетото, в данните за загубите на компонентите на задвижването на ремаркетото;

комбинираната карта на ефективността осигурява работа на задвижването на превозното средство и задвижването на ремаркетото да имат най-висока комбинирана ефективност за дадена комбинация от мощност и скорост на превозното средство, изисквана от водача; силовото предаване на превозното средство включва първа тягова батерия; задвижването на ремаркетото включва втора тягова батерия; задвижването на превозното средство допълнително включва двигател; контролерът е допълнително програмиран да предизвиква прехвърляне на електрическа енергия между първата тягова батерия и втората тягова батерия, за да поддържа работата на задвижването на превозното средство и задвижването на ремаркетото съгласно комбинираната карта на ефективност; контролерът е допълнително програмиран да реализира работата на задвижването на превозното средство и задвижването на ремаркетото чрез векторизиране на въртящия момент между задвижващите оси и източниците на задвижване на задвижването на превозното средство и задвижването на ремаркетото и да генерира комбинирана карта на ефективност при свързване на ремаркетото към превозното средство; ремаркетото е електрифицирано, тъй като задвижването на ремаркетото включва тягова батерия за осигуряване на мощност за задвижване на ремаркетото.

В едно от примерните изпълнения, за разлика от задвижването на превозното средство, задвижването на ремаркетото не включва двигател. Силовото предаване на ремаркетото допълнително включва електрически мотор, който е механично свързан към задвижваща ос, която е механично свързана с колелата на ремаркетото. Тяговата батерия за съхранява електрическа енергия, може да се използва от електрическия мотор за задвижване на ремаркетото. Тяговият акумулатор е електрически свързан към електрическия мотор чрез инвертор на силовото предаване на ремаркетото. Инвертора преобразува постоянен ток от тяговата батерия в променливотоково захранване, за да управлява електрически мотор за задвижване на ремарке.

В патентен документ US6424907 са разкрити системи за измерване на стабилността на превозното средство и по-специално метод и свързано устройство за откриване на промени в центъра на тежестта на масата на превозно средство.

Устройство за откриване на опасност от преобръщане на превозно средство, включва най-малко две оси и най-малко три колела; средство за отчитане на състояние на шофиране при маневра в завой; средство за определяне на вариации в центъра на тежестта на масата; средство за откриване на първи променливи на състоянието, които съответстват на съответното натоварване на колелата върху най-малко две колела; средство за сравняване на откритите променливи на първо състояние с референтни стойности, представителни за съответната маневра на завиване; средство за изчисляване на промяната в центъра на тежестта от разликите между откритите първи променливи на условието и референтните стойности и средство за определяне на втора променлива на условието, представителна за опасността от преобръщане, от изчисленото изменение на центъра на тежестта; средства за измерване на поне един ход на пружината на окачването на колелото, налягането на пружината на окачването на колелото, налягането на амортизатора на амортизатора, налягането на въздуха вътре в гумата, страничната деформация на гумата и използване на такова измерване като основа за първата променлива на условието, която съответства на натоварването на колелото; средство за измерване на поне едно от напречното ускорение, което действа

върху центъра на тежестта на превозното средство, ъгъла на накланяне на превозното средство, ъгъла на наклона на превозното средство, скоростта на превозното средство, ъгъла на завиване, скоростта на отклонение и използване на такова измерване като основа за втората променлива на условието, представителна за опасността от преобръщане; превключвател за крайна позиция, осигурен на амортисьор или върху пружинен елемент на окачването на колелото, точката на превключване на превключвателя за крайна позиция е свързана чрез времева корелация с втората променлива на условието, представителна за опасността от преобръщане.

Анализът на обекта на проучването и известните решения показва следното:

Новост

При извършен сравнителен анализ между обекта, предмет на настоящето проучване и намерените при проучването документи, се установи, че в нито един от намерените документи не е разкрита универсалната автономна електро-гравитационна система за превантивно механо-динамично противодействие на центробежната сила на ремаркета в завои, която да включва всички технически признаци на обекта на проучването.

Следователно, обектът на настоящето проучване, съгласно така формулираните претенции е нов по смисъла на чл. 8 от Закона за патентите и регистрацията на полезните модели (ЗПРПМ).

Ilka
Radoslavova
Stefanova

Digitally signed by
Ilka Radoslavova
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Date: 2025.04.28
17:22:37 +03:00

И. Стефанова
Главен експерт ДЕЗИПМПД



ДОКЛАД ОТ ПРОУЧВАНЕ ПРОУЧВАНЕ ВХ. № СпИС-382/17.04.2025 г.		
I. Уточнен класификационен индекс по МПК/КПК		
МПК: B60T7/20; G06F7/00; B62D63/06; B60D1/01; B60D1/06 КПК: B60T7/20; G06F7/00; B62D63/06; B60D1/01; B60D1/06; B60T8/248		
II. ОБЛАСТ НА ПРОУЧВАНЕ:		
Минимален обхват на проучване		
Вид класификация:	Класификационен индекс:	
МПК/КПК	МПК: B60T7/20; G06F7/00; B62D63/06; B60D1/01; B60D1/06 КПК: B60T7/20; G06F7/00; B62D63/06; B60D1/01; B60D1/06; B60T8/248	
Бази данни, в които е извършено проучването		
Esp@cenet; IPAS; Internet		
III. Релевантни документи		
Категория	Цитиране на документите с посочване на релевантните части	Релевантни претенции
A	US2024010036 (A1), [DRIV AUTOMOTIVE INC [САЩ] ±], публ. на 11.01.2024г.	1-31
A	US2023356707 (A1), [FORD GLOBAL TECH LLC [US] ±], публ. на 09.11.2023г.	1-31
A	US6424907, [CONTINENTAL TEVES AG & CO OHG [DE] ±], публ. на 23.07.2002г.	1-31
Категория на документа		
X – документът е от особено значение; претендираното изобретение няма новост или изобретателска стъпка, когато документът се разглежда самостоятелно Y – документът е от особено значение; претендираното изобретение няма изобретателска стъпка, когато документът се комбинира с един или повече други документи и тази комбинация е очевидна за специалист в областта A – документ, отразяващ нивото на техниката, който не е от особено значение D – документ, цитиран в описанието O – неписмено разкриване, използване, прилагане, докладване или демонстриране P - документ, публикуван преди датата на подаване в България, но след приоритетната дата E – по-рано подадена българска заявка, но публикувана на или след датата на подаване L – документ, който може да предизвика съмнение относно приоритета на претенциите или който е цитиран с цел установяване датата на публикация на друг документ или друга специална причина (както е посочено) & - документ от една патентна фамилия		
Дата: 28.04.2025	Име: И. Стефанова	Ilka Radoslavova Stefanova Digitally signed by Ilka Radoslavova Stefanova Date: 2025.04.28 17:22:26 +03:00 И. Стефанова Главен експерт ДЕЗИПМПД



US 20030033063A1

(19) **United States**(12) **Patent Application Publication****Kawashima et al.**(10) **Pub. No.: US 2003/0033063 A1**(43) **Pub. Date: Feb. 13, 2003**(54) **SUSPENSION CONTROL APPARATUS****Publication Classification**(75) Inventors: **Mitsunori Kawashima**, Wako-shi (JP);
Shinji Suto, Wako-shi (JP); **Hirokazu**
Kitazawa, Wako-shi (JP)(51) **Int. Cl.⁷** **B60G 17/00**
(52) **U.S. Cl.** **701/37; 280/5.5**

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KABUSHIKI KAISHA, Minato-ku (JP)(21) Appl. No.: **10/211,006**(22) Filed: **Aug. 2, 2002**(30) **Foreign Application Priority Data**

Aug. 8, 2001 (JP) 2001-240879

(57) **ABSTRACT**

The invention provides a suspension control system that executes control of actuators that apply a vertical force to the vehicle wheels depending on the travel state of the vehicle, and at the same time, carry out power generation by using the energy of the vertical motion of the wheels and use the obtained energy to drive the actuator motor. In the suspension control apparatus in which the vehicle wheels can receive a force in the vertical direction from a motor, power generation is carried out by the motor depending on the vertical motion of the vehicle wheels, and a storage cycle that stores energy in a storage apparatus and a discharge cycle that drives the motor by the energy stored in the storage apparatus are repeated.

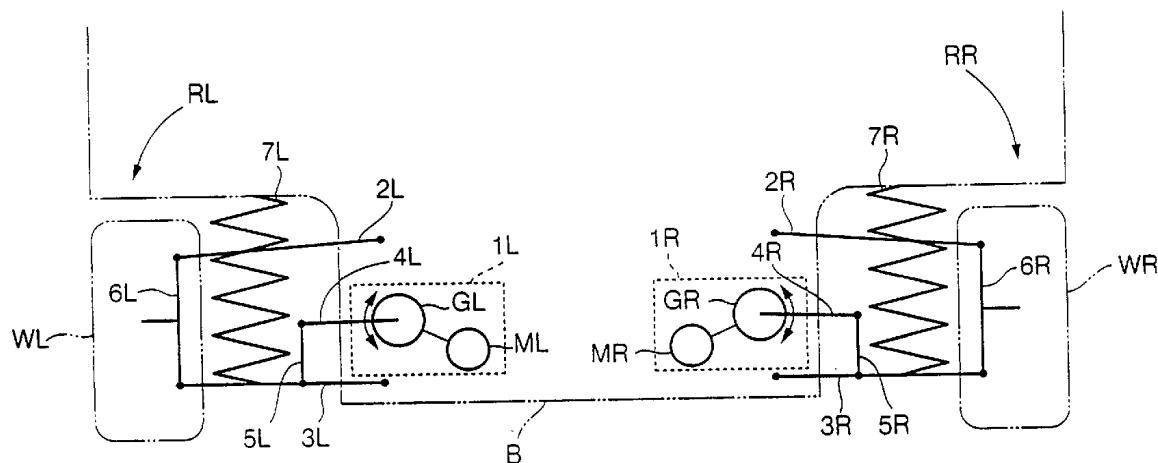


Fig. 1

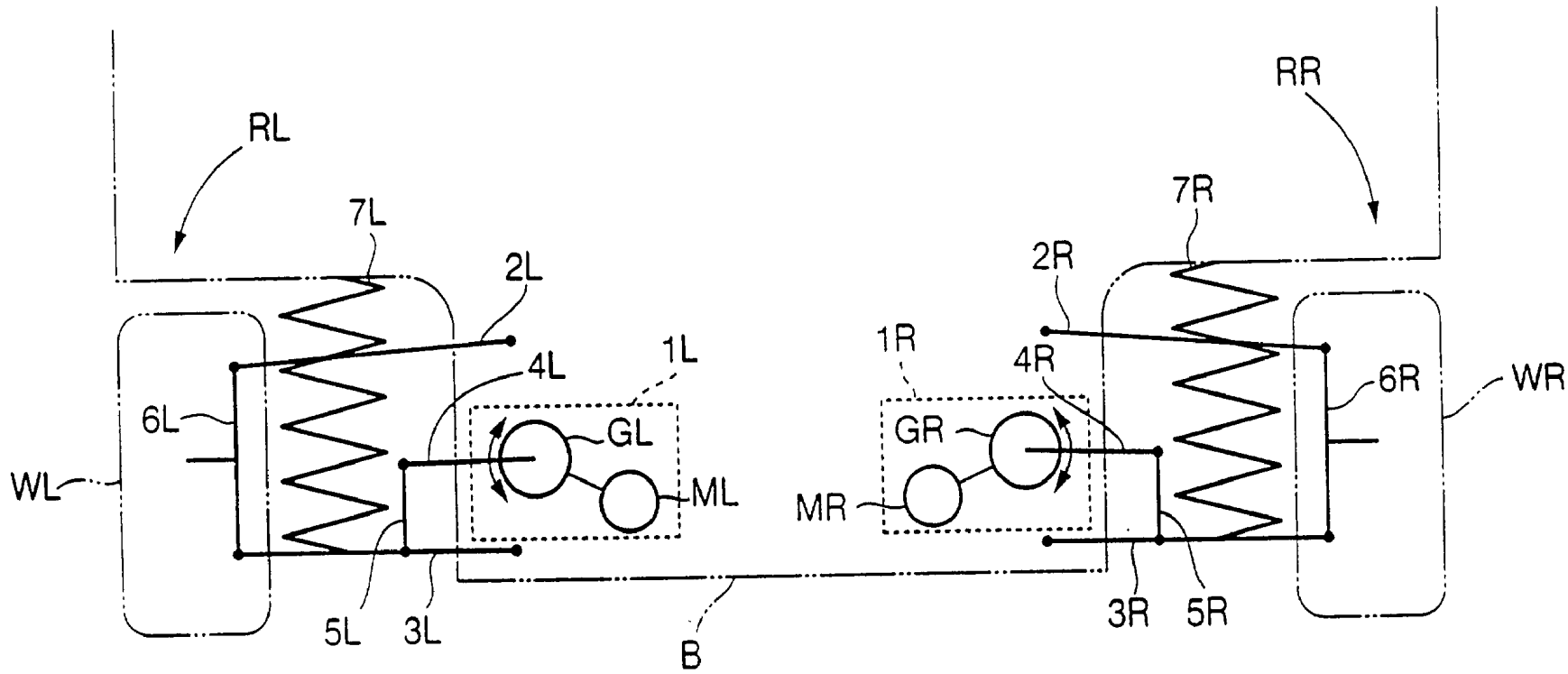


Fig. 2

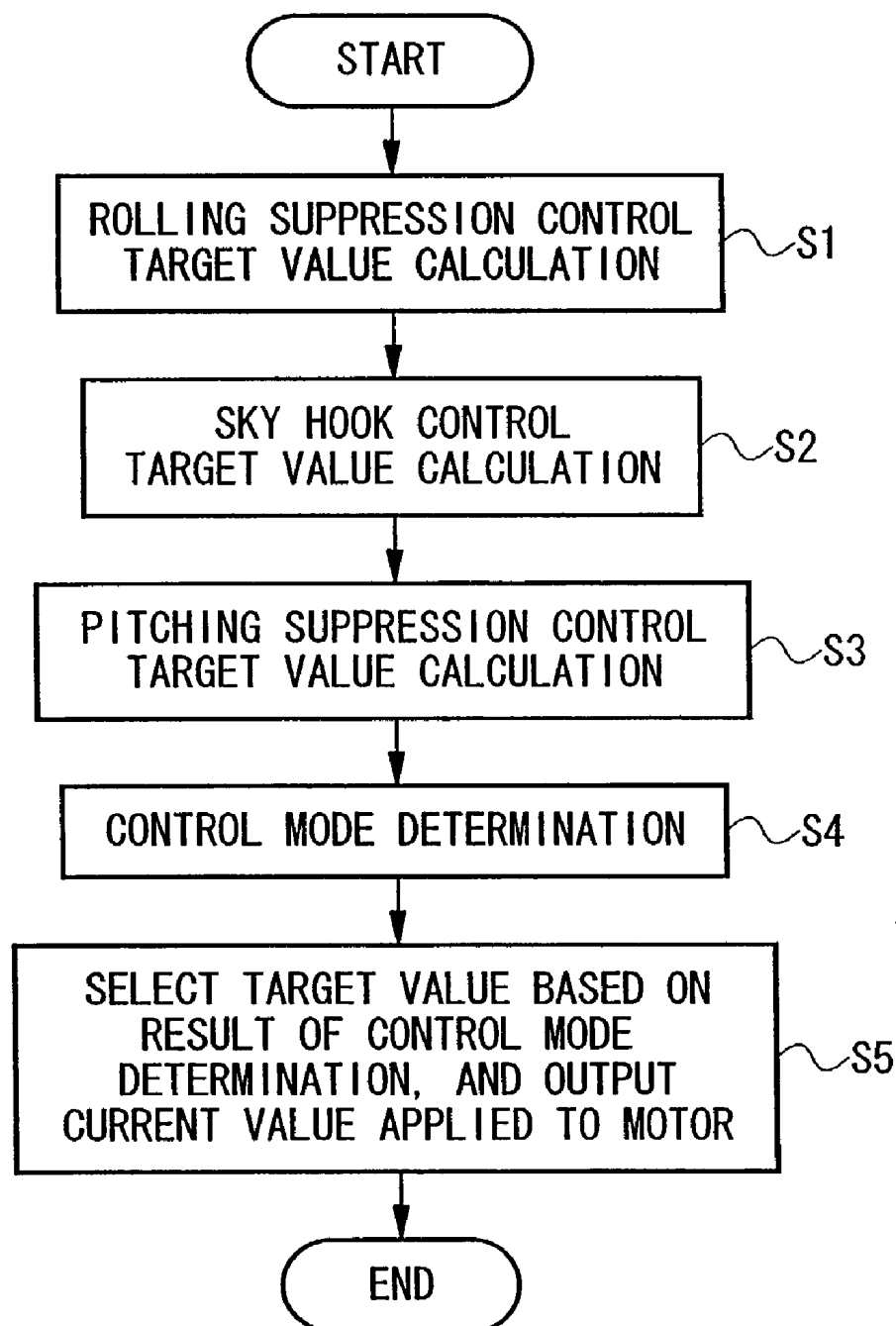


Fig. 3

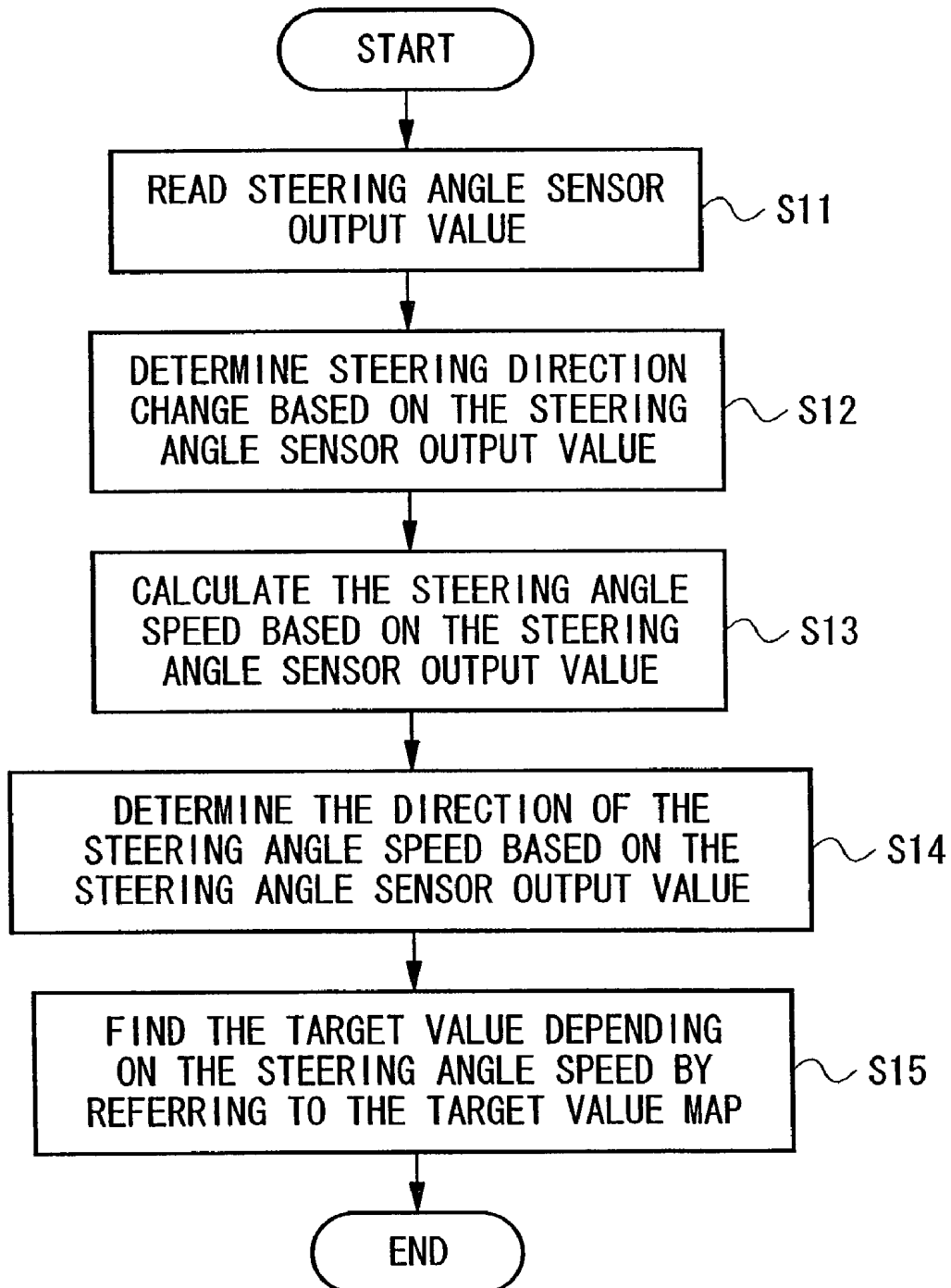


Fig. 4

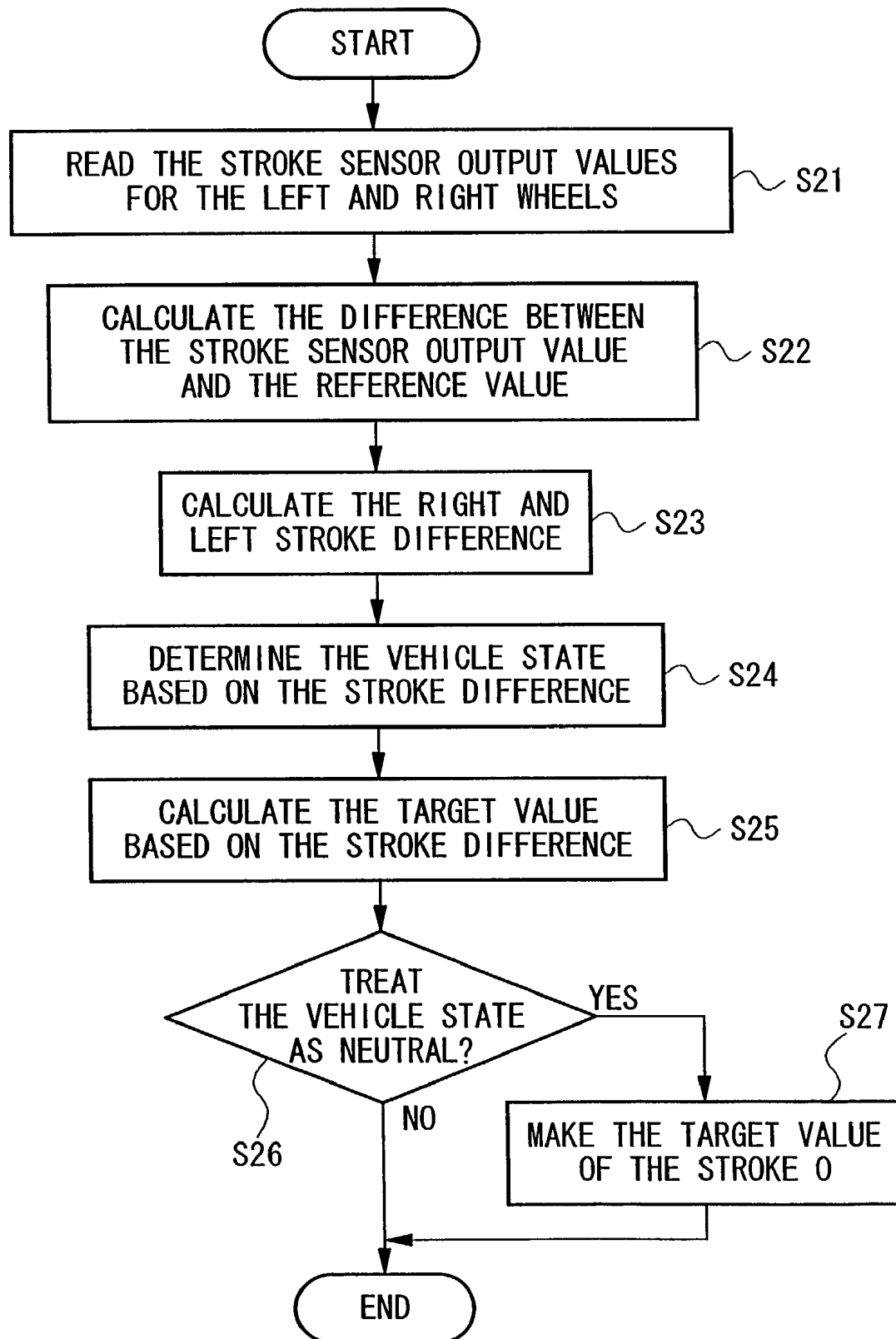


Fig. 5

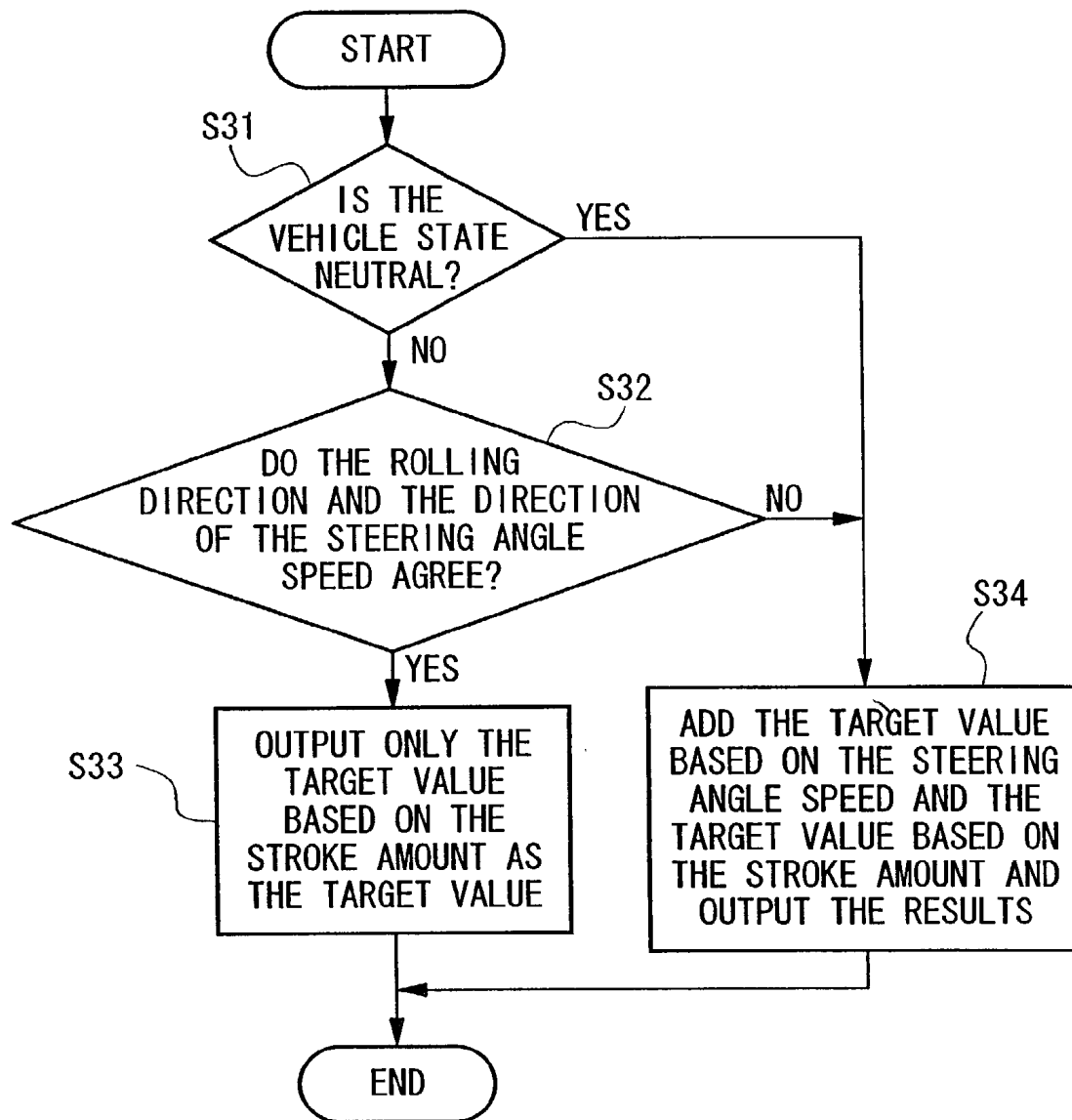


Fig. 6

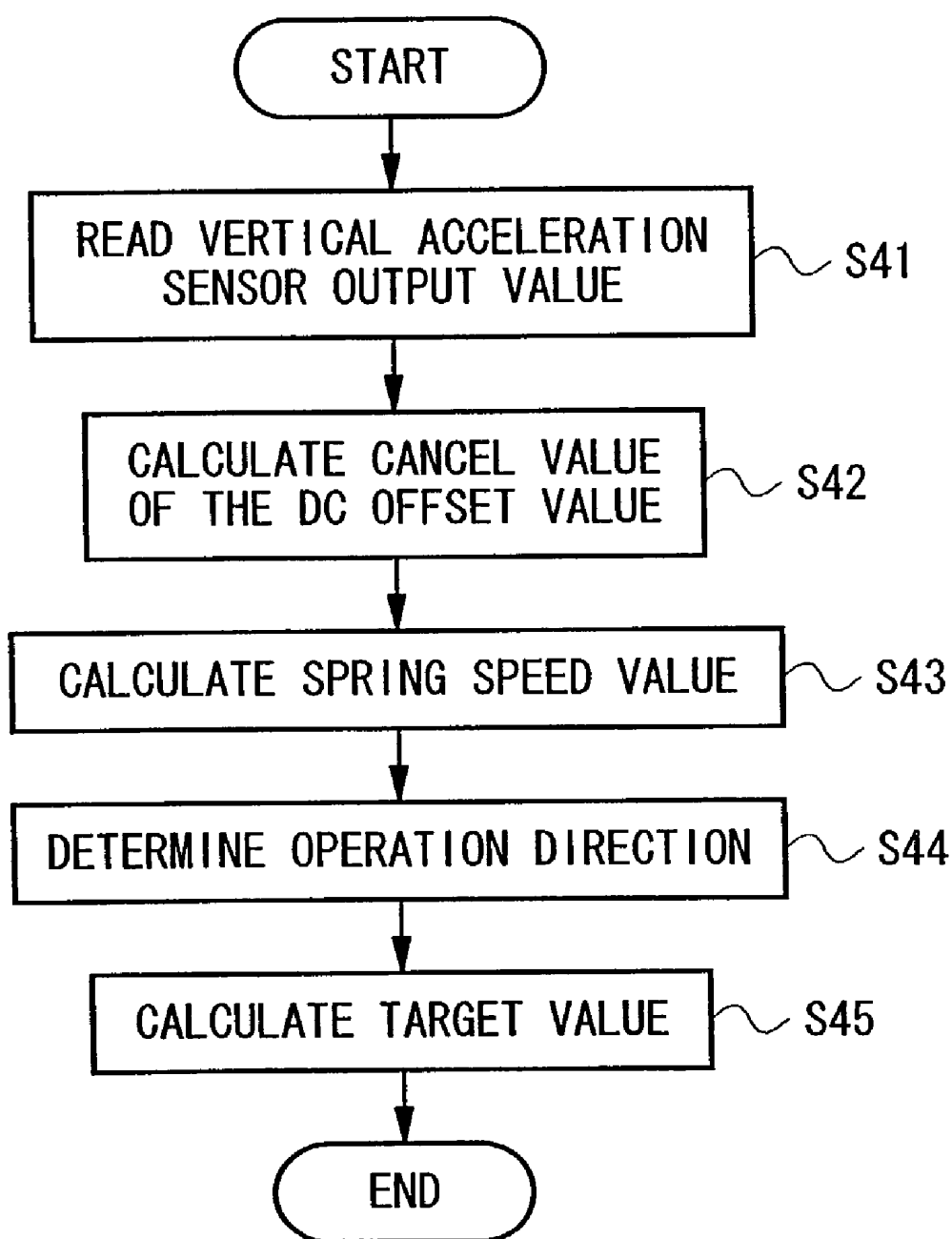


Fig. 7

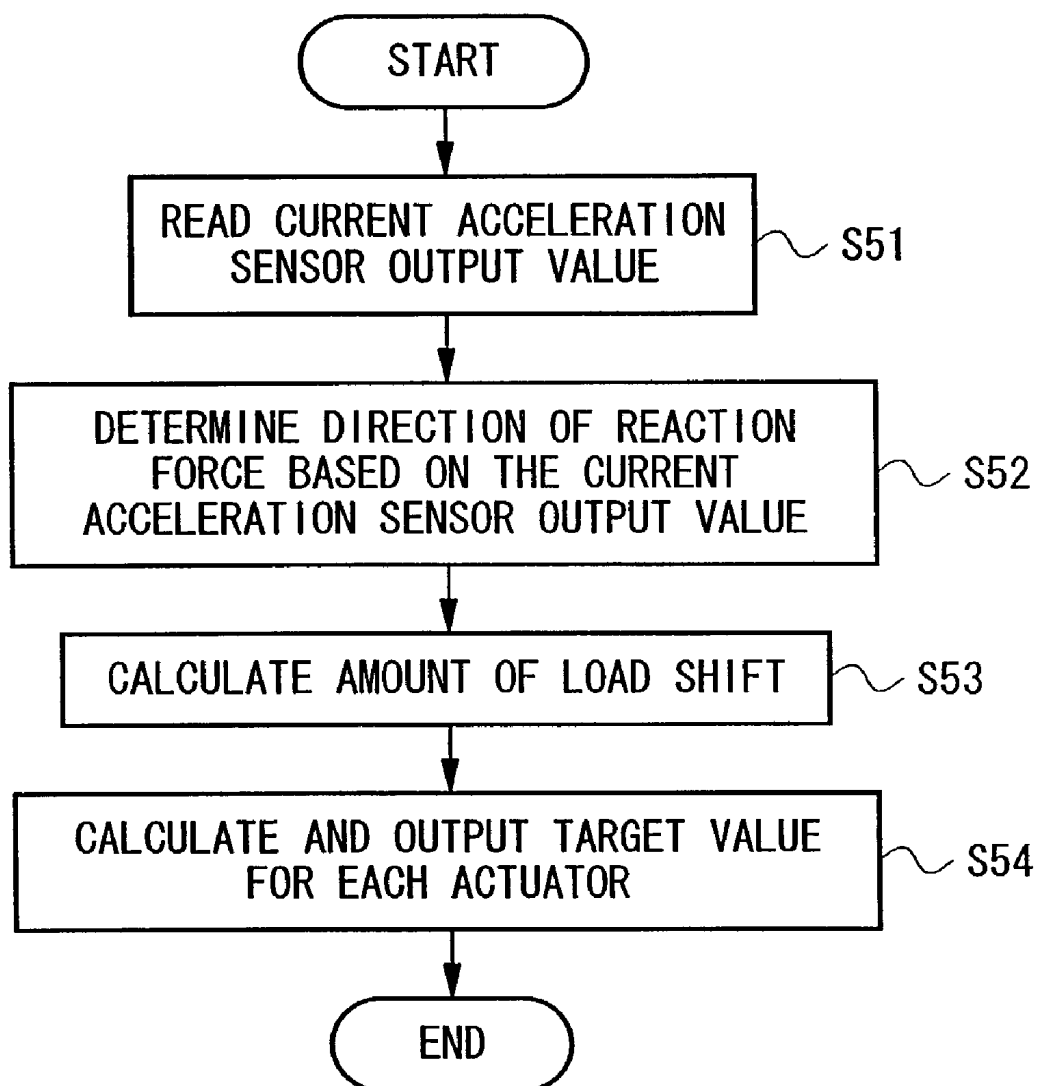


Fig. 8

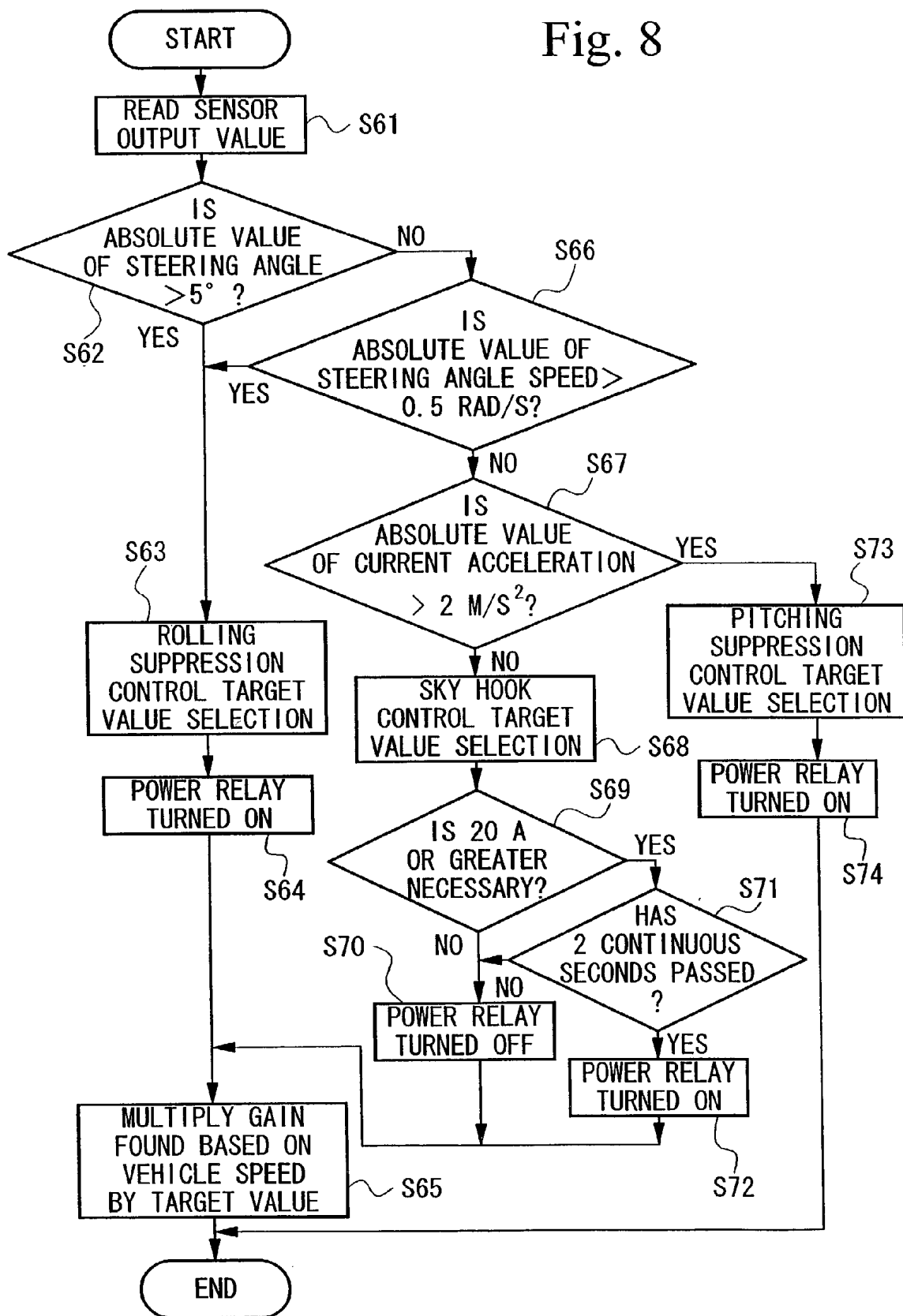


Fig. 9

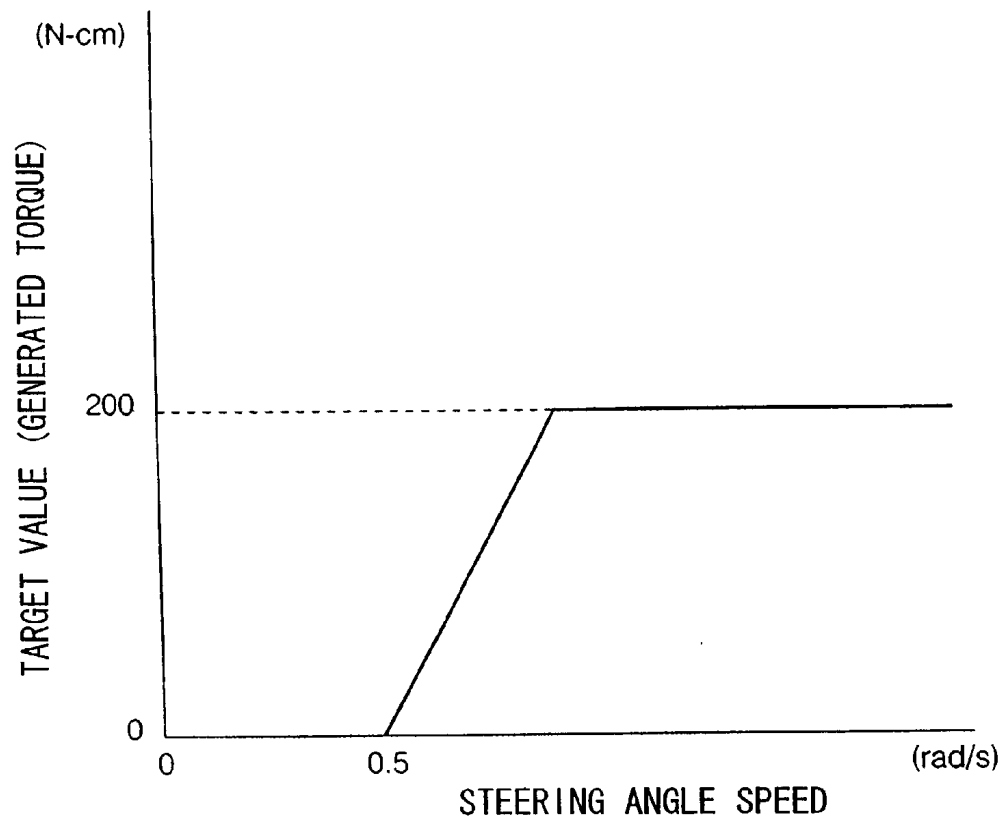


Fig. 10

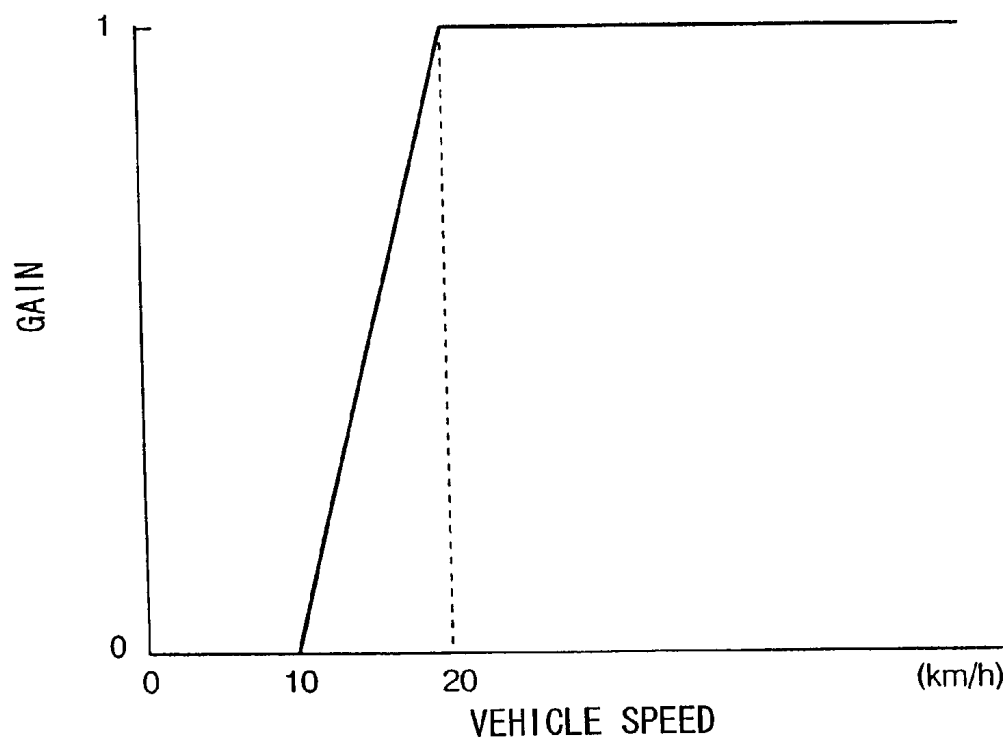


Fig. 11

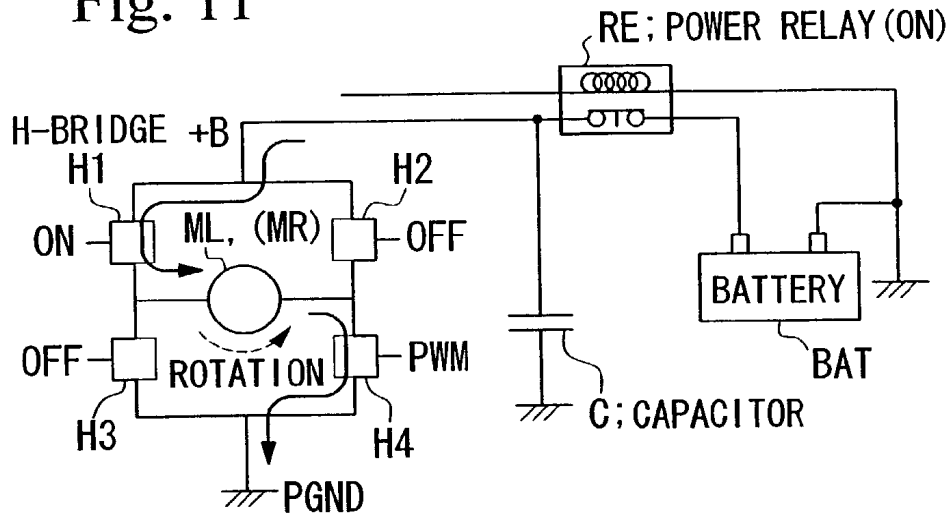


Fig. 12

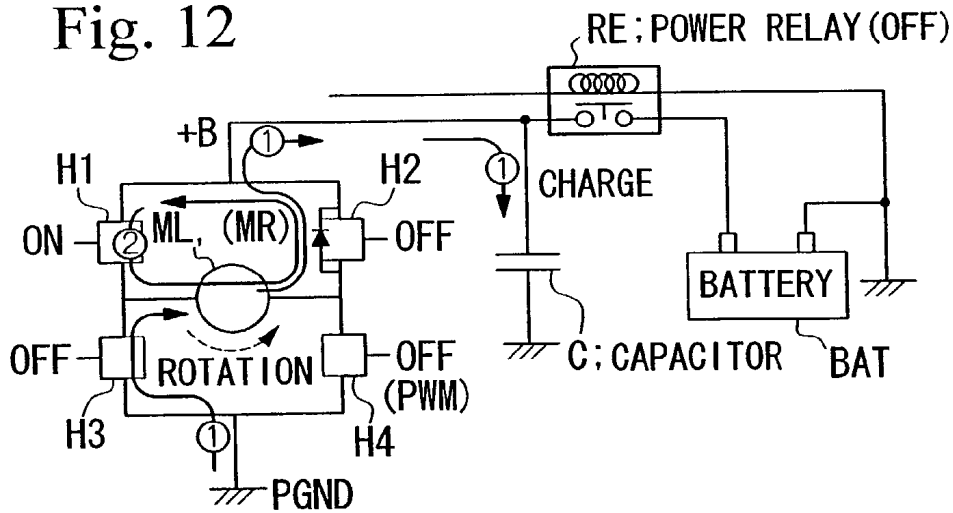


Fig. 13

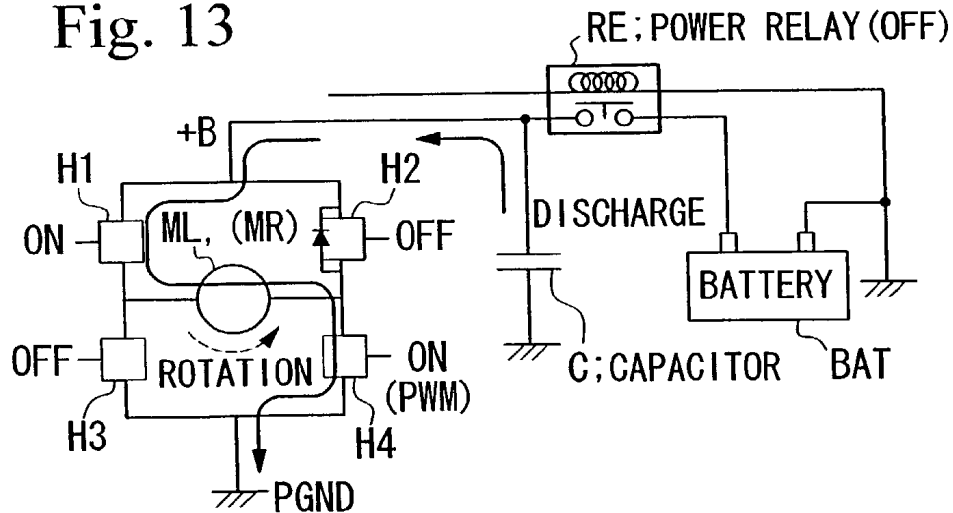


Fig. 14

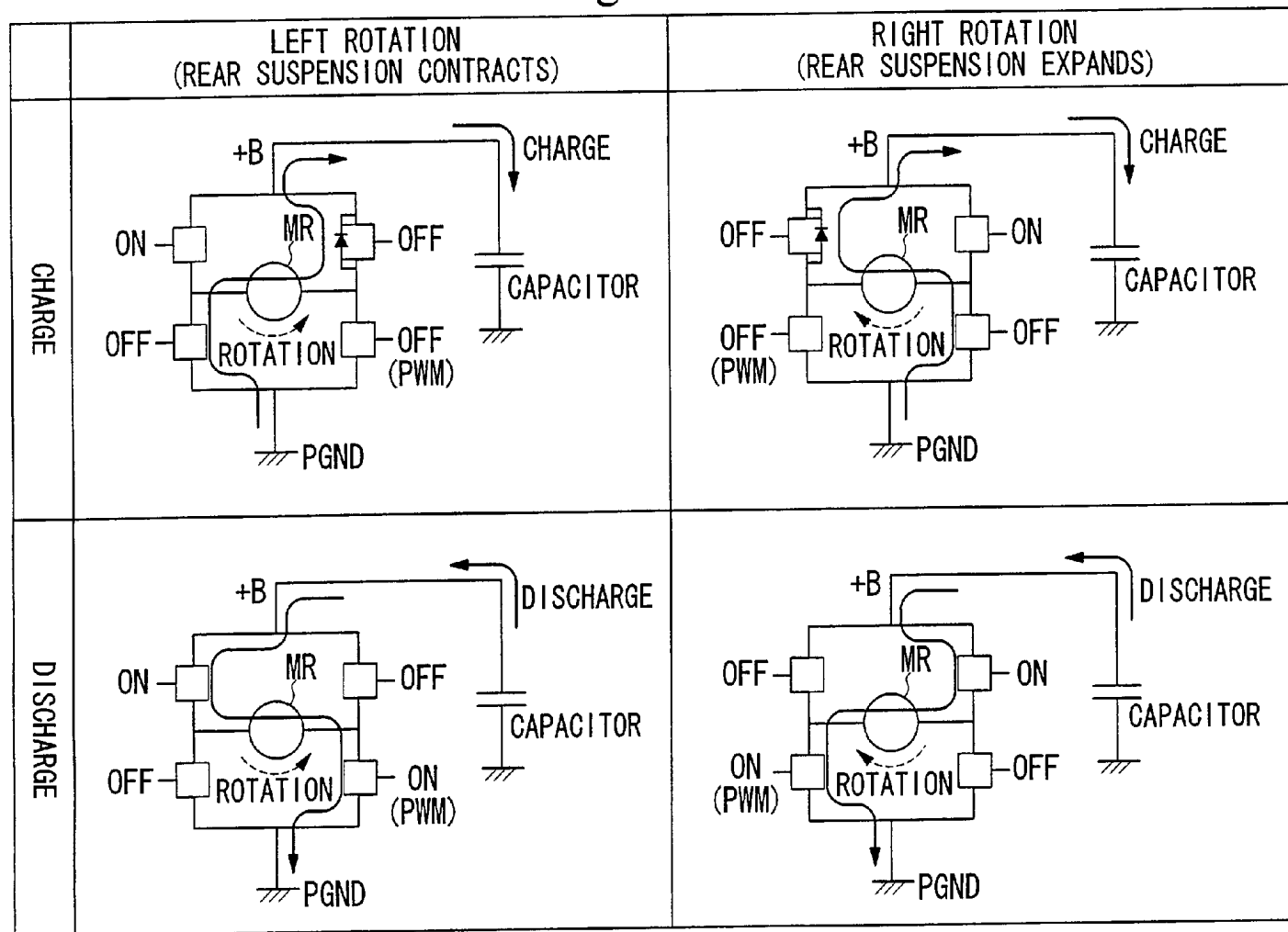


Fig. 15

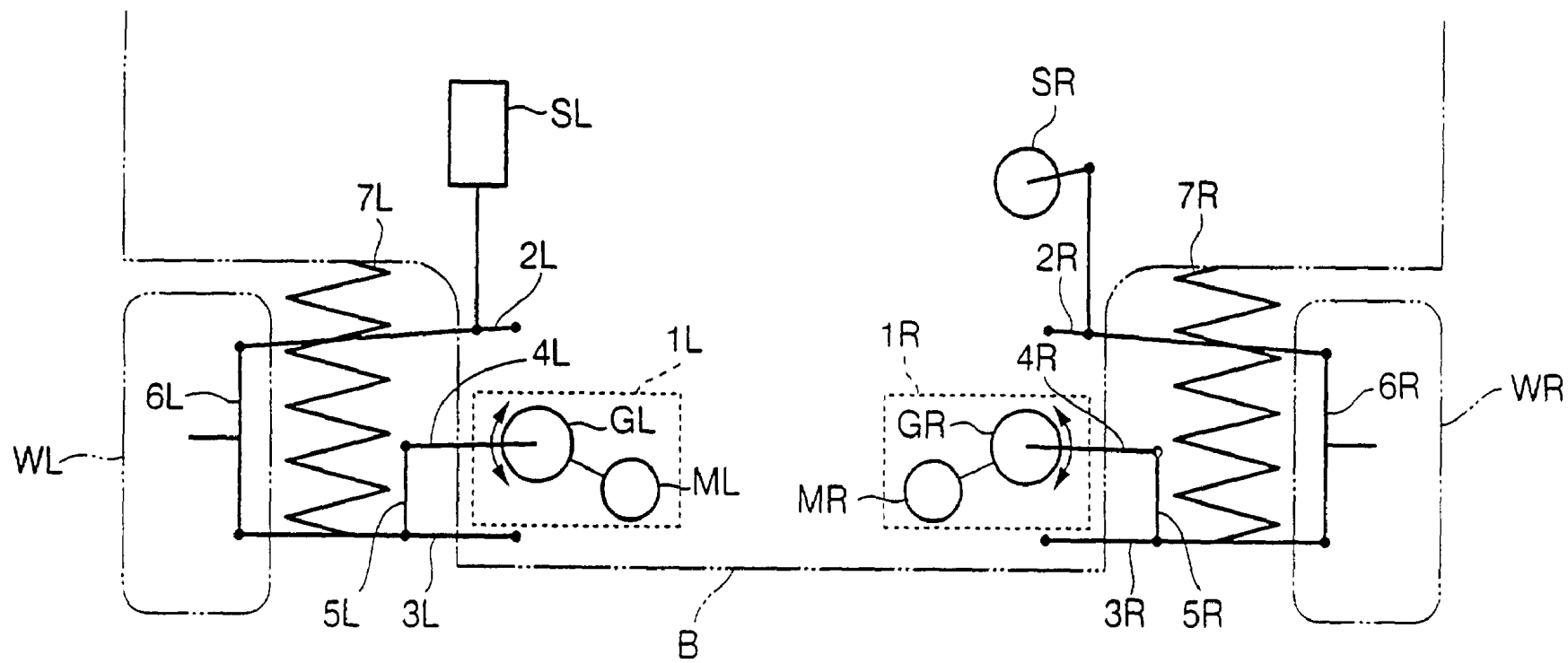
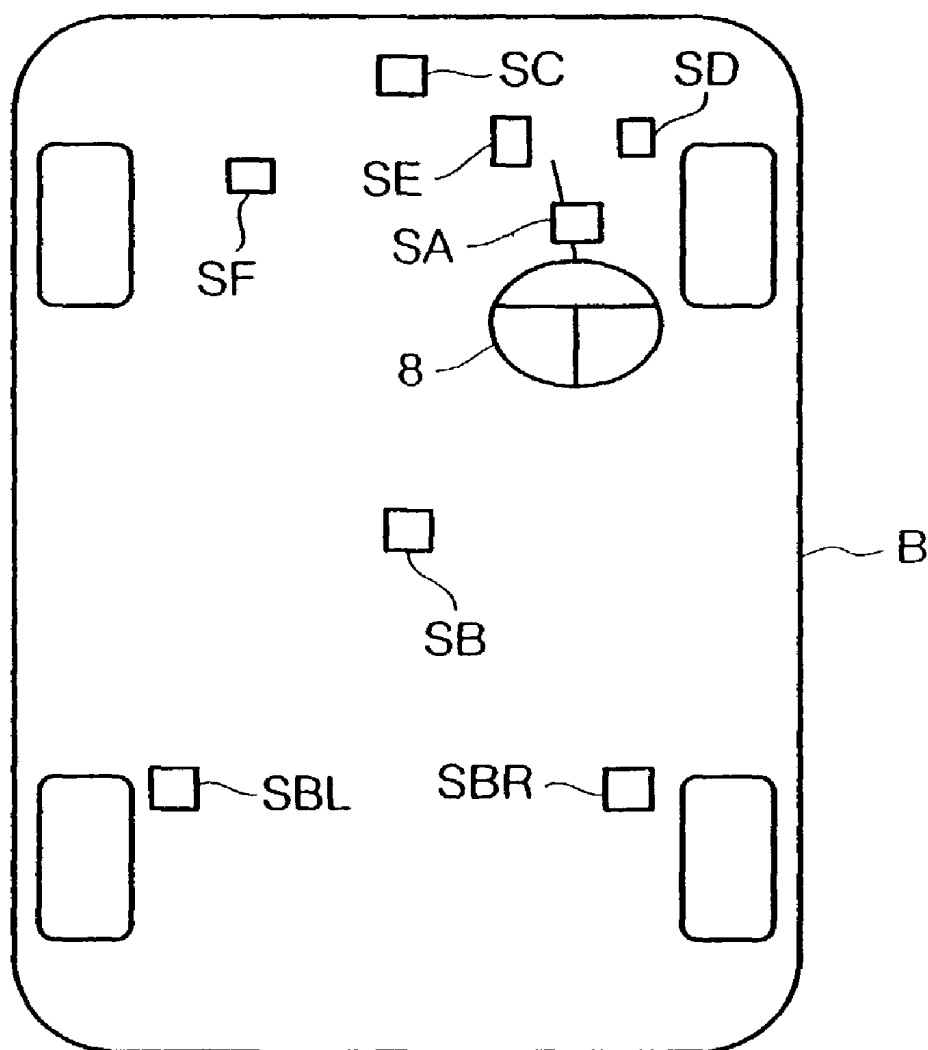


Fig. 16



SUSPENSION CONTROL APPARATUS

BACKGROUND OF THE INVENTION

[0001] 1. Field of the Invention

[0002] The current invention relates to a control system for a suspension provided on a vehicle such as an automobile, and in particular, relates to a control apparatus for a suspension provided with an actuator that applies a force to the vehicle wheels in a vertical direction.

[0003] 2. Description of the Related Art

[0004] The suspension provided on a vehicle has as the function of improving the ride comfort by absorbing shocks from the road surface with which the wheels are in contact, and generally, coil springs and shock absorbers that suppress the vibration of these coil springs are provided. When the spring rate of the coil spring provided on the suspension is made small, the ride comfort has a tendency to become smooth.

[0005] In contrast, when the spring rate is made small in order to make the ride comfort smooth, rolling occurs easily during the turning of the vehicle, pitching occurs easily during the acceleration and deceleration of the vehicle, and there is a tendency for the controllability to decrease.

[0006] As explained above, because it is difficult to improve the controllability and ride comfort of the vehicle together, generally the suspension of a vehicle is frequently set prioritizing either the ride comfort or prioritizing the controllability, depending on the characteristics of the vehicle.

[0007] In addition, stabilizers for increasing the roll resistance of the vehicle are conventionally known. In the case that the left and right wheels move in phase, the stabilizer does not act as a spring, but in the case that the left and right wheels move out of phase as happens during rolling, the stabilizer acts as a spring and thereby the rolling is suppressed. Because the stabilizer acts as a spring only in the case that the left and right wheels are moving out of phase, rolling can be suppressed even if the spring rate of the coil spring is not made large.

[0008] When the spring rate of the coil spring is made small in order to improve the ride comfort, rolling can easily occur during turning, and thus it is necessary to make the reaction force produced by the stabilizer large by making the diameter of the stabilizer large or making the length of the part that generates the torsion short.

[0009] However, when the reaction force generated by the stabilizer is made large, in the case of driving straight on a road surface having irregular bumps and holes, the result is the same as having made the spring rate of the coil spring large, and there is the problem that the ride comfort is adversely influenced. In addition, the stabilizer can only suppress rolling, and it is not possible to attain the effect of suppressing pitching and bouncing.

[0010] In consideration of the problems described above, it is an object of the current invention to provide a suspension control apparatus that executes control of an actuator that applies a force to the vehicle wheels in the vertical direction depending the state of travel of the vehicle, and at the same time, carries out electrical generation by using the

energy of the vertical movement of the vehicle wheels, and uses the obtained electrical energy to drive the actuator.

SUMMARY OF THE INVENTION

[0011] A first aspect of the invention is a suspension control apparatus wherein the vehicle wheels can receive a force applied in the vertical direction by motors (for example, the motors ML and MR in the embodiments), wherein this suspension control apparatus carries out electrical power generation by the motor depending on the vertical movement of the vehicle wheels, and a storage cycle that stores electricity in a storage apparatus (for example, the battery BAT in the embodiments) and a discharge cycle in which the motor is driven by the electrical power stored in the storage device are repeated.

[0012] According to this invention, because electrical generation is carried out by the motor depending on the vertical movement of the vehicle wheels and a charge cycle that stores electricity in a storage apparatus and a discharge cycle that drives a motor by electrical power stored in the storage device are repeated, and thereby the effect can be attained that the consumption of power necessary for driving the motor that controls the suspension can be reduced. As a result, it becomes possible to provide a motor for carrying out control of the suspension without having to increase the storage capacity of the battery originally provided for driving the electrical equipment of the vehicle.

[0013] A second aspect of the invention is a suspension control apparatus comprises two storage members whose storage apparatuses have differing storage capacities, where a connection control device (for example, the power relay RE in the embodiments) controls the connection between the main storage member (for example, the battery BAT in the embodiment) having a larger storage capacity and the motor, and during the storage cycle the connection between the main storage member and the motor is cut by the connection control device, and the auxiliary storage member having the small storage capacity (for example, the capacitor C in the embodiments) is charged.

[0014] According to this invention, a suspension control apparatus comprises two storage members whose storage apparatuses have differing storage capacities, where a connection control device controls the connection between the main storage member having a large storage capacity and the motor, and during the storage cycle the connection between the main storage member and the motor is cut by the connection control device, and the auxiliary storage member having the small storage capacity is charged, and thereby the effects are attained that the charge and discharge can be realized without providing a complicated control circuit for carrying out the charge and discharge of the auxiliary storage apparatus having the small storage capacity, and the structure can be simplified.

[0015] A third aspect of the invention is a suspension control apparatus that drives the motor by the auxiliary storage member when carrying out control of the ride comfort of the vehicle (for example, sky hook control in the embodiments) and drives the motor by the main storage member when carrying out control other than the ride comfort of the vehicle (for example, roll suppressing control and pitching suppressing control in the embodiments).

[0016] According to this invention, during control for improving the ride comfort of the vehicle during forward travel in a normal area, the motor is driven by the electrical power stored in the auxiliary storage apparatus, and thereby the effects can be attained that the reducing of necessary power consumption can be effectively implemented, and further energy reductions can be implemented.

BRIEF DESCRIPTION OF THE DRAWINGS

[0017] FIG. 1 is a flowchart showing the main processing for the control of the suspension shown in FIG. 1.

[0018] FIG. 2 is a flowchart showing the detailed processing in step S 1 shown in FIG. 1.

[0019] FIG. 3 is a flowchart showing the detailed processing in step S 1 shown in FIG. 1.

[0020] FIG. 4 is a flowchart showing the detailed processing in step S 1 shown in FIG. 1.

[0021] FIG. 5 is a flowchart showing the detailed processing in step S 1 shown in FIG. 1.

[0022] FIG. 6 is a flowchart showing the detailed processing in step S 2 shown in FIG. 1.

[0023] FIG. 7 is a flowchart showing the detailed processing in step S 3 shown in FIG. 1.

[0024] FIG. 8 is a flowchart showing the detailed processing in step S 4 shown in FIG. 1.

[0025] FIG. 9 is a flowchart showing the detailed processing in step S 3 shown in FIG. 1.

[0026] FIG. 10 is a flowchart showing the detailed processing in step S 4 shown in FIG. 1.

[0027] FIG. 11 is an explanatory drawing showing the motor drive operation during roll suppressing and pitching suppressing control.

[0028] FIG. 12 is an explanatory drawing showing the operation of charging during sky hook control.

[0029] FIG. 13 is an explanatory drawing showing the operation of discharging during sky hook control.

[0030] FIG. 14 is a schematic diagram showing the division of the charging and discharging events for each of the rotation directions of the motors.

[0031] FIG. 15 is an explanatory drawing showing the disposition of the sensors that detect the state of the vehicle body B.

[0032] FIG. 16 is an explanatory drawing showing the disposition of the sensors that detect the state of the vehicle body B.

DETAILED DESCRIPTION OF THE INVENTION

[0033] In the invention of the current application, by supplementing the spring rate of the spring in the suspension by the torque generated by the actuator, controllability and ride comfort become compatible, and at the same time, power generation is carried out using the energy of the vertical motion of the wheels and the actuator that generates torque is driven by the obtained electrical power.

[0034] Thereby, depending on the conditions of travel of the vehicle, because the firmness of the ride of the vehicle can be substantially increased and the leaning of the vehicle due to rolling and pitching can be mitigated, the stability of the travel of the vehicle can be attained, and at the same time, the ride comfort of the passengers can be improved.

[0035] When setting the spring rate of the spring to a large value from the beginning in order to increase the rigidity of the vehicle, in a state of driving forward in the road surface, the ride comfort deteriorates because the vehicle directly receives the shock due to irregularities in the road surface.

[0036] However, according to the invention of the current application, the torque generated by the actuator compensates the difference between the spring rate for obtaining rigidity of the vehicle necessary during cornering and deceleration by setting the spring rate of the spring to a value that can reduce the shock due to the condition of the road surface during forward travel, and thus the controllability and ride comfort can be improved. Furthermore, the charge cycle in which the storage apparatus is charged and the discharge cycle in which the motor is driven by the electrical power stored in the storage apparatus is repeated, and thus consumption of electrical power necessary for driving the motor that controls the suspension can be saved.

[0037] Below, the embodiments of the current invention will be explained with reference to the drawings.

[0038] FIG. 1 is a perspective drawing from the back of the vehicle showing the structure of the suspension in the rear (rear wheels) side of the vehicle according to an embodiment of the current invention.

[0039] In the suspension of the rear wheel in this figure, the knuckle 6L that supports freely rotatably the wheel WL is supported so as to be able to move vertically by the A-shaped upper arm 2L and the lower arm 3L. The upper arm 2L is connected to the upper part of the knuckle 6L by the joint provided at the distal end, and connected to the vehicle body B by the joint provided on the proximal end. The lower arm 3L is connected to the lower part of the knuckle 6L by the joint provided at the distal end, and connected to the vehicle body B by the joint provided at the proximal end. The lower part of the spring 7L is supported by the center part of the lower arm 3L, and the upper part of the spring 7L is supported by the vehicle body B. The actuator 1L is connected via the link 5L and the drive arm 4L to the proximal end of the lower arm 3L. In addition, a shock absorber (not illustrated) is provided between the vehicle body B and the lower arm 3L. Here, the actuator 1L is formed by the decelerator GL and the motor ML. The torque generated by the motor ML increases depending on the deceleration ratio of the decelerator GL, and is applied to the lower arm 3L.

[0040] Furthermore, the suspension of the right rear wheel has an identical structure, except that the 'L' in the accompanying reference numerals for the essential components described above has been changed to 'R'.

[0041] In addition, the front parts of the knuckle 6L and the knuckle 6R are connected together by a stabilizer (not illustrated), and each of the back parts of the vehicle body B, the knuckle 6L, and the knuckle 6R are connected by a lateral link (not illustrated).

[0042] Due to this structure, in the suspension of the back left wheel, the vehicle body B moves vertically with respect to the road surface due to cornering, and the lower arm 3L and the upper arm 2L connected to the knuckle 6L move vertically with the proximal end connected to the vehicle body B serving as the starting point. Thereby, the spring 7L and the shock absorber connected to the lower arm 3L expand and contract depending on the vertical motion, and the vertical motion of the vehicle body B with respect to the road surface is absorbed. At this time, the actuator 1L is driven, and when the drive arm 4L is rotated in a direction identical to that of the axis of rotation, the torque (N-m) of the vertical motion is transmitted to the lower arm 3L connected via the link 5L to the drive arm 4L, and the spring rate of the spring 7L is compensated.

[0043] In addition, at the same time, in the suspension of the rear right wheel as well, simply by changing the accompanying reference numerals for the essential components from 'L' to 'R', a movement identical to that of the movement of the suspension of the left rear wheel is carried out. Thereby, by linking together the actuators 1L and 1R provided in the wheel WL and the wheel WR and carrying out control, the spring rate of the springs 7L and 7R can be compensated, and the leaning of the vehicle body due to rolling and pitching can be actively controlled.

[0044] That is, the suspension of the rear left wheel shown in FIG. 1 depends on the vertical movement of the vehicle body B due to cornering and the like, and in the case that the spring 7L connected to the lower arm 3L expands and contracts, depending on the amount of this expansion or contraction, a torque is applied to the vehicle body B, and this works to correct the leaning of the vehicle body B.

[0045] Below, for ease of explanation, each of the torques applied to the lower arms 3L and 3R when the actuators 1L and 1R rotate the drive arms are denoted by TTL and TTR.

[0046] Here, the suspension that detects the state of the vehicle body B will be explained with reference to FIG. 15 and FIG. 16. FIG. 15 is an explanatory drawing showing the disposition of the stroke sensor installed on the vehicle body B in order to detect the amount of the suspension stroke shown in FIG. 1. In this figure, reference symbols SL and SR are stroke sensors that detect and output the relative amount of the movement (stroke amount) of the vehicle body B and the lower arm 3L. The example shown in FIG. 15 is one wherein a stroke potentiometer is installed in the suspension of the rear left wheel to serve as the stroke sensor SL. In the example shown here, the stroke sensor SL is installed in the vehicle body B, and joined to the upper arm 2L by a link. In addition, the suspension of the rear right wheel is an example using a rotating potentiometer as the stroke sensor SR, and the amount of movement of the upper arm 2R is detected by the rotation angle. The stroke sensors SL and SR are not limited to stroke or rotation type potentiometers, and any sensors that can detect the amount of the stroke of the suspension can be used.

[0047] What here is being called the stroke amount is the value corresponding to the amount of the displacement of the springs 7L and 7R, and thus any sensor that can detect the amount of each of the displacements of the springs 7L and 7R can be used. For example, the amount of the left and right stroke corresponding to the acceleration applied in the transverse direction of the vehicle B can be found in

advance, and then depending on the acceleration detected by the acceleration sensors, the stroke amount can be found in by referring to stroke amounts found in advance.

[0048] FIG. 16 is a schematic diagram viewing the vehicle from above. In this figure, reference symbol SA is a steering angle sensor that detects and outputs the steering angle and the steering direction of the steering wheel 8. The steering angle sensor SA outputs a signed numerical value, and for example when steering to the left, a positive numerical value that depends on the angle of the steering is output, and when steering to the right, a negative numerical value that depends on the angle of the steering is output. Reference symbol SB is an acceleration sensor that is installed at the position of the center of gravity of the vehicle body and detects the acceleration of the vehicle body B in the vertical direction (the direction perpendicular to the road surface). Reference symbol SC is an acceleration sensor that is installed at a forward position in the vehicle body B, and detects and outputs the acceleration in the forward direction of the vehicle body B. Reference symbol SD is an accelerator opening sensor that detects and outputs the opening of the accelerator. Reference symbol SE is a leg-power sensor that detects and output the leg power on the brake pedal. Reference symbol SF is a vehicle speed sensor that detects and outputs the speed of the vehicle.

[0049] Moreover, an acceleration sensor that detects the acceleration of the vehicle body B in the vertical direction can be installed on each of the left and right rear damper mount of the vehicle body B, as shown by reference symbols SBL and SBR. When the acceleration sensors that detect the acceleration in the vertical direction in this manner are installed at positions on both the left and right suspension, the state of the vehicle body B can be detected in more detail.

[0050] In addition, the control of the actuators 1L and 1R and the detection of the output of the stroke sensors SL and SR, the steering angle sensor SA, the acceleration sensor SB, and the acceleration sensor SC are carried out by a control unit (not illustrated). This control unit is formed by a CPU and a storage unit such as memory, and the CPU carries out control of the actuators 1L and 1R following a program stored in the storage unit.

[0051] Next, the operation in which the control unit carries out control of the suspension shown in FIG. 1 will be explained with reference to FIG. 2 to FIG. 8. FIG. 2 is a flowchart showing the main processing of the suspension control operation. First, the control unit detects the state of the vehicle B at the current point in time, and then calculates the target values necessary when carrying out control for suppressing the rolling of the vehicle body B (step S 1). The target values used in the following explanation are the target torque (TTL and TTR) generated by the actuators 1L and 1R and the rotation direction that generates this torque.

[0052] Moreover, the processing in step S 1 is explained later in detail with reference to FIG. 3 to FIG. 5.

[0053] Next, the control unit detects the state of the vehicle body B, and calculates the target values necessary for carrying out the sky hook control for improving the ride comfort in the case that the vehicle is in a state of forward motion (step S 2). Sky hook control denotes control in which the vibration of the spring (vehicle body B) is dampened by an actuator based on the Sky Hook Theorem.

[0054] Moreover, the processing in step S 2 is explained later in detail with reference to FIG. 6.

[0055] Next, the control unit detects the state of the vehicle body B at the current point in time, and calculates target values necessary for carrying out control for suppressing pitching of the vehicle body B (step S 3).

[0056] Moreover, the processing of step S 3 is explained later in detail with reference to FIG. 7.

[0057] Next, the control unit determines the control mode depending on the state of the vehicle body B at the current point in time (step S 4). The determination of the control mode denotes determining the control of the suspension at the current point in time by determining which among three types of control (roll suppressing control, sky hook control, pitching suppressing control) described above is appropriate for carrying out control depending on the state of the vehicle B at the current point in time.

[0058] Here, the target values for carrying out the three controls are always calculated because thereby the change in the state of the vehicle can be responded to rapidly.

[0059] Next, the control unit selects the target values calculated in steps S 1, S 2, and S 3 based on the control mode determining results, finds the current amount depending on these target values, and outputs them to the motors ML and MR (step S 5).

[0060] For example, in the case that the control unit controls the torque of the motors ML and MR by PWM (pulse width modulation) control, in a constant cycle, the duty ratio of the width of the H level and the L level is found by the calculations depending on the target values, and the amount of current supplied to the motors ML and MR is adjusted.

[0061] At this time, in the case that the current is to be made small in order to reduce the torque, the control unit calculates the duty ratio so that the width of the H level is widened and the width of the L level is narrowed, and in the case that the current is to be made small in order to reduce the torque, the width of the L level is widened and the width of the H level is narrowed. In addition, the direction in which the torque is generated is controlled by reversing the direction of the current flowing to the motors ML and MR.

[0062] Below, for the sake of explanation, the current direction that generates a torque TL in the direction that expands the spring 7L by the actuator 1L is defined as "+", and the current direction that generates a torque TL in the direction that contracts the spring 7L by the actuator 1L is defined as "-".

[0063] Similarly, the current direction that generates a torque TR in the direction that expands the spring 7R by the actuator 1R is defined as "+", and the current direction that generates a torque TR in the direction that contracts the spring 7R by the actuator 1R is defined as "-".

[0064] Moreover, FIG. 2 shows the execution of the processing of steps S 1 to S 3 in sequence, but the calculation of three target values can be calculated simultaneously in parallel. Thereby, the period during which the control operation shown in FIG. 2 is executed can be shortened.

[0065] In this manner, depending on the state of the vehicle body B, either the roll suppressing control, sky hook

control, or pitching control is selected, and control of the suspension is carried out. Thereby, the controllability and the ride comfort of the passengers can be improved together.

[0066] Next, the operation of calculating the target values in the case of carrying out the roll suppressing control shown in step S 1 in FIG. 2 will be explained with reference to FIGS. 3 to 5. FIGS. 3 to 5 are flowcharts showing the processing in which the control unit calculates the target values for carrying out control of the actuators 1L and 1R.

[0067] It is assumed in this operation that the driver is riding in the vehicle, and that by turning on the ignition switch, the control unit stores in the storage unit the measured values MDL and MDR (unit: mm) output by the stroke sensors SL and SR at this point in time as the respective reference values DL and DR (unit: mm).

[0068] In addition, the control unit starts the processing of each of the flow charts shown in FIGS. 3 to 5 following a program.

[0069] The processing in these flow charts is repeated as one set of processes at each predetermined time interval (for example, every 10 msec), and based on the obtained results, the control of the generation of the torques TTL and TTR generated by each of the actuators 1L and 1R is carried out for each time interval.

[0070] First, with reference to FIG. 3, the calculation processing of the target values based on the speed of the steering will be explained. The torques TL and TR generated by the actuators 1L and 1R based on the speed of the steering are found by this processing.

[0071] First, the control unit reads the amount of the steering angle amount output by the steering angle sensor (step S 11). Next, the control unit determines the direction of the steering based on the output value of the steering angle sensor SA (step S 12). What is here termed the direction of the steering indicates whether the steering wheel 8 is steered in to the left or right direction from the center position, and takes any of the values "right", "center", or "left".

[0072] Next, the control unit calculates the change in the amount of the steering angle at the time interval described above, that is, the steering angle speed (rad/sec) as the differential value of the amount of steering (step S 13). This is in order to differentiate precisely the steering state using the direction of the steering angle speed. For example, the process when the steering wheel 8 returns to the central position again after turning to the left can only be differentiated by the steering direction, and thus the direction of the steering angle speed is necessary.

[0073] Next, the control unit finds the target values depending on the steering angle speed found in step S 13 by referring to a target value map stored in the storage unit. FIG. 11 is an example of a target value map that defines the relationship between the steering angle speed and the torques generated by the actuators 1L and 1R. The control unit selects each of the torques TR and TL corresponding to the steering angle speed by referring to the tangent value maps, and outputs this as the calculation result. This output is sent to the processing described below (the processing shown in FIG. 5).

[0074] Next, the calculation processing of the target values (YL and YR) based on the amount of the stroke will be

explained with reference to **FIG. 4**. First, the control unit reads each of the measured values MDL and MDR from the stroke sensors SL and SR. In addition, the control unit calculates the difference between the measured value and the reference value, that is, the amount of the stroke (step S 22).

[0075] Specifically, the control unit finds the stroke amount ΔDL of the left rear wheel by calculating the formula " $MDL - DL$ ", similarly finds the stroke amount ΔDR of the right rear wheel by calculating the formula " $MDR - DR$ ", and stores these in the storage unit. By these calculations, the change in the amount of the stroke can be found subsequent to the state when the ignition switch is turned ON.

[0076] Next, the control unit finds the stroke difference ΔLR between the stroke amount ΔDL of the left rear wheel and the stroke amount ΔDR of the right rear wheel found in step S 22 by calculating the equation " $\Delta DL - \Delta DR$ " (step S 23). Next, the control unit determines the state of the vehicle based on the stroke difference ΔLR that has been found (step S 24). The control unit determines that the vehicle body B is "centered" with respect to the road surface when the stroke difference $\Delta LR = 0$, determines that the vehicle body B is "turning right" when the stroke difference $\Delta LR > 0$, and determines that the vehicle is "turning left" when the stroke difference $\Delta LR < 0$. The results of this determination are stored together with the stroke difference ΔLR in the storage unit as the vehicle state value.

[0077] Next, the control unit calculates the target values based on the stroke difference ΔLR (step S 25).

[0078] Specifically, the stroke difference ΔLR (mm) is multiplied by the spring rate difference ΔJ between the target spring rate JT (unit: N/mm, where N is a Newton) set corresponding to the vehicle type and the basic spring rate JS (unit: N/mm) of the spring 7L actually installed in the suspension, and the resulting value serves as the roll resistance amount of the force FW by which the reaction force is insufficient.

[0079] Therefore, the control unit carries out a calculation based on the equation " $(JT - JS) \times \Delta LR$ " to find the force FW by which the reaction force is insufficient.

[0080] In addition, the torques TL and TR that are generated by each of the actuators 1L and 1R are the torques necessary for compensating the spring rate difference ΔJ , which is amount by which the base spring rate JS is insufficient with respect to the target spring rate JT, and the lever ratio DD (the approximate length of the drive arms 4L and 4R; unit: cm) is multiplied by the force FW by which the reaction force is insufficient found above.

[0081] Thus, the control unit carries out calculations based on the equation " $(FW \times DD)/2$ " to find each of the torques YL and YR. Here, the reason for dividing " $FW \times DD$ " by 2 is that for the torque necessary to complement the roll resistance (the apparent increase in the spring rate), the complementary actuators 1L and 1R each generate $\frac{1}{2}$ the torque towards the reaction directions.

[0082] Next, the control unit applies a sign to the " $(FW \times DD)/2$ " found above depending on whether the vehicle state value is "turning right" or "turning left", and calculates the torques YL and YR. The torques YL and YR found by this calculation serve as target values based on the amount of the stroke.

[0083] For example, in the case that the driver turns to the right and the vehicle body B rolls to the left, the control unit calculates the torque $TL (+FW \times DD/2)$ in order to apply a reaction force in the direction in which the actuator 1L lengthens the spring 7L, while in contrast calculates the torque $YR (-FW \times DD/2)$ in order to apply a reaction force in the direction in which the actuator 1R shortened the spring 6R. Below, for the sake of explanation, the torque in the direction of a lengthening of a spring is denoted-by "+", and the torque in the direction of a shortening of a spring is denoted by "-".

[0084] Next, the control unit determines whether or not the vehicle state should be treated as "center" based on the vehicle state value and the stroke difference ΔLR (step S 26). This determination determines whether the amount should be treated as substantially 0 even though the stroke difference ΔLR is not 0, and if the stroke difference ΔLR is within a range determined in advance, the stroke is processed as 0. Thereby, the vehicle state will be treated as "center" even if there is a minor stroke difference.

[0085] In the case that the result of this determination is not treated as "center", the control unit stores in the storage unit the values of the torques YL and YR found in step S 25, and the processing ends.

[0086] In contrast, in the case that the vehicle state is to be treated as "center", the control unit sets the vehicle state value to "center", and the torques YL and YR, which are the target values based on the stroke found above, are set to "0" (step S 27), these are stored in the storage unit, and the processing ends.

[0087] Next, the processing in which the target values (torques TTL and TTR) actually generated by the actuators 1L and 2R are round using the target values (TL and TR) based on the steering angle speed and the target values (YL and YR) based on the stroke amount will be explained with reference to **FIG. 5**.

[0088] First, the control unit carries out a determination of whether or not the vehicle state value is "center" (step S 31), and if the result of the determination is "center", the processing proceeds to step S 34. If the vehicle state is not "center", the processing proceeds to step S 32.

[0089] Next, based on the vehicle state value and the speed and direction of the steering speed angle, the control unit determines whether or not the direction of the roll of the vehicle body B (either the RL direction of a left turn or the RR direction of a right turn shown in **FIG. 1**) agrees with the direction of the steering angle speed (step S 32). Normally, due to the centrifugal force generated during turning, the vehicle body B rolls in the direction opposite to that of the steering angle. However, in the case of quick steering, there are cases in which the direction of the steering angle speed and the direction of the roll agree because the roll is generated at a timing that is delayed from the start of the steering.

[0090] The result of this determination, in the case that the roll direction and the speed and direction of the steering angle speed agree, only a target value based on the amount of stroke is output.

[0091] Specifically, the control unit outputs the torques YL and YR found from the stroke difference ΔLR as the torques TTL and TTR of the target values.

[0092] In contrast, in the case that the direction of the roll and the direction of the steering angle speed do not agree, the target value based on the steering angle speed and the target value based on the amount of stroke are added and output (step S 34).

[0093] Specifically, the control unit calculates the torque TTL generated by the actuator 1L based on the equation "YL+TL", calculates the torque TTR generated by the actuator 1R based on the equation "YR+TR", and outputs the torques TTL and TTR found by these calculations as the target value of the torque of.

[0094] Next, the operation in which the target values in the case that sky hook control shown in step S 2 of FIG. 2 is calculated is explained with reference to FIG. 6.

[0095] First, the control unit reads the acceleration GB in the vertical direction, which is the output of the acceleration sensor SB that outputs the acceleration of the vehicle body B in the vertical direction (step S 41). Next, the control unit calculates the cancel value for canceling the DC offset portion included in the acceleration GB in the vertical direction (step S 42). This cancel value DC(k) is calculated by the following equation:

$$DC(k) = (0.75 / (0.75 + T)) \cdot (DC(k-1) + GB(k) - GB(k-1)),$$

[0096] where T denotes the sampling time, GB denotes the acceleration in the vertical direction, k denotes the newest value, k-1 denotes the value used in the previous calculation processing.

[0097] Next, the control unit calculates the speed y(k) of the spring (vehicle body B) (step S 43). This speed y(k) is calculated by the following equation:

$$y(k) = (1 / (1.3 + T)) \cdot (1.3y(k-1) + T \cdot DC(k)),$$

[0098] where T denotes the sampling time and DC denotes the cancel value calculated in step S 42. In addition, k denotes the newest value, and k-1 is the value used in the previous calculation processing.

[0099] Next, the control unit determines the operation direction of the actuators 1L and 1R based on the value of the speed y(k) calculated in step S 43 (step S 44). If the determination of the direction of the operation is that the spring speed y(k) is a positive value, the actuators 1L and 1R operation in the shortening direction, and if the value is negative, the operation is in the lengthening direction.

[0100] Next, the control unit calculates the torques TTL and TTR of the target values based on the value of the spring speed y(k) (step S 45).

[0101] The target values TTL and TTR are calculated by the formula $TTL, TTR = |y(k)| \cdot C \cdot K$, where C denotes the damping coefficient and K denotes the torque constant.

[0102] Next, the operation in which the calculation of the target values for carrying out pitching suppressing control shown in step S 3 in FIG. 2 will be explained with reference to FIG. 7.

[0103] First, the control unit reads the current acceleration GC, which is the output value of the current acceleration sensor SC that detects the acceleration of the vehicle B in the current direction (step S 51). In addition, the control unit determines the direction of the reaction force generated by the actuators 1L and 1R based on the current acceleration

GC that has been read (step S 52). The direction of the reaction force is determined by identifying whether the vehicle is accelerating or decelerating based on the value of the current acceleration GC. In the case that the vehicle is accelerating, the springs 7L and 7R in the rear wheels are lengthened because the load shifts towards the back. In contrast, during deceleration, the springs 7L and 7R in the back wheels are lengthened because the load shifts to the front. Therefore, when acceleration is identified, the direction in which the reaction force is generated is the direction in which the springs 7L and 7R lengthen, while in contrast, when deceleration is identified, the direction in which the reaction force is generated is the direction in which the springs 7R and 7L shorten.

[0104] Next, the control unit calculates the amount of shift of the load due to acceleration (step S 53). The load shift amount ΔW is calculated by the following equation:

$$\Delta W = (1/2) \cdot (hg/L) \cdot W \cdot Xg,$$

[0105] where hg is the height of the center of gravity of the vehicle, L is the wheel base of the vehicle, W is the center of gravity of the vehicle, and Xg is the absolute value of the current acceleration GC. Next, the control unit calculates and outputs the target value TTL for one of the actuators (step S 54). The reaction force of the load shift amount ΔW can be generated simultaneously in the same direction by the two actuators 1L and 1R, and thus the target value torques TTL and TTR can be found by the formula $TTL, TTR = \Delta W / 2$.

[0106] Here, the operation in which the power generation is carried out using the energy of the vertical motion of the wheels WL and WR and the obtained electrical power is used to drive the actuators 1L and 1R will be explained with reference to FIGS. 11 to 14. In FIGS. 11 to 13, the reference symbol BAT is a battery (secondary battery) provided in advance in the vehicle for driving the electrical equipment in the vehicle. Reference symbols H1 to H4 are FET (field-effect transistors) for driving the motors ML and MR, and the structure of the FETs H1 to H4 is called an H-bridge structure. In these four FETs, ON/OFF driving and PWM driving are carried out by control signals from the control unit (not illustrated). Reference symbol C is a capacitor for carrying out discharging or charging in order to drive the motors ML and MR. Reference symbol RE is a power relay, and depending on commands from the control unit, supplies or stops supplying voltage of the battery BAT to the H-bridge.

[0107] FIG. 11 is a drawing showing the operation during the roll suppressing control and the pitching suppressing control. First, in step S 4, in the case that the identified control mode is the roll suppressing control or the pitching suppressing control, the control unit turns ON the power relay RE. Thereby, the voltage of the battery BAT is supplied to the H-bridge. In addition, in the case that the motor ML is being rotated to the left, the control unit turns ON the FET H1, turns OFF the FET H2, turns OFF the FET H3, and turns ON the FET H4 (PWM drive), and current is flows to the motor ML. In addition, in the case that the motor ML is being rotated to the right, the control unit turns OFF the FET H1, turns ON the FET H2, turns ON the FET H3 (PWM drive), and turns OFF the FET H4, and the current flows to the motor ML. Thereby, roll suppressing control and pitching suppressing control are carried out.

[0108] FIGS. 12 and 13 are drawings showing the operation during sky hook control, where FIG. 12 shows the charging operation and FIG. 13 shows the discharging operation. First, in step S 4, in the case that the identified control mode is the sky hook control and the capacitor C is charged, the control unit turns OFF the power relay RE. Thereby, the power source supply from the battery BAT is stopped. At this time, an electromotive force is generated from the motor ML due to the wheels moving (the lengthening or shortening of the suspension) in a certain direction (in an upward direction or downward direction) due to bumps and holes in the road surface. In addition, the control unit turns ON the FET H1, turns OFF the FET H2, turns OFF the FET H3, and turns OFF the FET H4. Thereby, the regenerative braking current generated by the motor ML temporarily charges the capacitor C by passing through the parasitic diode (the flow shown by ① in FIG. 12). In addition, when the charging is complete, there is reflux as shown by ② in FIG. 12.

[0109] In contrast, as shown in FIG. 13, in the case of discharging, the control unit turns the power relay OFF, and at the same time, turns ON the FET H1, turns OFF the FET H2, turns OFF the FET H3, and turns ON the FET H4 (PWM drive). Thereby, the motor ML is driven by the electricity stored in the capacitor C, and the force for dampening the spring is generated. This dampening force becomes large in proportion to the amount of the regenerative braking current, and the control of this amount of regenerative braking current is carried out depending on the duty ratio of the PWM that is driving the FET H4. Like the sky hook control, control is carried out such that the duty ratio changes depending on the deviation of the target current value and the regenerative braking current value, and if the deviation is large, the duty also becomes large.

[0110] Moreover, only in the case that a large road surface input is continuously received such that it cannot be absorbed in the dampening operation, the power relay RE can be turned ON, the voltage from the battery BAT can be applied, and the driving force of the actuator can be increased. In addition, in the case of transferring to roll suppressing control in order to increase the roll resistance during turning, at the same time as the transition, the power relay is turned ON, and thereby power of the battery BAT is supplied to the motor ML. Thereby, the driving stability is increased by increasing the out of phase wheel rate. Furthermore, in the case of transferring to pitching suppressing control, like the roll suppressing control, the power of the battery BAT is supplied. This is because the situation in which this control is carried out is limited to the acceleration or deceleration of the vehicle, and because the amount of vertical movement of the tires is small and the electromotive force of the motors ML and MR is almost zero.

[0111] FIG. 14 is a schematic diagram showing the operation of the motor MR provided on the vehicle's right side, and is a drawing wherein the operation of the FETs in the case that the motor MR turns left and turns right is shown separately depending on whether there is a charging and discharging event. As shown in the figure, the control of the actuator that applies force in the vertical direction to the wheels is carried out by driving the FETs depending on the rotation direction and whether they are a charging or discharging, and at the same time, electrical generation is

carried out by using the energy of the vertical motion of the wheels, and the obtained electrical power can be used for driving the actuators.

[0112] Moreover, the direction of rotation of the motor MR is determined by the sign of the value of the voltage value subtracted from the output value of the acceleration sensor SB.

[0113] Next, the control mode determination shown in step S 4 of FIG. 2 and the operation that carries out the charging/discharging of the regenerative braking that accompanies the central mode will be explained with reference to FIG. 8.

[0114] First, the control unit reads the sensor output values (step S 61). The sensor output values read here are obtained during the processing operation in which the target values described above are obtained, and the output values stored in the storage unit are read. In addition, the control unit determines whether or not the absolute value of the steering angle is larger than 5° (step S 62). In the case that the result of this determination is that it is larger than 5° , it is determined that the vehicle is turning, and the target values for roll suppressing control are selected (step S 63). In addition, the control unit turns ON the power relay RE (step S 64). Thereby, control is carried out in which the voltage of the battery BAT is supplied to the H-bridge, and each FET, as described above, is driven depending on control signals from the control unit.

[0115] Next, the control unit multiplies the gain found based on the vehicle speed at this point in time by the selected target values (step S 65). The gain multiplied by these target values is found by referring to the gain map shown in FIG. 12 that defines the relationship between the vehicle speed and the gain. As shown in FIG. 12, in the gain map the X axis is the vehicle speed and the Y axis is the gain, and the gain is a value from 0 to 1. In the example shown here, the gain is 0 when the vehicle speed of 0 to 10 km/h, and the gain is 1 when the vehicle speed is equal to or greater than 20 km/h. Between but not including 10 to 20 km/h, the gain is found by linear interpolation.

[0116] Therefore, this means that when the vehicle speed is between 0 to 10 km/h, control by the actuators 1L and 1R is not carried out, and when the vehicle speed is equal to or above 20 km/h, control is carried out by the target values set in step S 63.

[0117] In this manner, by multiplying the gain by the target values, the target values can be changed depending on the speed of the vehicle, and thus at a low speed less than 10 km/h, minor control operations are carried out due to the influence of bumps and holes in the road surface, and a deterioration in the ride comfort can be avoided.

[0118] Next, in step S 62, in the case that the absolute value of the steering angle is equal to or less than 5° , the control unit determines whether or not the absolute value of the steering angle speed is larger than 0.5 rad/sec (step S 66). In the case that the result of the determination is that the absolute value of the steering angle speed is larger than 0.5 rad/sec, the steering angle is small while the steering angle speed is high, and thus it is determined that the possibility that rolling will occur is high, and the processing proceeds to step S 63, and the processing described above is carried out.

[0119] In contrast, in the case that the absolute value of the steering angle speed is equal to or less than 0.5 rad/sec, the control unit determines whether or not the absolute value of the current acceleration is larger than 2 m/s^2 (step S 67). In the case that the result of this determination is that the absolute value of the current acceleration is equal to or less than 2 m/s^2 , the control unit selects the target values for the sky hook control (step S 68). In addition, the control unit determines whether the necessary current is equal to or greater than 20 A (step S 69). In the case that the result of this determination 20 A or greater is necessary, the control unit turns OFF the power relay RE (step S 70). Thereby, the supply of the voltage of the battery BAT to the motors ML and MR is stopped, and by a control signal from the control unit, control is carried out for driving each of the FETs as described above, and the charge and discharge of the capacitor C is carried out.

[0120] Next, in the case that the necessary current resulting from the determination in step S 69 is equal to or greater than 20 A, the control unit determines whether or not the state requiring 20 A or greater has continued for two seconds (step S 71). If the result of this determination is that two seconds have not passed, the processing proceeds to step S 70. In contrast, in the case that two seconds have passed, the control unit turns ON the power relay RE (step S 72). Thereby, the current of the battery BAT is supplied to motors ML and MR, and by control signals from the control unit, the control in which each FET is driven as described above is carried out. After carrying out step S 70 or S 72, the processing proceeds to step S 65, and the gain is multiplied by the found target values based on the speed.

[0121] In addition, in the case that the result of the determination in step S 67 is that the absolute value of the current acceleration is larger than 2 m/s^2 , the control unit selects the target values of the pitching suppressing control (step S 73). In addition, the control unit turns ON the power relay RE (step S 64). Thereby, the voltage of the battery BAT is supplied to the motors ML and MR, and by control signals from the control-unit, control in which each of the FETs is driven as described above is carried out. In the case that the target values of the pitching suppressing control are selected, the control unit does not carry out multiplying the gain based on the vehicle speed (step S 64). This is because the control unit detects whether or not the current acceleration is equal to or greater than a predetermined value at which pitching occurs easily, and thus multiplying the gain based on the vehicle speed is not necessary in order to carry out a determination of pitching suppressing control based on the absolute value of the current acceleration.

[0122] Moreover, the explanation described above was given in which the power generated by the MR was charged in the capacitor C shown in FIGS. 11 to 14, but this can be charged to a battery BAT provided in advance in the vehicle. In this case as well, the capacity of the battery provided in the vehicle does not have to be made large because the motor itself generates the energy that the motors ML and MR require.

[0123] In this manner, based on the output value of the sensor provided in the vehicle, the state of the vehicle body B is determined depending on whether or not predetermined conditions have been satisfied, and in the case that as a result of the examination it is determined that the vehicle is in a

turning state, control is carried out in which the rolling is suppressed with respect to the suspension, and thereby an effect can be obtained that is identical to the case in which only the spring rate of the suspension is made large only during turning.

[0124] In addition, in the case that as a result of the determination it is determined that the vehicle is progressing straight forward, control is generally carried out in which the bouncing is suppressed with respect to the suspension, and the ride comfort is improved, and thereby, an effect can be obtained that is identical to the case in which the spring rate of the suspension is made small while progressing straight forward.

[0125] In addition, in the case that as a result of the determination it is determined that the vehicle is progressing straight forward and accelerating or decelerating, generally control is carried out in which the pitching is suppressed, and thus an effect can be obtained that is identical to the case in which the spring rate of the suspension is made large only during acceleration and deceleration.

[0126] Furthermore, during the sky hook control for improving the ride comfort, power generation is carried out by the motors depending on the vertical movement of the wheels, and the storage cycle in which power is stored in the storage apparatus and the discharge cycle in which the motor is driven by the power stored in the storage apparatus are repeated, and thereby the effect can be attained wherein the consumption of the power required to drive the motors that control the suspension can be saved. As a result, it becomes possible to provide motors ML and MR for carrying out control of the suspension can be provided without increasing the storage capacity of the battery BAT that was originally installed for driving the electrical equipment of the vehicle.

[0127] In addition, further providing in the vehicle a capacitor C having a storage capacity smaller than the battery BAT provided in advance and a power relay RE that controls the connection between the battery BAT and the motors ML and MR, in the storage cycle the connection between the battery BAT and the motors ML and MR is cut and electricity is stored in the capacitor C, and thereby charging and discharging can be carried out without providing complicated control circuits, and the structure can be simplified.

[0128] In addition, when controlling the ride comfort of the vehicle during forward travel in a typical area (sky hook control), the motors ML and MR are driven by the power stored in the capacitor C, and thereby the savings of necessary power consumption can be efficiently carried out, and further energy saving can be implemented.

[0129] Moreover, in the explanation given above, the case of providing actuators 1L and 1R on the rear wheels of the vehicle was explained, but actuators can similarly be provided on the front wheel suspensions as well, and the control described above carried out. Furthermore, control of each of the vehicle's four wheels can be carried out by providing actuators on all the suspensions of the front and rear wheels. In addition, only one capacitor C needs to be provided in a vehicle, not provided for each of the left and right motors ML and ML, and used by both the two motors ML and MR. In addition, instead of a capacitor C, a secondary battery having a storage capacity less than the battery BAT can be provided.

[0130] As explained above, according to the first aspect of the invention, power generation is carried out by a motor depending on the vertical motion of the wheels, and repeating a storage cycle that stores electricity in a storage apparatus and a discharge cycle that drives the motors by the electrical power stored in the storage apparatus, and thereby the effect can be attained where consumption of necessary power for driving the motors that control the suspension can be reduced. As a result, the storage capacity of the battery originally provided for driving the electrical equipment of the vehicle does not need to be increased, and a motor for carrying out control of the suspension can be provided.

[0131] In addition, according to a second aspect of the invention, a storage apparatus comprises two storage parts having different storage capacities, and a connection control device is provided that control the connection between the main storage member having a large storage capacity and the motors. During the storage cycle, the connection between the main storage member and the motors is cut by the connection control device and the auxiliary storage member having a small storage capacity is charged. Thereby, the effects can be attained that charging and discharging can be realized without providing complicated control circuits for carrying out charging and discharging of the auxiliary storage apparatus having a small storage capacity, and the structure can be simplified.

[0132] In addition, according to a third aspect of the invention, when controlling the ride comfort of the vehicle during forward travel in normal area, the motors are driven by power stored in the auxiliary storage apparatus, and thereby saving of necessary power consumption can be efficiently carried out, and further energy saving can be implemented.

What is claimed is:

1. A suspension control apparatus having the vehicle wheels that can receive a force applied in the vertical direction by motors, wherein:

said suspension control apparatus carries out electrical power generation by the motor depending on the vertical movement of said vehicle wheels, and a storage cycle that stores electricity in a storage apparatus and a discharge cycle in which said motor is driven by the electrical power stored in said storage device are repeated.

2. A suspension control apparatus according to claim 1 comprises:

said storage apparatus comprising a main storage members having a larger storage capacity and an auxiliary storage member having a smaller storage capacity;

a connection control device that controls the connection between said main storage member and said motor,

wherein, during the storage cycle, the connection between said main storage member and said motor is cut by said connection control device, and said auxiliary storage member is charged.

3. A suspension control apparatus according to claim 1 wherein:

said motor is driven by said auxiliary storage member when carrying out control of the ride comfort of the vehicle and said motor is driven by said main storage member when carrying out control other than the ride comfort of the vehicle.

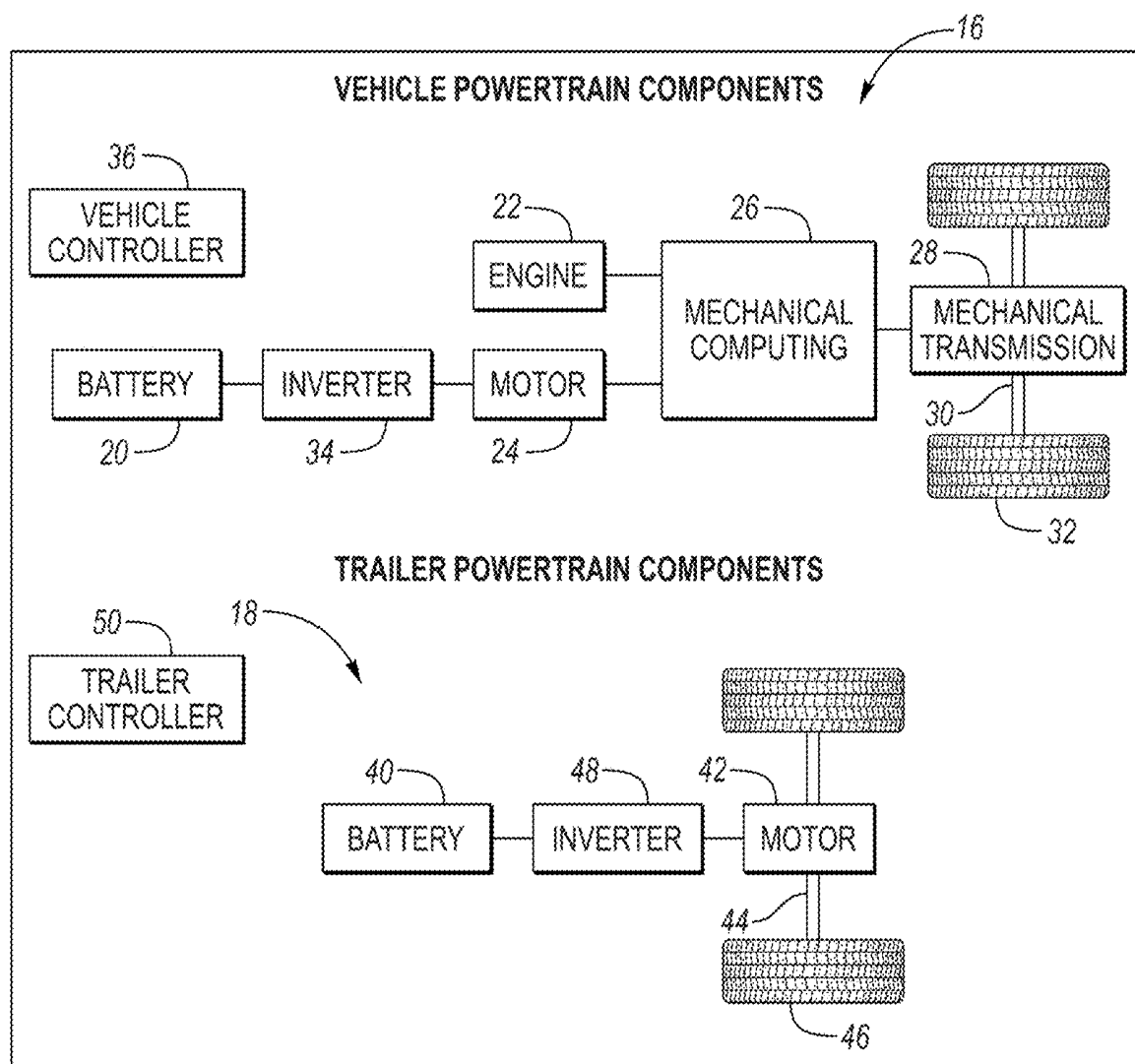
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(19) **United States**(12) **Patent Application Publication**
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INTEGRATED POWERTRAIN OF
COMBINED VEHICLE SYSTEM HAVING
VEHICLE AND ELECTRICALLY POWERED
TRAILER TOWED BY THE VEHICLE**(71) Applicant: **Ford Global Technologies, LLC,**
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2510/1025 (2013.01); **B60W 2510/102**
(2013.01)(57) **ABSTRACT**

A system includes a vehicle having an electrically powered trailer towed thereto. The vehicle has a vehicle powertrain, and the trailer has a trailer powertrain. A controller, in order to propel the vehicle with the trailer towed thereto, operates the vehicle powertrain and the trailer powertrain according to a combined efficiency map based on (i) an efficiency map associated with the vehicle powertrain and (ii) an efficiency map associated with the trailer powertrain.



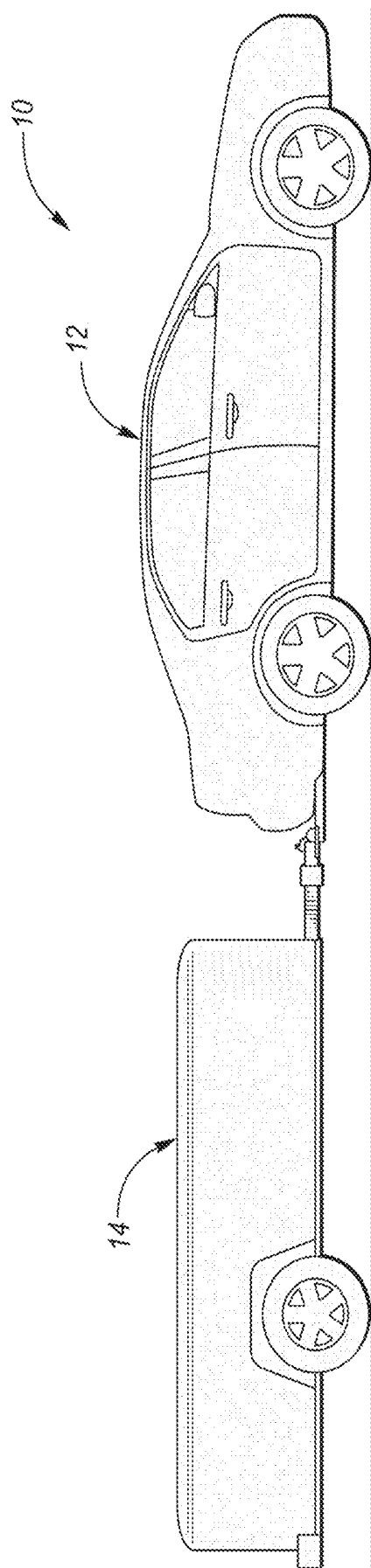


FIG. 1

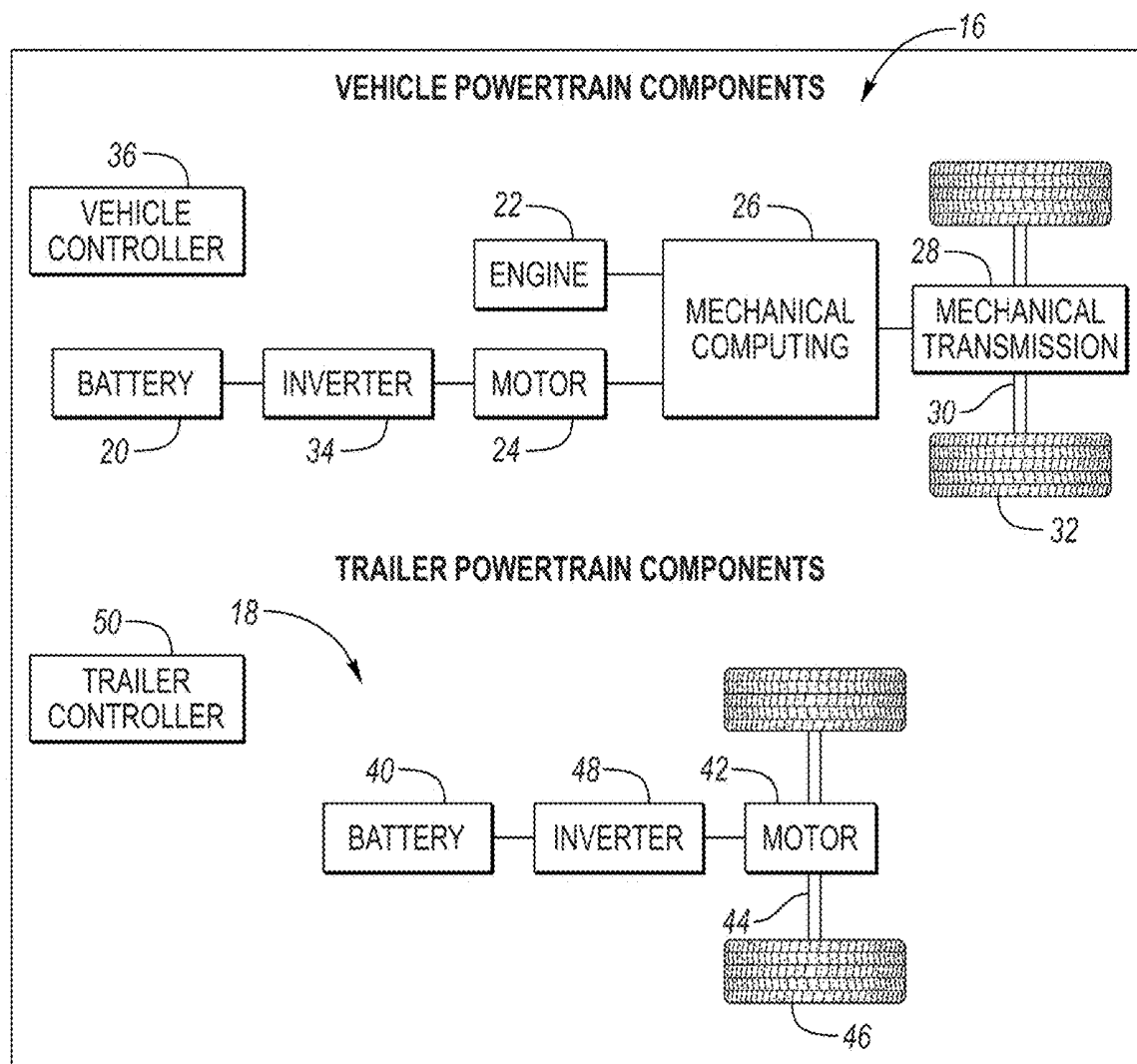


FIG. 2A

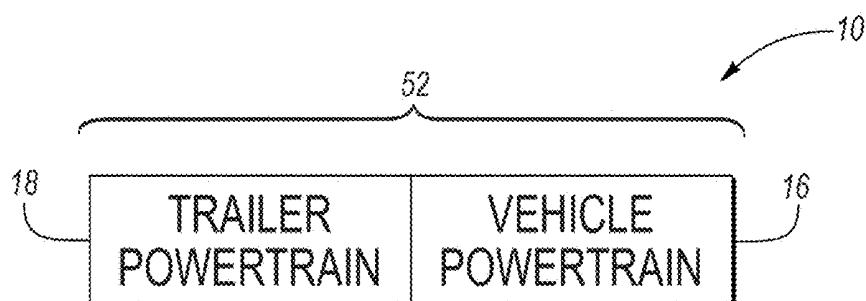


FIG. 2B

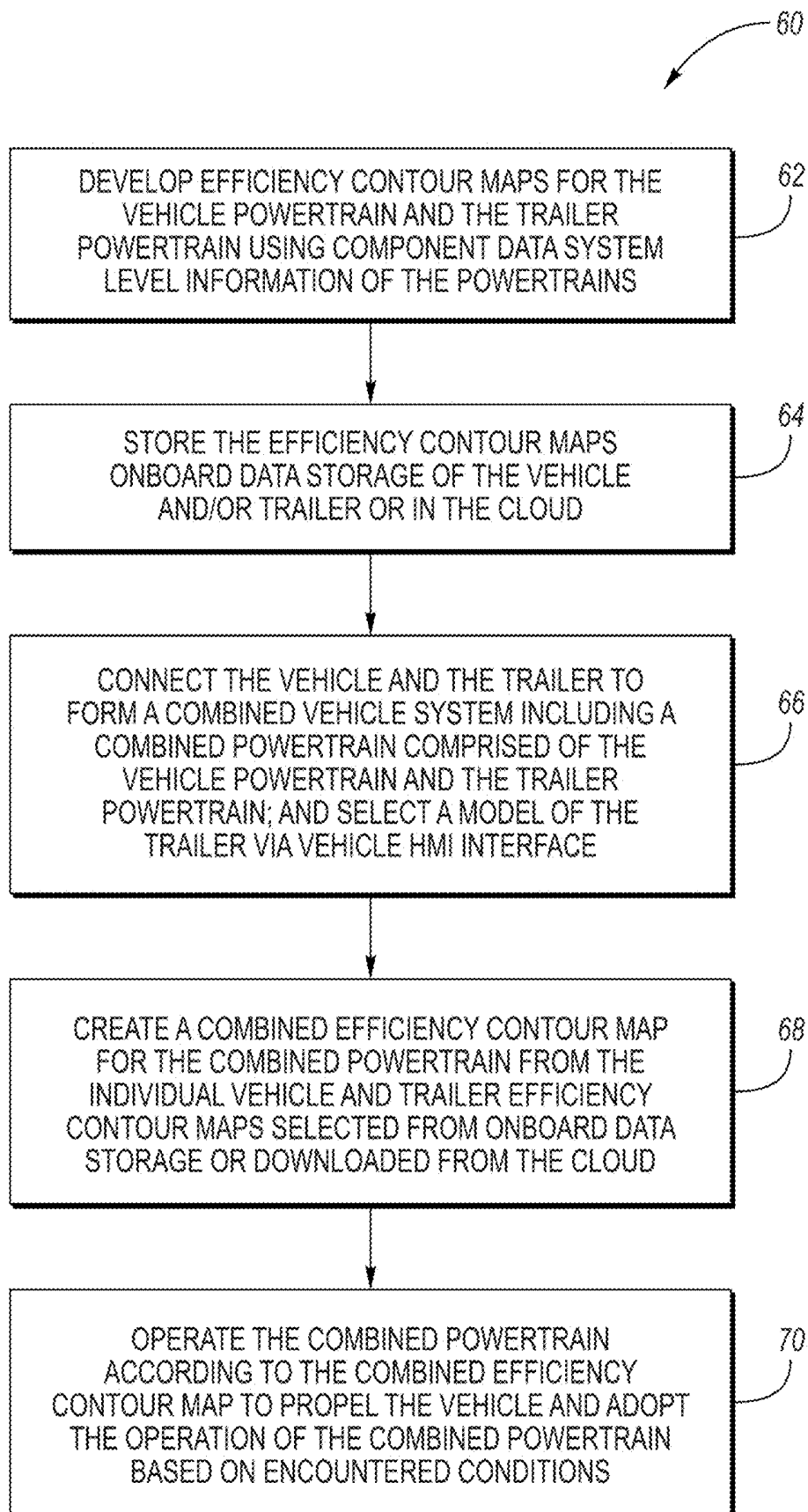


FIG. 3

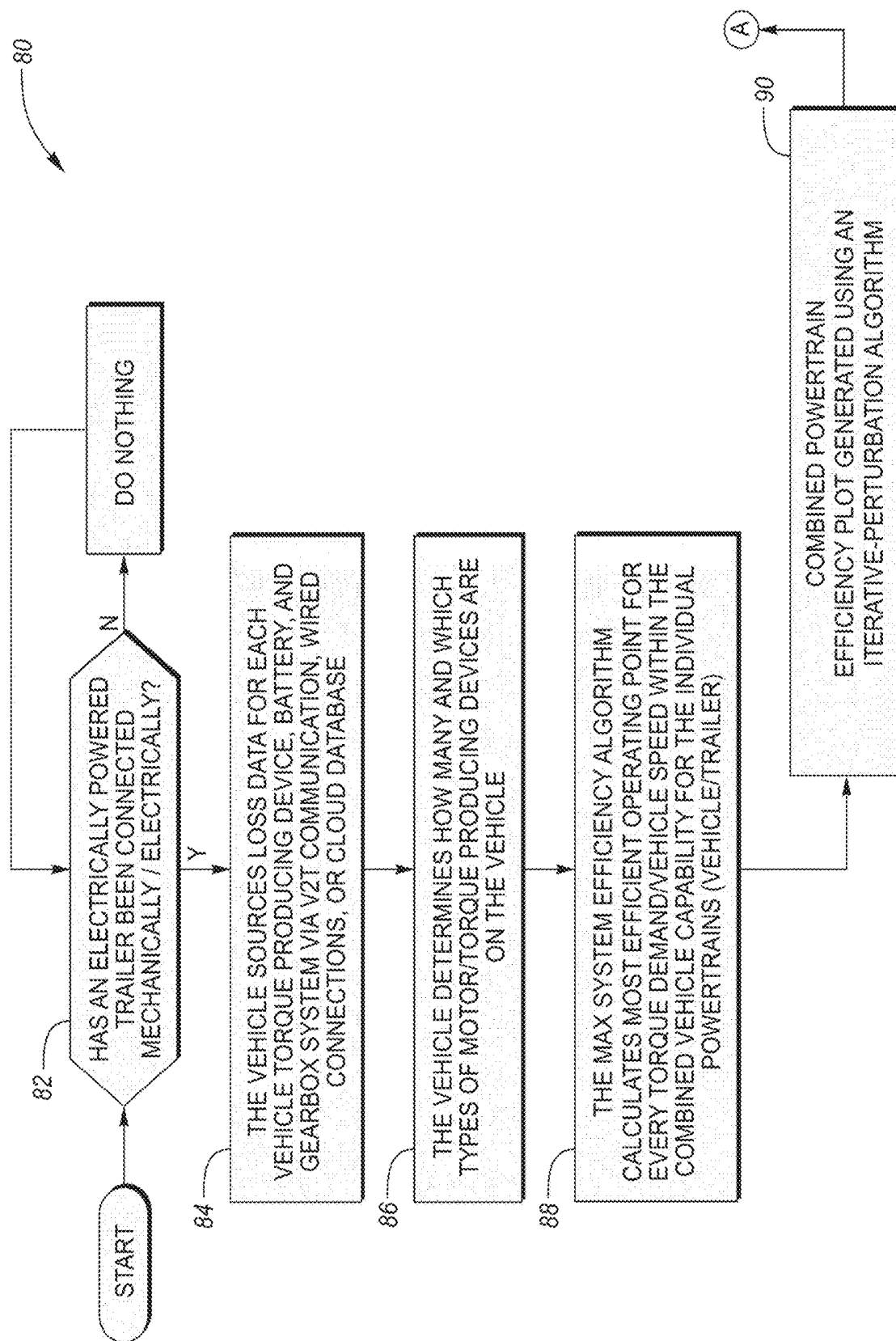


FIG. 4A

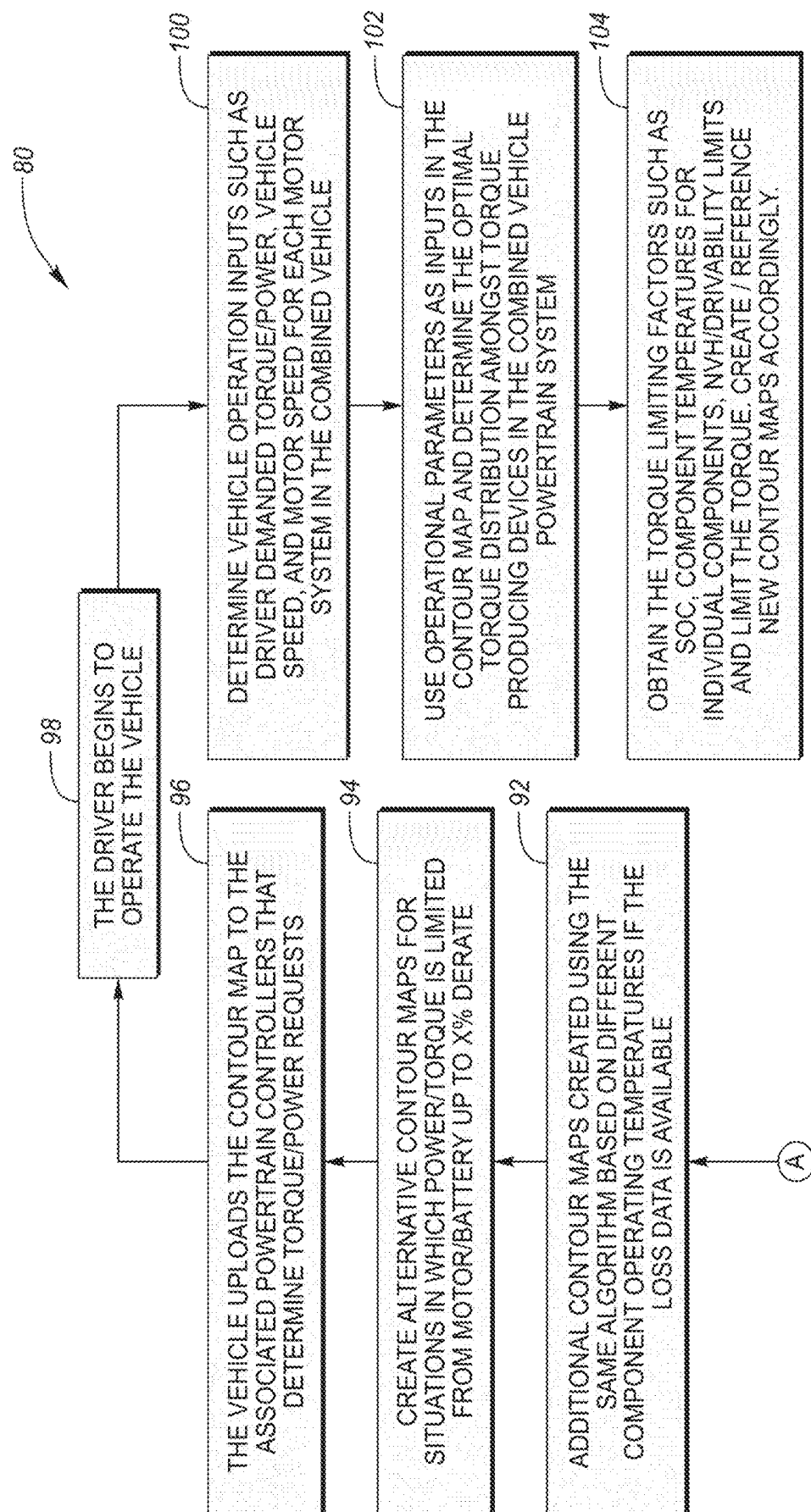


FIG. 4B

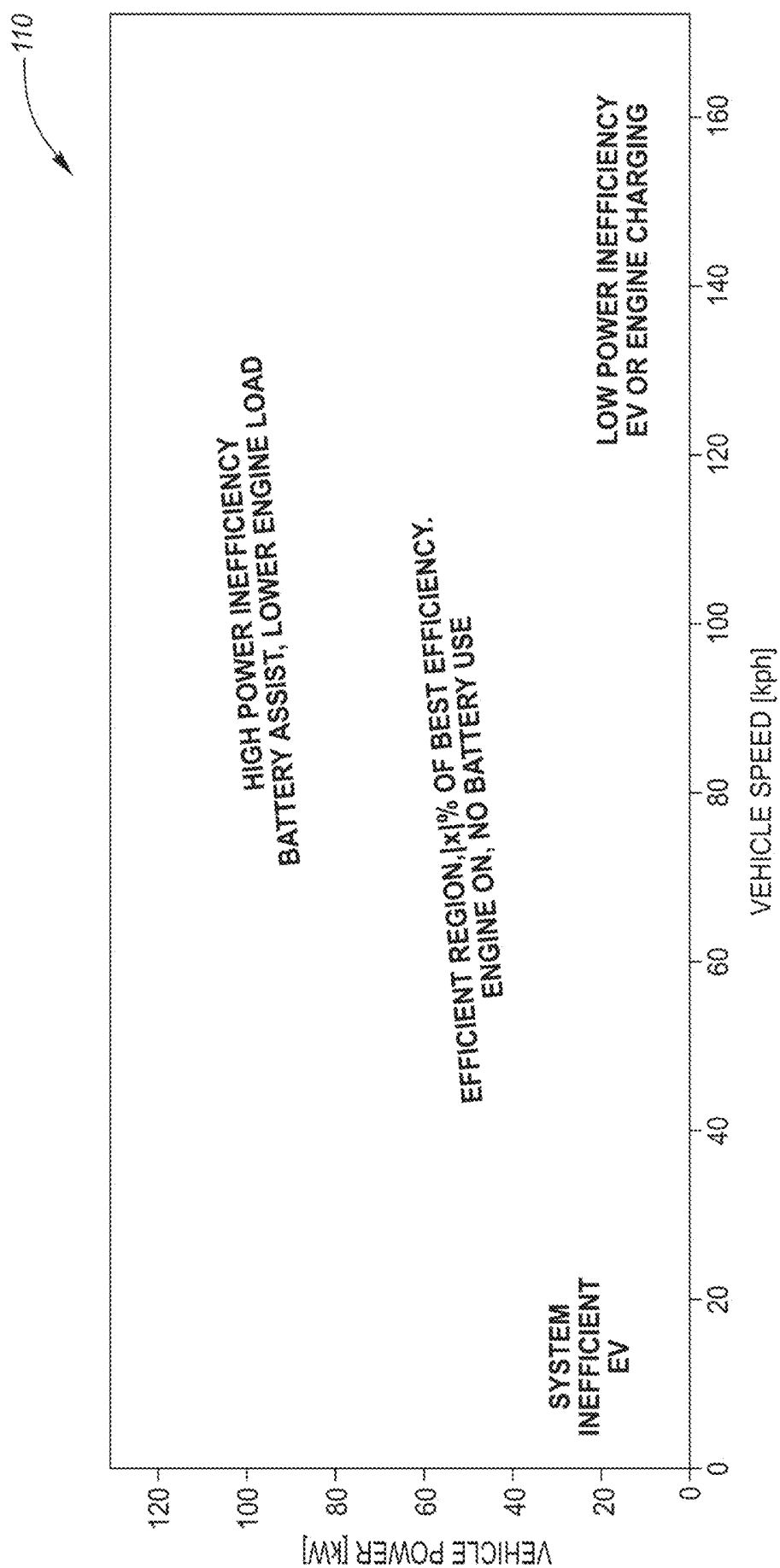


FIG. 5A

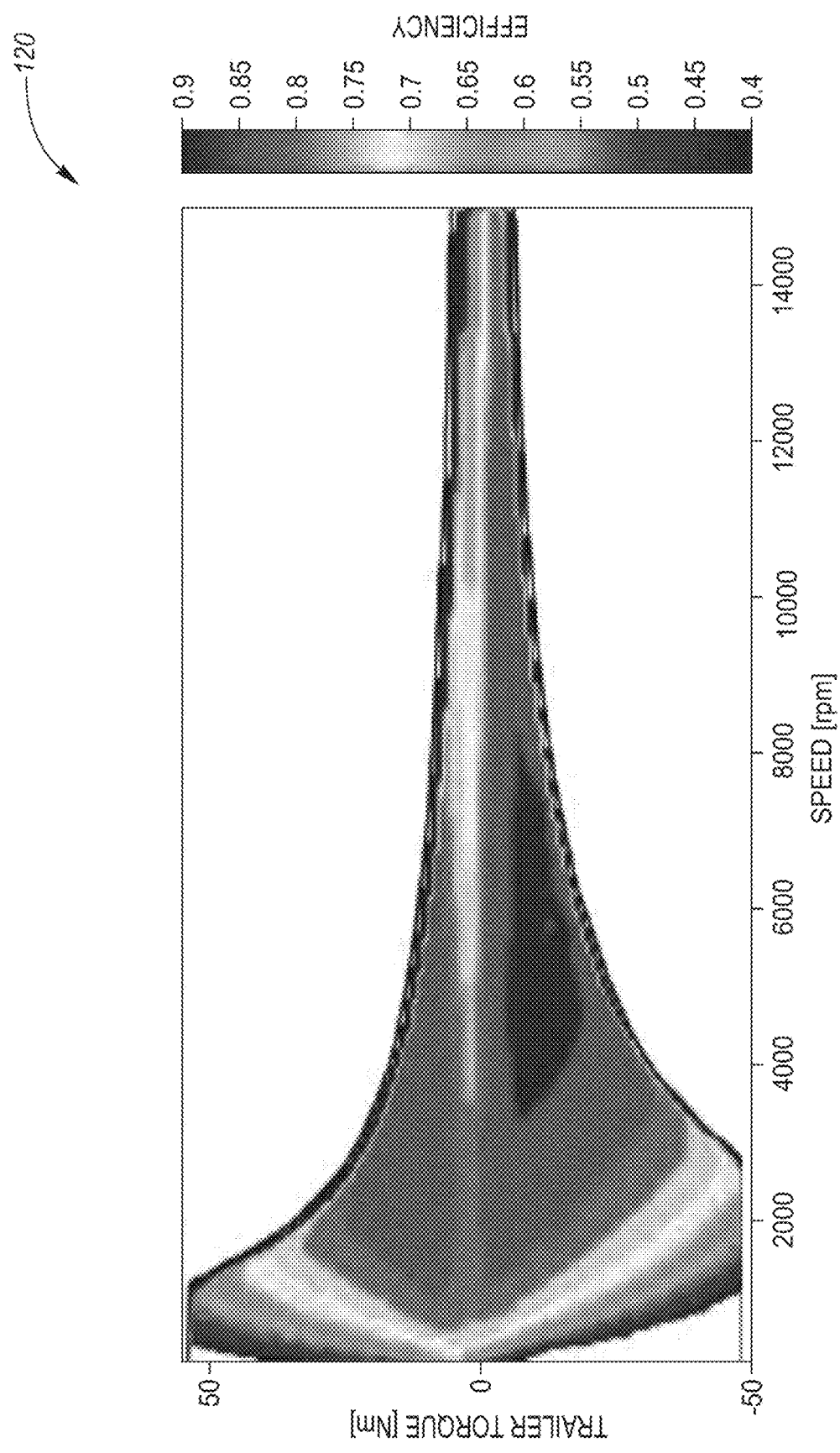
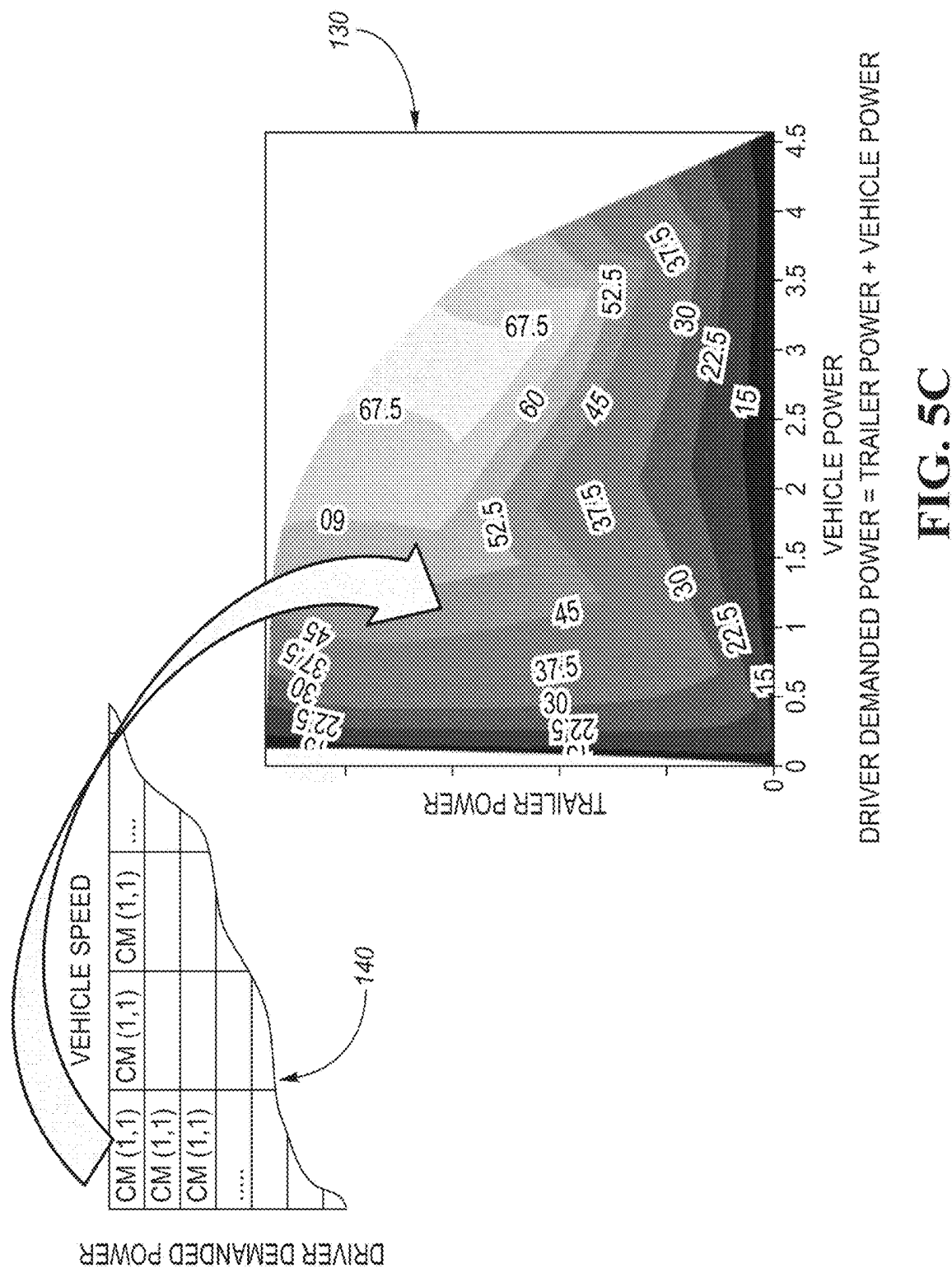


FIG. 5B



**ENERGY OPTIMIZATION FOR
INTEGRATED POWERTRAIN OF
COMBINED VEHICLE SYSTEM HAVING
VEHICLE AND ELECTRICALLY POWERED
TRAILER TOWED BY THE VEHICLE**

TECHNICAL FIELD

[0001] This disclosure relates to a combined vehicle system having a vehicle and an electrically powered trailer towed by the vehicle.

BACKGROUND

[0002] A combined vehicle system having a vehicle and an electrically powered trailer towed by the vehicle includes two powertrains. One of the powertrains belongs to the vehicle and the other one of the powertrains belongs to the trailer. An issue is how these two powertrains are to be operated due to the presence of one another.

SUMMARY

[0003] A system including a vehicle having an electrically powered trailer towed thereto is provided. The vehicle has a vehicle powertrain, and the trailer has a trailer powertrain. A controller is programmed to, in order to propel the vehicle with the trailer towed thereto, operate the vehicle powertrain and the trailer powertrain according to a combined efficiency map based on (i) an efficiency map associated with the vehicle powertrain and (ii) an efficiency map associated with the trailer powertrain.

[0004] The controller may be further programmed to control components of the vehicle powertrain and components of the trailer powertrain for efficiency or torque split based on operational status of the vehicle in realizing operation of the vehicle powertrain and the trailer powertrain according to the combined efficiency map.

[0005] The controller may be further programmed to obtain loss data of components of the vehicle powertrain and to obtain loss data of components of the trailer powertrain. In this case, the efficiency map associated with the vehicle powertrain may factor in the loss data of components of the vehicle powertrain and the efficiency map associated with the trailer powertrain may factor in loss data of components of the trailer powertrain. The loss data can be estimated from component models as well.

[0006] The combined efficiency map may provide for operation of the vehicle powertrain and the trailer powertrain to have a highest combined efficiency for a given driver demanded power and vehicle speed combination.

[0007] The combined efficiency map may provide for operation of the vehicle powertrain to be in a most efficient region of the efficiency map associated with the vehicle powertrain.

[0008] The vehicle powertrain may include a first traction battery, and the trailer powertrain may include a second traction battery. The vehicle powertrain may further include an engine. The controller may be further programmed to cause electrical power to be transferred between the first traction battery and the second traction battery in order to maintain operation of the vehicle powertrain and the trailer powertrain according to the combined efficiency map.

[0009] The controller may be further programmed to realize operation of the vehicle powertrain and the trailer

powertrain via torque vectoring between drive axles and propulsion sources on the vehicle powertrain and the trailer powertrain.

[0010] The controller may be further programmed to generate the combined efficiency map upon the trailer being connected to the vehicle.

[0011] The controller may be further programmed to share the efficiency map associated with the vehicle powertrain and/or the efficiency map associated with the trailer powertrain with a third party.

[0012] A control system for a combined vehicle system having a vehicle and an electrically powered trailer towed by the vehicle with the vehicle including a vehicle powertrain and the trailer including a trailer powertrain is also provided. The control system includes a controller programmed to, in order to propel the combined vehicle system, operate the vehicle powertrain and the trailer powertrain according to a combined efficiency map based on (i) an efficiency map associated with the vehicle powertrain and (ii) an efficiency map associated with the trailer powertrain.

[0013] A method is also provided. The method includes, upon an electrically powered trailer having a trailer powertrain being connected to a vehicle having a vehicle powertrain for the trailer to be towed by the vehicle, generating, by a controller of the vehicle, a combined efficiency map based on (i) an efficiency map associated with the vehicle powertrain and (ii) an efficiency map associated with the trailer powertrain. The method further includes operating, by the controller of the vehicle, in order to propel the vehicle with the trailer towed thereto, the vehicle powertrain and the trailer powertrain according to the combined efficiency map.

BRIEF DESCRIPTION OF THE DRAWINGS

[0014] FIG. 1 illustrates a perspective view of a combined vehicle system having an electrified vehicle (EV) and an electrically powered trailer towed by the EV;

[0015] FIG. 2A illustrates a block diagram of a vehicle powertrain of the EV and a trailer powertrain of the trailer;

[0016] FIG. 2B illustrates a block diagram of an integrated powertrain of the combined vehicle system, the integrated powertrain being comprised of the vehicle powertrain and the trailer powertrain;

[0017] FIG. 3 illustrates a flowchart depicting operation of an energy management optimization strategy for the integrated powertrain of the combined vehicle system;

[0018] FIGS. 4A and 4B illustrate a more detailed flowchart depicting operation of the energy management optimization strategy;

[0019] FIG. 5A illustrates an exemplary efficiency contour map for the vehicle powertrain;

[0020] FIG. 5B illustrates an exemplary efficiency contour map for the trailer powertrain; and

[0021] FIG. 5C illustrates an exemplary combined efficiency contour map for the integrated powertrain, the combined efficiency contour map being a function of the efficiency contour map for the vehicle powertrain and the efficiency contour map for the trailer powertrain, and the combined efficiency contour map being generated from an exemplary three-dimensional efficiency contour table for the integrated powertrain.

DETAILED DESCRIPTION

[0022] Embodiments are disclosed herein. It is to be understood, however, that the disclosed embodiments are merely examples and other embodiments may take various and alternative forms. The figures are not necessarily to scale. Some features may be exaggerated or minimized to show details of particular components. Therefore, specific structural and functional details disclosed herein are not to be interpreted as limiting, but merely as a representative basis for teaching one skilled in the art.

[0023] Referring now to FIGS. 1, 2A, and 2B, a combined vehicle system 10 will be described. FIG. 1 illustrates a perspective view of combined vehicle system 10. As shown in FIG. 1, combined vehicle system 10 includes an electrified vehicle (EV) 12 and an electrically powered trailer 14 towed by the EV. As shown in the block diagrams of FIGS. 2A and 2B, EV 12 includes a powertrain 16 and trailer 14 includes a powertrain 18. Thus, as shown in FIG. 2B, combined vehicle system 10 includes an integrated powertrain 52 (or “combined powertrain”) comprised of vehicle powertrain 16 and trailer powertrain 18.

[0024] Vehicle powertrain 16 includes a traction battery 20 for providing power to propel EV 12. In this example, vehicle powertrain 16 further includes an engine 22. As such, in this example, for instance, vehicle powertrain 16 is a “hybrid electric vehicle” (HEV) powertrain. In other embodiments, vehicle powertrain 16 does not include an engine. As such, in these other embodiments, for instance, vehicle powertrain 16 is a “battery-only electric vehicle” (BEV) powertrain.

[0025] Vehicle powertrain 16 further includes an electric motor 24 mechanically connected via a mechanical computing device 26 to a transmission 28. Transmission 28 is mechanically connected to a drive axle 30 that is mechanically connected to wheels 32 of EV 12. Engine 22 is also mechanically connected via mechanical computing device 26 to transmission 28. Electric motor 24 can provide propulsion capability for EV 12 while engine 22 is turned on or off.

[0026] Traction battery 20 stores electrical energy that can be used by electric motor 24 for propelling EV 12. Traction battery 20 typically provides a high-voltage (HV) direct current (DC) output. Traction battery 20 is electrically connected to electric motor 24 via an inverter 34 of vehicle powertrain 16. Inverter 34 provides the ability to transfer energy from traction battery 20 to electric motor 24. For example, traction battery 20 provides a DC voltage while electric motor 24 requires an alternating current (AC) input to function. Inverter 34 converts DC power from traction battery 20 into AC power to operate electric motor 24 for propelling EV 12.

[0027] Vehicle powertrain 16 further includes a controller (“vehicle controller”) 36. Vehicle controller 36 is operable to control and monitor the operation of the components of vehicle powertrain 16. Vehicle controller 36 includes electronics, software, or both, to perform control functions for operating vehicle powertrain 16. In embodiments, vehicle controller 36 is a combination vehicle system controller and powertrain control module (VSC/PCM). Although vehicle controller 36 is shown as a single device, vehicle controller 36 may include one or more controllers in the form of one or more hardware devices, or one or more software controllers with one or more hardware devices.

[0028] Trailer 14 is an electrified trailer in that trailer powertrain 18 includes a traction battery 40 for providing power to propel the trailer. In this example, unlike vehicle powertrain 16, trailer powertrain 18 does not include an engine. Trailer powertrain 18 further includes an electric motor 42. Electric motor 42 is mechanically connected to a drive axle 44 that is mechanically connected to wheels 46 of trailer 14. Traction battery 40 stores electrical energy that can be used by electric motor 42 for propelling trailer 14. Traction battery 40 is electrically connected to electric motor 42 via an inverter 48 of trailer powertrain 18. Inverter 48 converts DC power from traction battery 40 into AC power to operate electric motor 42 for propelling trailer 14.

[0029] Trailer powertrain 18 further includes a controller (“trailer controller”) 50. Trailer controller 50 is operable to control and monitor the operation of the components of trailer powertrain 18. Trailer controller 50 includes electronics, software, or both, to perform control functions for operating trailer powertrain 18. In embodiments, trailer controller 50 is a combination trailer system controller and powertrain control module (TSC/PCM). Although trailer controller 50 is shown as a single device, trailer controller 50 may include one or more controllers in the form of one or more hardware devices, or one or more software controllers with one or more hardware devices.

[0030] As indicated, trailer 14 is connected to EV 12 to form combined vehicle system 10. As EV 12 includes vehicle powertrain 16 and as trailer 14 includes trailer powertrain 18, combined vehicle system 10 includes integrated powertrain 52 comprised of vehicle powertrain 16 and trailer powertrain 18. This integration of trailer powertrain 18 with vehicle powertrain 16 into a combined powertrain can assist with increasing the range of EV 12 and enabling the EV to tow heavier loads.

[0031] Power and/or control hardware connection lines may extend between vehicle powertrain 16 and trailer powertrain 18 via the connection between EV 12 and trailer 14. As such, for instance, vehicle controller 36 and trailer controller 50 may communicate control signals with one another whereby the controllers may coordinate the operation of vehicle powertrain 16 and trailer powertrain 18, a controller of one of the powertrains may control the controller of the other powertrain whereby the one controller is effectively the sole controller for integrated powertrain 52, and the like; traction battery 20 and traction battery 40 may transfer power between one another such as for recharging; etc. Of course, such control communications may be done wirelessly.

[0032] For EV 12, regardless of the type of its powertrain architecture, there is a set of operational conditions of vehicle powertrain 16 for each torque demand, vehicle speed, vehicle mass, etc., in which the overall vehicle, after accounting for aerodynamics, weight, desired maneuver, etc., will have the highest system level energy efficiency.

[0033] In the exemplary example, vehicle powertrain 16 is a hybrid electric vehicle powertrain having electric motor 24 and engine 22 coupled with transmission 28 to propel EV 12. Due to the relatively sophisticated nature of how the components of vehicle powertrain 16 work together, an energy management strategy optimization and traction battery energy management calibration tool is used to define when and how much to charge and discharge traction battery 20, when and how much to use engine 22, etc. This tool takes

in inputs like fuel maps, electrical and mechanical losses, hardware configuration, gear ratio, vehicle road loads, state-of-charge (SOC) windows, drivability constraints, and the like to determine the optimal system efficiency of vehicle powertrain 16.

[0034] In further detail, HEV powertrains control the engine (e.g., engine on time including when to start and stop the engine, the engine power, etc.) and the traction battery (e.g., when the battery charges and discharges, the traction battery power, etc.) according to energy management calibration tables to change the operation conditions of the vehicle. Such energy management tables can be calibrated to operate the vehicle powertrains in desired regions to achieve an optimal system level fuel economy.

[0035] AWD (all-wheel drive) BEV powertrains have the ability to independently control torque distribution between the front and rear sets of wheels through independent motors on each axle. Due to various factors including electric motor efficiency and system losses varying based on speed and torque output, determining the optimal torque distribution for each axle with respect to a system energy management perspective is desired in order to optimize energy usage during vehicle operation. For an eAWD vehicle, optimal efficiency torque split can differ significantly from an equal proportion, especially in applications where the axle capacity varies. Operating in an efficiency split allows for a best possible energy efficiency. Switching to tables to define amount of torque to send to the different axles is employed to optimize vehicle overall performance.

[0036] Trailer powertrain 18 has the same type of architecture as an AWD BEV powertrain in that torque distribution to drive axle 44 is accomplished through electric motor 42 on the drive axle. Due to many factors including efficiency of electric motor 42 and system losses varying based on speed and torque output, determining the optimal torque distribution for drive axle 44 with respect to a system energy management perspective is desired to optimize energy usage during operation of trailer powertrain 18.

[0037] With a purpose of incorporating trailer 14 with EV 12 being to increase range and capability to the EV, an issue is how vehicle powertrain 16 and trailer powertrain 18 are to operate together. That is, an issue is how integrated powertrain 52, comprised of vehicle powertrain 16 and trailer powertrain 18, is to be operated.

[0038] Combined vehicle system 10 in accordance with this disclosure employs an energy management optimization strategy for integrated powertrain 52. The energy management optimization strategy factors in system level electrical and mechanical losses across various operation conditions for both vehicle powertrain 16 and trailer powertrain 18 to determine a most efficient (i.e., optimized) operation for integrated powertrain 52. This optimized operation for integrated powertrain 52 may be realized via energy management calibration for vehicle powertrain 16 (which is a HEV powertrain) and torque vectoring for trailer powertrain 18 (which is a BEV powertrain) (and for vehicle BEV powertrains) between each powered axle and propulsion source on EV 12 and trailer 14. A controller such as vehicle controller 36, in coordination with trailer controller 50, is configured to control vehicle powertrain 16 and trailer powertrain 18, and thereby control integrated powertrain 52, pursuant to the energy management optimization strategy.

[0039] Three general aspects are involved in developing and deploying the energy management optimization strat-

egy. The first aspect includes obtaining component level operation data (for efficiency) for each component in the architecture of integrated powertrain 52 and developing system level efficiency contour maps for vehicle powertrain 16 and trailer powertrain 18. The efficiency contour maps for vehicle powertrain 16 and trailer powertrain 18 are developed to map efficiencies across speed, torque, traction battery state-of-charge (SOC), and other relevant inputs. Much of this loss map and road load determination is typically completed during vehicle and trailer development.

[0040] The second aspect involves communicating information regarding system optimal efficiencies between vehicle controller 36 and trailer controller 50 upon trailer 14 being connected to EV 12. Using an in-vehicle or cloud-based optimization algorithm, such as used during vehicle development for HEV or BEV type vehicles, coupling between vehicle powertrain 16 and trailer powertrain 18 for optimal efficiency operation occurs to create a new combined system level efficiency contour map for integrated powertrain 52. Using this combined efficiency map for integrated powertrain 52, a new set of calibration tables may be created corresponding to the architecture of integrated powertrain 52. These calibrations are uploaded to vehicle controller 36. Vehicle controller 36 can thereby control vehicle powertrain 16 in a desired region of the combined efficiency map for integrated powertrain 52 and/or maintain optimal torque split between powered axles of EV 12 and trailer 14 based on the current operational status of EV 12.

[0041] The optimization algorithm initially focuses on optimizing for most commonly encountered conditions to minimize wait/down time after coupling of trailer 14 to EV 12. A base calibration providing a generic vehicle to trailer operation can be employed until a more refined optimized calibration can be finished. Once a calibration is created based on a vehicle and a trailer, the calibrations can be used for other users by identifying similar vehicle and trailer characteristics. Essentially pre-built energy optimized calibrations can be accessed via the cloud by users of different vehicles. These pre-built combined vehicle energy management optimized calibrations can be uploaded to vehicles via over-the-air updates.

[0042] The communicating of information regarding system optimal efficiencies between vehicle controller 36 and trailer controller 50 upon trailer 14 being connected to EV 12 may occur using wireless (e.g., Bluetooth, vehicle modem, etc.) or wired (e.g., controller area network (CAN)/local area network (LAN)) communications. Information indicative of EV 12 of the communicated information may include information such as powertrain architecture and capability (power split, BEV, etc.) of vehicle powertrain 16, loss maps associated with the vehicle powertrain, road loads (F terms), size/capacity of traction battery 20, existing optimal vehicle powertrain operation, and weight and towing capacity of the EV. Information indicative of trailer 14 of the communicated information may include information such as information of trailer powertrain 16 including number of motors, number of wheels, road loads, loss maps, size/capacity of traction battery 40, capability of the trailer powertrain, and weight of the trailer. The loss maps can be estimated from component models as well.

[0043] The energy management-based optimization occurs based on powertrain architecture and control strategy used on the vehicle. Main inputs from driving the vehicle include traction battery SOC, driver demanded wheel

torque, vehicle speed, combined vehicle weight, and relative powertrain capability. Optimization outputs can include for all powertrain types, torque split between vehicle and trailer and powered axles on trailer during regenerative braking and propulsion; for BEV powertrains, torque split between axles; for modular hybrid transmission (MHT) powertrains, when to shift, engine on time including when to start and stop the engine, traction battery charging, etc.; and for power-split powertrains, engine speed, traction battery charging, when to start and stop the engine, etc.

[0044] Further, calibration ranges can also be predetermined such that the calibration that is created for optimal energy management does not violate predetermined noise, vehicle, harshness (NVH)/drivability rules. Some of these rules could be embedded to algorithms/rules which create the calibration tables from the system efficiency maps.

[0045] The third aspect involves applying optimal torque vectoring calibration to EV 12 and trailer 14 during steady state and varying operating conditions. Adjustments from optimal operating conditions during driving of combined vehicle system 10 based on encountered operational conditions are made. For example, in the case of excessive thermal status of components such as traction battery 20 and/or electric motor 24 of vehicle powertrain 16 becoming very hot, an adjustment is to switch to more propulsion from trailer powertrain 18; in the case of excessive wheel torque being demanded, an adjustment is to apply full torque capability from both vehicle powertrain 16 and trailer powertrain 18; in the case of excessive slip being detected, an adjustment is to request equally distributed torque split, up to capacity, across all axles; and in the case of instability of trailer 14, an adjustment is requesting trailer powertrain 18 based instability arresting torque as needed.

[0046] Further, during the optimization, rules can be applied to account for drivability/NVH concerns such as leveraging existing drivability/efficiency optimized calibrations within EV 12 and trailer 14. For example, in AWD BEVs, generally, it is advantageous to command at least some torque on the secondary drive unit (SDU) (even though entirely primary drive unit (PDU) based torque is more efficient) for drivability, lash, efficiency, and other vehicle performance/drivability concerns.

[0047] Referring now to FIG. 3, with continual reference to FIGS. 1, 2A, and 2B, a flowchart 60 depicting operation of an energy management optimization strategy for integrated powertrain 52 of combined vehicle system 10 is shown. The operation begins with developing efficiency contour maps for vehicle powertrain 16 and trailer powertrain 18 using component data system level information of the vehicle and trailer powertrains, as indicated by process block 62. The efficiency contour maps are stored in onboard data storage of EV 12 and/or trailer 14 or in the cloud, as indicated by process block 64. Trailer 14 is then connected to EV 12 to form combined vehicle system 10 which includes integrated (combined) powertrain 52 comprised of vehicle powertrain 16 and trailer powertrain 18, as indicated by process block 66. A user of combined vehicle system 10 then selects the model or type of trailer 14 such as via a human-machine interface (HMI) of EV 12, as further indicated by process block 66. This selection is for identifying trailer 14 whereby the efficiency contour map for the particular model or type of trailer powertrain 18, as opposed to

the efficiency contour map for some other model or type of trailer powertrain, will be used for the energy management optimization strategy.

[0048] The operation further includes creating a combined efficiency contour map for integrated powertrain 52 from the individual efficiency contour maps for vehicle powertrain 16 and trailer powertrain 18, as indicated by process block 68. The individual efficiency contour maps for vehicle powertrain 16 and trailer powertrain 18 are selected from onboard data storage or downloaded from the cloud, as further indicated by process block 68. Integrated powertrain 52 is then operated according to the combined efficiency contour map to propel EV 12, with trailer 14 connected thereto, as indicated by process block 70. The operation of integrated powertrain 52 is handled by vehicle controller 36 alone or in conjunction with trailer controller 50 as described above. The operation of integrated powertrain 52 is adapted based on encountered conditions, as further indicated by process block 70.

[0049] Referring now to FIGS. 4A and 4B, with continual reference to FIGS. 1, 2A, 2B, and 3, a more detailed flowchart 80 depicting operation of the energy management optimization strategy is shown. For simplicity, it is assumed that vehicle controller 36 alone and without trailer controller 50 handles aspects of the operation.

[0050] The operation begins with vehicle controller 36 detecting whether an electrically powered trailer has been connected to EV 12, as indicated by decision block 82. The operation continues once an electrically powered trailer, such as trailer 14, is connected to EV 12. Upon continuation of the operation, vehicle controller 36 sources loss data for each torque producing device (e.g., electric motor 24), traction battery (e.g., traction battery 20), and gearbox system (e.g., transmission 28) of vehicle powertrain 16 and sources loss data for each torque producing device (e.g., electric motor 42) and traction battery (e.g., traction battery 40) of trailer powertrain 18, as indicated by process block 84. As described, the loss map creation occurs during development of the vehicle powertrain/vehicle and during development of the trailer powertrain/trailer. Optimal torque vectoring between each electric motor is predetermined based on wheel torque and vehicle (trailer) speed.

[0051] Vehicle controller 36 then determines how many and which types of motor/torque producing devices are on vehicle powertrain 16 (and on trailer powertrain 18), as indicated by process block 86.

[0052] A maximum system efficiency algorithm is then used to calculate a most efficient operating point for every torque demand/vehicle speed combination within the combined vehicle capability for the individual vehicle powertrain 16 and trailer powertrain 18, as indicated by process block 88. The algorithm iterates through each potential speed/torque combination/option to meet a desired torque/power request by changing the torque/power produced by each torque producing device on vehicle powertrain 16 and trailer powertrain 18 (trailer powertrain 18 produces "X" Nm, vehicle powertrain 16 produces "Y" Nm, loss is "Z" Nm, and repeat). Once the algorithm iterates through each operating point, the algorithm determines the torque split/operating conditions for each point with the lowest energy consumption/lowest loss. These are the ideal operating points for integrated powertrain 52 and are programmed into the contour map and combined vehicle calibration. In conjunction with process block 88, a combined powertrain

efficiency plot for integrated powertrain 52 is generated using an iterative-perturbation algorithm, as indicated by process block 90.

[0053] The operation of process blocks 88 and 90 will be described in further detail with reference to FIGS. 5A, 5B, and 5C. FIG. 5A illustrates an exemplary efficiency contour map 110 for vehicle powertrain 16; FIG. 5B illustrates an exemplary efficiency contour map 120 for trailer powertrain 18; and FIG. 5C illustrates an exemplary combined efficiency contour map 130 for integrated powertrain 52 that is generated from an exemplary three-dimensional (3D) efficiency contour table 140 for the integrated powertrain. As described, the operation of process blocks 88 and 90 entails creating a combined efficiency contour map (i.e., combined efficiency contour map 130) for integrated powertrain 52 from the individual efficiency contour maps (i.e., efficiency contour map 110 for vehicle powertrain 16 and efficiency contour map 120 for trailer powertrain 18). In this way, combined efficiency contour map 130 for integrated powertrain 52 is a function of (i) efficiency contour map 110 for vehicle powertrain 16 and (ii) efficiency contour map 120 for trailer powertrain 18.

[0054] More particularly, as noted, the maximum system efficiency algorithm determines the torque split/operating conditions for each point with the lowest energy consumption/lowest loss. These are the ideal operating points for integrated powertrain 52. The operating conditions are a function of driver demanded power and vehicle speed; and the driver demanded power is the demanded power of combined vehicle system 10 (i.e., the driver demanded power is the sum of (i) the power of vehicle powertrain 16 and (ii) the power of trailer powertrain 18). Initially, the maximum system efficiency algorithm uses efficiency contour map 110 for vehicle powertrain 16 and determines power to be drawn from vehicle powertrain 16 that is in the most efficient region of efficiency contour map 110. Next, the maximum system efficiency algorithm uses efficiency contour map 120 for trailer powertrain 18 and determines efficiency for the remaining power to be drawn from trailer powertrain 18 (vehicle speed is a constant). (The remaining power is the power from vehicle powertrain 16 subtracted from the driver demanded power.) The maximum system efficiency algorithm then runs an iterative algorithm, which is based on perturbation, to determine whether the combined efficiency could be increased with different torque splits. This information is stored as three-dimensional efficiency contour table 140, which converts into combined efficiency contour map 130 for integrated powertrain 52. Combined efficiency contour map 130 is a contour map for integrated powertrain 52 that is generated for each driver demanded power and vehicle speed combination of combined vehicle system 10.

[0055] Turning back to FIG. 4, the operation may further continue with creating additional contour maps using the same algorithm based on different powertrain component operating temperatures when the loss data is available, as indicated by process block 92. Alternative contour maps for situations in which power/torque is limited from motor/battery up to X % derate may be created, as indicated by process block 94.

[0056] The operation further includes EV 12 uploading combined efficiency contour map 130 for integrated powertrain 52 to the associated powertrain controllers that determine torque/power requests, as indicated by process

block 96. In this example, combined efficiency contour map 130 is uploaded to vehicle controller 36.

[0057] Operation of combined vehicle system 10 by a driver of EV 12 then begins, as indicated by process block 98. During operation of combined vehicle system 10, vehicle controller 36 determines vehicle operation inputs such as driver demanded torque/power, vehicle speed, motor speeds, and the like, as indicated by process block 100. In turn, vehicle controller 36 uses operational parameters as inputs in combined efficiency contour map 130 for integrated powertrain 52 and determines therefrom the optimal torque distribution amongst torque producing devices in the powertrains (i.e., vehicle powertrain 16 and trailer powertrain 18) comprising the integrated powertrain, as indicated by process block 102.

[0058] The operation may further include obtaining torque limiting factors such as SOC, component temperatures for individual components, and NVH/drivability limits, and limiting the torque, and accordingly creating or referencing new contour maps, as indicated by process block 104.

[0059] The description of the operation of flowchart 80 is exemplary. Other considerations that could be accounted for by creating new contour maps can include: NVH constraints; thermal derate of traction battery/engine/electric motors/inverter/gearbox/etc. on the vehicle and/or the trailer (separate contour maps can be created for either scenario based on detecting a thermal derate event); stability constraints during braking; and supply energy constraints and trip metrics.

[0060] Regarding supply energy constraints and trip metrics, based on the estimated length of the trip and the energy available for use on the vehicle/trailer, certain powertrains could be leveraged more to ensure both powertrains can provide propulsion assistant throughout the duration of the trip. When the vehicle powertrain is gas based and the trailer powertrain is electrified, the trailer might have a more limited energy supply than the vehicle and as a result the output torque should be rationed or regen braking should occur as frequently as possible to recharge the traction battery of the trailer. When the vehicle powertrain is of a hybrid variant, the engine can run to charge the traction batteries of the vehicle and the trailer and separate contour maps should be created. When both powertrains are electrified and the energy storage system on the trailer is separate from the vehicle, biasing the torque of the units may be needed. When both powertrains are electrified and a shared energy system is used (the situation that is assumed in flowchart 80), the true most efficient operating conditions can be realized throughout the trip due to the vehicle/trailer having the ability to pull energy from the same energy storage device or transfer energy between storage devices.

[0061] As described, this disclosure relates to an electrically powered vehicle and an electrically powered trailer towed by the vehicle. The trailer includes its own powertrain having a traction battery and an electric motor for propelling the trailer. A combined vehicle-trailer integrated powertrain energy management method is provided that factors in system level electrical and mechanical losses across all operation conditions to determine a most efficient powertrain operation for the combined vehicle powertrain system. The method uses torque vectoring between each powered axle and propulsion source on the vehicle and the trailer. The method maps efficiency contour maps for the vehicle and the trailer for various inputs such as speed,

torque, SOC, and others. The method develops a new combined efficiency contour map when the trailer is connected to the vehicle, updates the calibration tables for the various vehicle and trailer components, and controls the components for efficiency or torque split based on the operational status of the vehicle. The unique calibrations or contour maps for a vehicle and trailer pair may be shared with other vehicles such as via V2X.

[0062] As described, advantages of the integrated powertrain energy management optimization strategy include being implementable for various types of vehicle and trailer powertrain architectures, allowing for electrified vehicles to tow heavier trailer loads, reducing need for tow specific capable vehicles, increasing vehicle range, and leveraging existing vehicle and trailer data to optimize usage of the integrated vehicle/trailer powertrain.

[0063] The algorithms, methods, or processes disclosed herein can be deliverable to or implemented by a computer, controller, or processing device, which can include any dedicated electronic control unit or programmable electronic control unit. Similarly, the algorithms, methods, or processes can be stored as data and instructions executable by a computer or controller in many forms including, but not limited to, information permanently stored on non-writable storage media such as read only memory devices and information alterably stored on writable storage media such as compact discs, random access memory devices, or other magnetic and optical media. The algorithms, methods, or processes can also be implemented in software executable objects. Alternatively, the algorithms, methods, or processes can be embodied in whole or in part using suitable hardware components, such as application specific integrated circuits, field-programmable gate arrays, state machines, or other hardware components or devices, or a combination of firmware, hardware, and software components. The words processor and processors may be interchanged herein, as may the words controller and controllers.

[0064] While exemplary embodiments are described above, it is not intended that these embodiments describe all possible forms encompassed by the claims. The words used in the specification are words of description rather than limitation, and it is understood that various changes may be made without departing from the spirit and scope of the disclosure. Additionally, the features of various implementing embodiments may be combined to form further embodiments.

What is claimed is:

1. A system comprising:

- a vehicle having an electrically powered trailer towed thereto, the vehicle having a vehicle powertrain and the trailer having a trailer powertrain; and
- a controller programmed to, in order to propel the vehicle with the trailer towed thereto, operate the vehicle powertrain and the trailer powertrain according to a combined efficiency map based on (i) an efficiency map associated with the vehicle powertrain and (ii) an efficiency map associated with the trailer powertrain.

2. The system of claim 1 wherein:

the controller is further programmed to control components of the vehicle powertrain and components of the trailer powertrain for efficiency or torque split based on operational status of the vehicle in realizing operation of the vehicle powertrain and the trailer powertrain according to the combined efficiency map.

3. The system of claim 1 wherein:

the controller is further programmed to obtain loss data of components of the vehicle powertrain and to obtain loss data of components of the trailer powertrain; and the efficiency map associated with the vehicle powertrain factors in the loss data of components of the vehicle powertrain and the efficiency map associated with the trailer powertrain factors in loss data of components of the trailer powertrain.

4. The system of claim 1 wherein:

the combined efficiency map provides for operation of the vehicle powertrain and the trailer powertrain to have a highest combined efficiency for a given driver demanded power and vehicle speed combination.

5. The system of claim 1 wherein:

the combined efficiency map provides for operation of the vehicle powertrain to be in a most efficient region of the efficiency map associated with the vehicle powertrain.

6. The system of claim 1 wherein:

the vehicle powertrain includes a first traction battery; and the trailer powertrain includes a second traction battery.

7. The system of claim 6 wherein:

the vehicle powertrain further includes an engine.

8. The system of claim 6 wherein:

the controller is further programmed to cause electrical power to be transferred between the first traction battery and the second traction battery in order to maintain operation of the vehicle powertrain and the trailer powertrain according to the combined efficiency map.

9. The system of claim 1 wherein:

the controller is further programmed to realize operation of the vehicle powertrain and the trailer powertrain via torque vectoring between drive axles and propulsion sources on the vehicle powertrain and the trailer powertrain.

10. The system of claim 1 wherein:

the controller is further programmed to generate the combined efficiency map upon the trailer being connected to the vehicle.

11. The system of claim 1 wherein:

the controller is further programmed to share the efficiency map associated with the vehicle powertrain and/or the efficiency map associated with the trailer powertrain with a third party.

12. A control system for a combined vehicle system having a vehicle and an electrically powered trailer towed by the vehicle, the vehicle having a vehicle powertrain and the trailer having a trailer powertrain, the control system comprising:

a controller programmed to, in order to propel the combined vehicle system, operate the vehicle powertrain and the trailer powertrain according to a combined efficiency map based on (i) an efficiency map associated with the vehicle powertrain and (ii) an efficiency map associated with the trailer powertrain.

13. The control system of claim 12 wherein:

the controller is further programmed to obtain loss data of components of the vehicle powertrain and to obtain loss data of components of the trailer powertrain; and the efficiency map associated with the vehicle powertrain factors in the loss data of components of the vehicle powertrain and the efficiency map associated with the trailer powertrain factors in loss data of components of the trailer powertrain.

- 14.** The control system of claim **12** wherein:
the combined efficiency map provides for operation of the vehicle powertrain to be in a most efficient region of the efficiency map associated with the vehicle powertrain.
- 15.** The control system of claim **12** wherein:
the controller is further programmed to cause electrical power to be transferred between a traction battery of the vehicle powertrain and a traction battery of the trailer powertrain in order to maintain operation of the vehicle powertrain and the trailer powertrain according to the combined efficiency map.
- 16.** The control system of claim **12** wherein:
the controller is further programmed to realize operation of the vehicle powertrain and the trailer powertrain via torque vectoring between drive axles and propulsion sources on the vehicle powertrain and the trailer powertrain.
- 17.** A method comprising:
upon an electrically powered trailer having a trailer powertrain being connected to a vehicle having a vehicle powertrain for the trailer to be towed by the vehicle,

- generating, by a controller of the vehicle, a combined efficiency map based on (i) an efficiency map associated with the vehicle powertrain and (ii) an efficiency map associated with the trailer powertrain; and
operating, by the controller of the vehicle, in order to propel the vehicle with the trailer towed thereto, the vehicle powertrain and the trailer powertrain according to the combined efficiency map.
- 18.** The method of claim **17** further comprising:
controlling, by the controller of the vehicle, components of the vehicle powertrain and components of the trailer powertrain for efficiency or torque split based on operational status of the vehicle in realizing operation of the vehicle powertrain and the trailer powertrain according to the combined efficiency map.
- 19.** The method of claim **17** wherein:
the vehicle powertrain includes a first traction battery; and
the trailer powertrain includes a second traction battery.
- 20.** The method of claim **19** wherein:
the vehicle powertrain further includes an engine.

* * * * *



US006424907B1

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Rieth et al.

(10) **Patent No.: US 6,424,907 B1**
 (45) **Date of Patent: Jul. 23, 2002**

(54) **METHOD AND DEVICE FOR
 DETERMINING AND DETECTING THE
 OVERTURNING HAZARD OF A VEHICLE**

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(52) **U.S. Cl.** **701/124**; 701/29; 340/438;
 340/440

(58) **Field of Search** 701/1, 70, 72,
 701/79, 80, 124, 29; 180/170, 197; 340/431,
 438, 440

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 PLLC

(57) **ABSTRACT**

A method and device is disclosed for determining and detecting the rollover hazard of a vehicle through a dynamic detection of variations in the center of gravity of the vehicle. During cornering of the vehicle, first condition variables, which correspond to the respective wheel load, are detected on at least two wheels. The detected first condition variables are compared to reference values representative of the respective cornering maneuver. A corresponding change in the center of gravity is calculated from the differences between the detected first condition variables and the reference values.

8 Claims, 4 Drawing Sheets

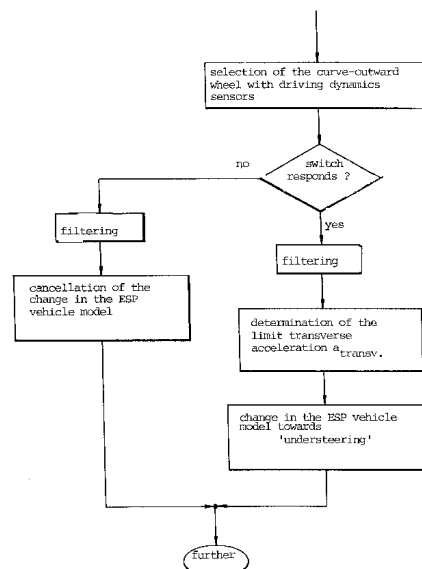
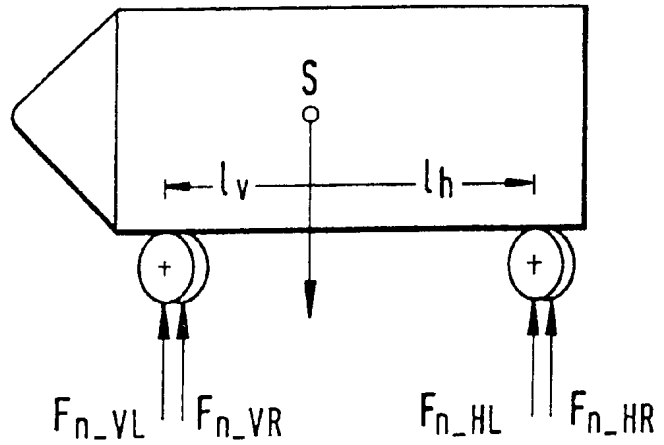


Fig. 1a

PRIOR ART

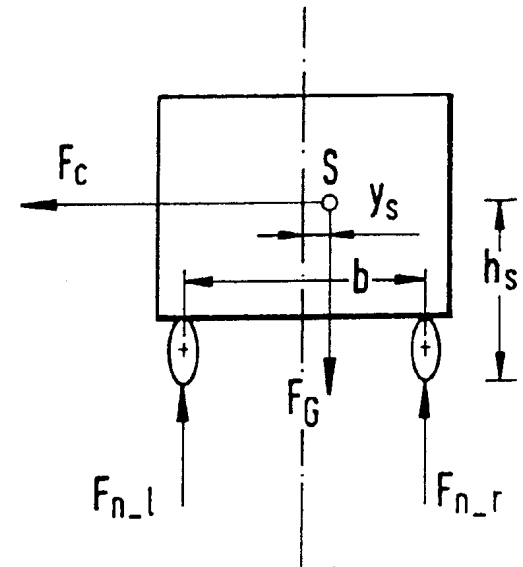


$$F_G = F_{n_VL} + F_{n_VR} + F_{n_HL} + F_{n_HR}$$

and in the static case for an axle

Fig. 1b

PRIOR ART



$$: F_{G_A}^{xle} * (b/2 + y_s) = F_n * b$$

Fig. 2

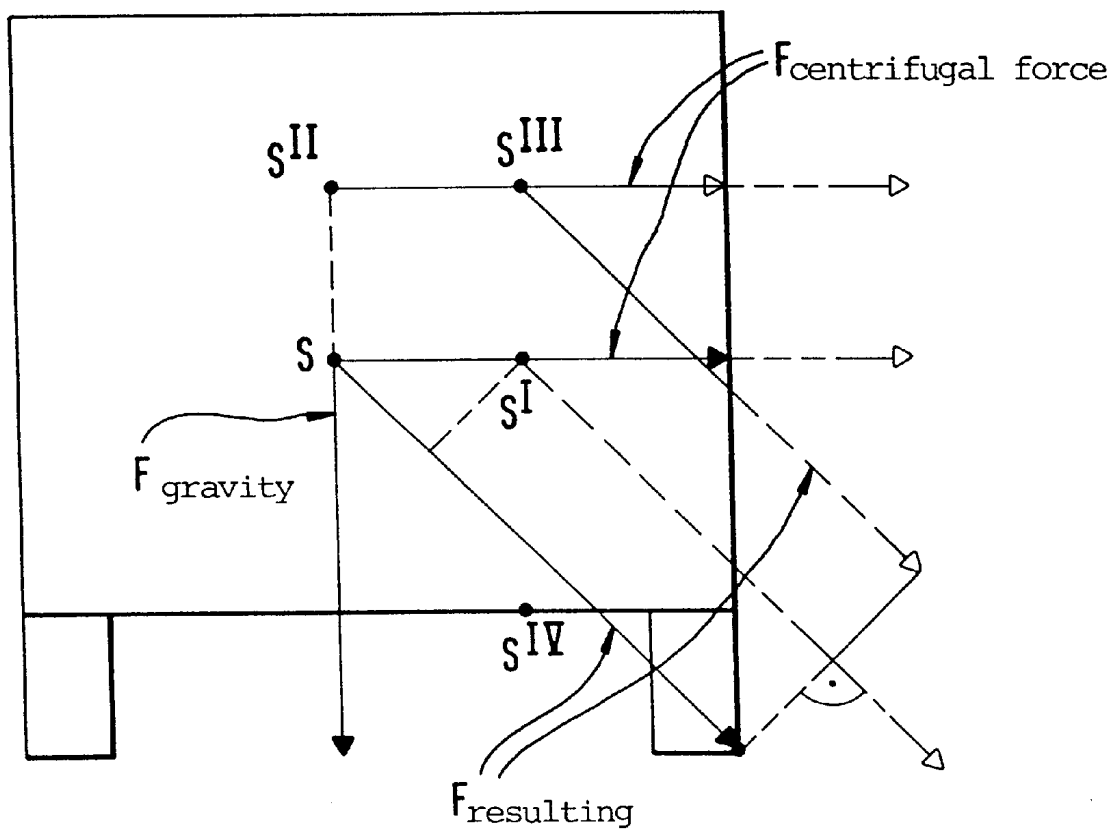


Fig. 3

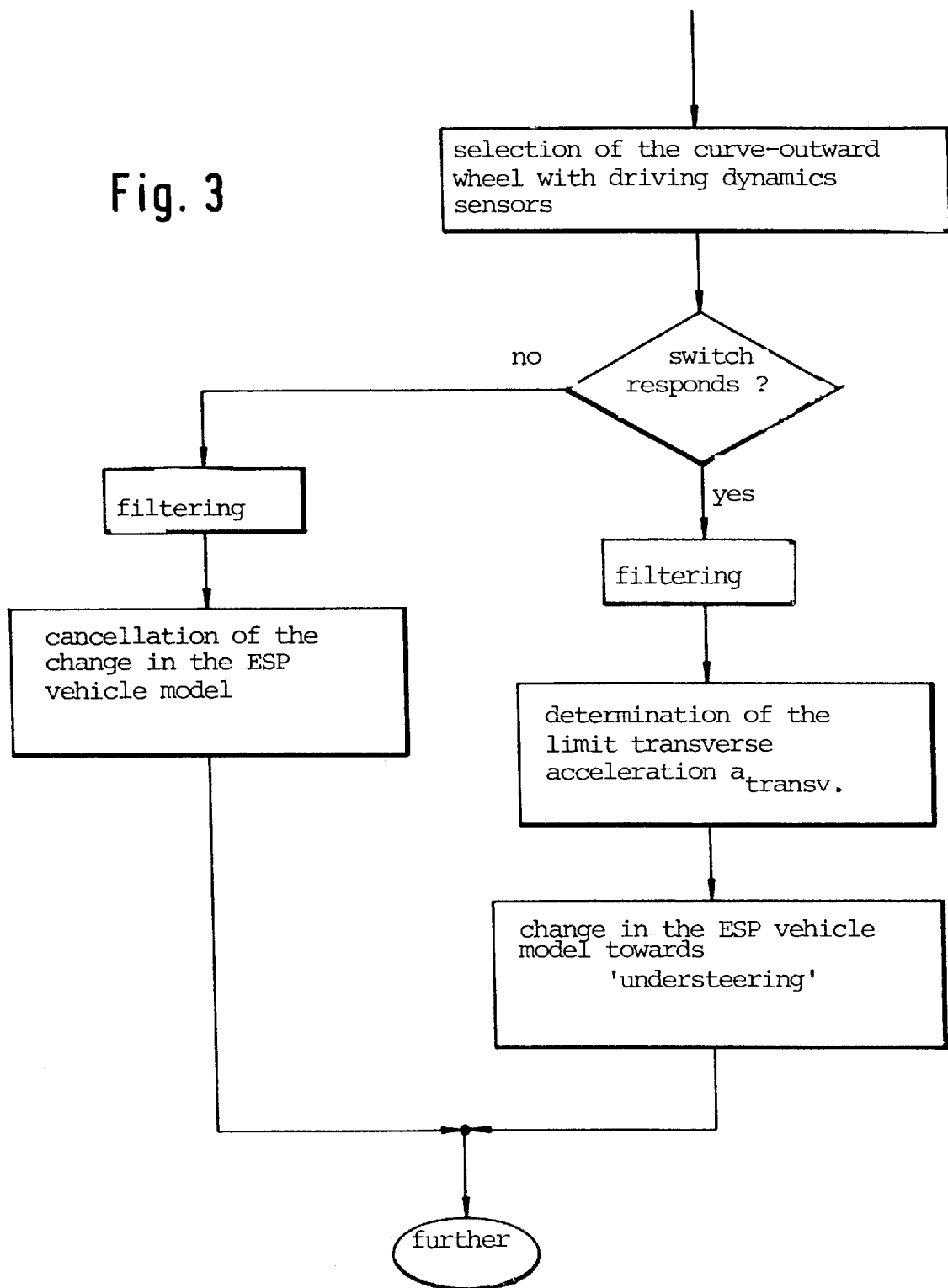


Fig. 4a

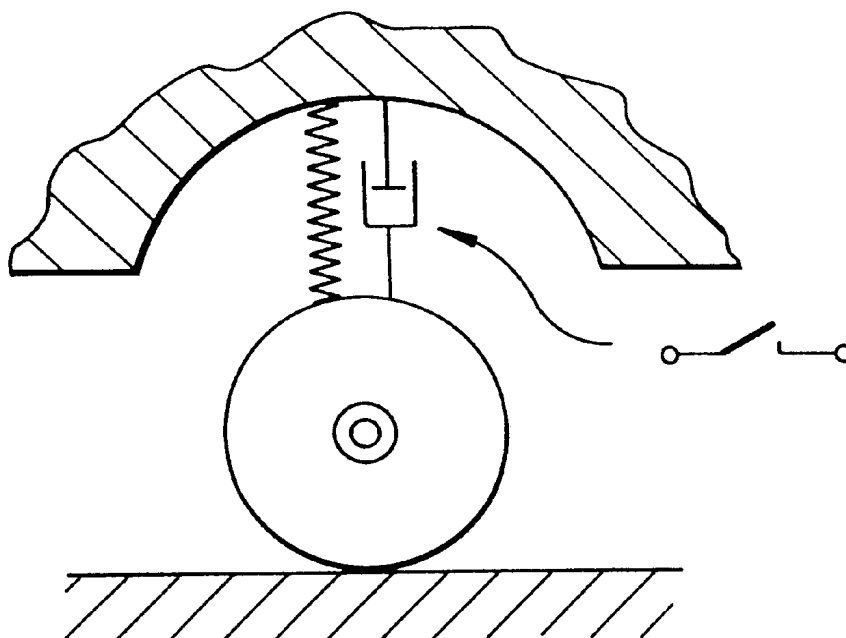
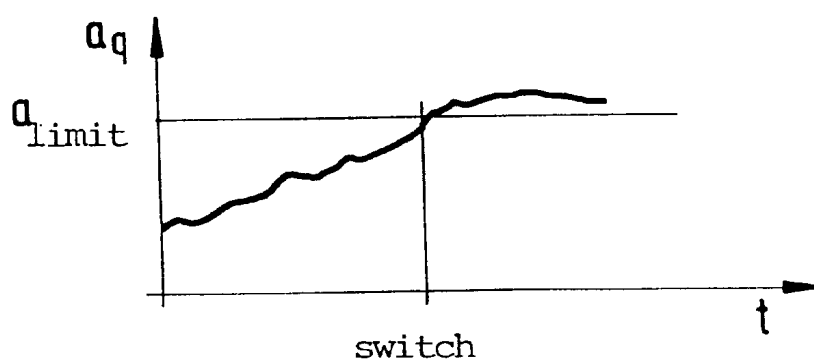


Fig. 4b



METHOD AND DEVICE FOR DETERMINING AND DETECTING THE OVERTURNING HAZARD OF A VEHICLE

CROSS-REFERENCES TO RELATED APPLICATIONS

The present application is a U.S. national phase application of International Application PCT/EP99/05079 (International Publication WO00103900), filed Jul. 16, 1999.

TECHNICAL FIELD

The present invention generally relates to systems for measuring vehicle stability, and more particularly relates to a method and a related device for detecting variations of the center of gravity of the mass of a vehicle.

BACKGROUND OF THE INVENTION

Various models for rollover accidents are described on pages 309 to 333, chapter 9, in "Fundamentals of Vehicle Dynamics", by T. D. Gillespie, Society of Automotive Engineers, Inc., Warrendale 1992. Starting with a quasi-stationary model for a rigid vehicle, via a quasi-stationary model for a sprung vehicle up to dynamic models in consideration of inherent roll frequencies, conditions for existing rollover hazards are indicated.

While it was known at the time the above-mentioned book was published that in trucks, truck trailer units, buses, small buses, and off-road vehicles having a high center of gravity and/or a small tread width, there is a rollover risk during cornering in the event of a major roll motion, it has been found only recently that lateral motions of passenger vehicles are also likely to build up until the vehicle rolls over. Such a risk of rollover is considerably increased by improper loading, e.g. extremely toward one side or on the vehicle roof, because the center of gravity of the mass of the vehicle is shifted upwards or to one side.

As is shown in FIGS. 1a and 1b, the analysis of the forces and moments is decisive in the static case for the stable equilibrium of a vehicle.

FIG. 1a shows the distribution of the vehicle weight force F_G to the wheel tread forces $F_{A_{xx}}$. The lateral offset y_s of the center of gravity S from the vehicle center is shown in FIG. 1b. The offset of moments with respect to a wheel shows the influence of the shift of the center of gravity in the static and dynamic case.

If, during driving operations, a disturbing force occurs, such as the centrifugal force during cornering or a wheel load variation due to a rough road section, the vehicle will move into the labile or unstable condition depending on the magnitude of the disturbing force.

With an Electronic Stability Program (ESP), for example, the system developed and sold by the applicant, a system is on hand which is apt to detect critical driving conditions by comparing the course of driving predefined by a driver with the actual track of the vehicle, and to prevent these conditions by a well-defined brake intervention. Wheel rotational speeds, the transverse acceleration, the steering angle, and the yaw rate are provided as data for this purpose. In case the driving condition defined by these data reaches a critical range, the proper steering behavior of the vehicle defined by the vehicle model of the ESP controller will be maintained by a wheel-selective brake intervention. Thus, this control action can only influence all two-dimensional processes and condition variables, that means processes which can be

described by transverse acceleration and rotation about the vertical vehicle axis (yaw).

Once the position of the center of gravity in a longitudinal and/or transverse direction of the vehicle changes, e.g., due to loading, the changed cornering limit speeds are automatically taken into account in the ESP system because the proper steering behavior determined in the vehicle model shall be maintained. When the center of gravity shifts, e.g. in a rearward direction, the augmented oversteering tendency during cornering is counteracted by reducing the allowed cornering speed, in the event that the basic setting is established as understeering.

A change in the height of the center of gravity will now cause a changed side-tilt behavior of the vehicle in longitudinal and transverse directions. When a transverse acceleration acts on the vehicle, according to FIG. 1b, the analysis of moments

$$F_c \times h_s + F_{n_{r,l}} \times b = F_G \times (b/2 + y_s) \quad (1)$$

applies, wherein F_c designates the centrifugal force due to the transverse acceleration a_{transv} , F_G is the center of gravity, m is the vehicle mass, $F_{n_{r,l}}$ designates the vertical wheel force (wheel load) for the right and the left wheel, h_s , y_s the height of the center of gravity, y_s the lateral distance of the center of gravity from the vehicle center, and b designates the tread width. From this follows directly the correlation for the transverse acceleration a_{transv} .

$$a_{transv} = 1/h_s \times [g(b/2 + y_s) - F_{n_{r,l}} \times b/m] \quad (2)$$

Rollover of the vehicle will thus be reached when a critical transverse acceleration is exceeded for which the following correlation applies:

$$a_{transv,critical} = (b/2 + y_s)/h_s \times g \quad (3)$$

Thus, the critical transverse acceleration depends directly on the center of gravity.

Besides, in prior art vehicle motion control systems, wherein the objective is to increase driving stability, especially during cornering, that means to avoid lateral rollover in particular, it is generally the transverse vehicle acceleration or the roll angle of the vehicle which is taken as a basis for critical condition variables that describe the rollover hazard.

Thus, DE-A 197 46 889 describes a system to increase the lateral stability during cornering which is equipped with a side-tilt detection device. This side-tilt detection device measures either the difference in height between the right and the left vehicle side, or the transverse acceleration of the vehicle in order to detect the roll angle between the vehicle horizontal and the roadway horizontal. When the side-tilt detection device detects a rollover hazard, a countersteering yaw torque will be produced by decelerating the curve-outward front wheel.

However, as described hereinabove, the allowed transverse acceleration and the allowed roll angle are greatly responsive to the position of the center of gravity of the vehicle, especially the height of the center of gravity. In the generic methods and devices known from the state of the art, the center of gravity of the vehicle, especially variations in the center of gravity, is not taken into consideration to a sufficient degree as far as a preventive rollover prediction is concerned.

BRIEF SUMMARY OF THE INVENTION

Please delete the abstract on page 13 and replace it with the abstract set forth immediately below in clean form.

Additionally, in accordance with 37 CFR 1.121(b)(iii), all paragraphs amended herein are set forth in a marked up version on the sheets attached to this amendment.

Therefore, an object of the present invention is to provide a method and a device which render it possible to counteract a rollover hazard as early as possible. The primary objective of the present invention is a prediction of an imminent rollover condition of a vehicle to permit a best possible prevention.

According to the present invention, this object is achieved in that first condition variables which correspond to the respective wheel load are detected on at least two wheels during cornering, in that the detected first condition variables are compared with reference values representative of the respective cornering maneuver, and in that a related variation of the center of gravity is calculated from the differences between the detected first condition variables and the reference values.

The present invention is based on the recognition that the most appropriate coefficient of influence in a best possible preventive detection of the rollover hazard of a vehicle is the position of the center of gravity of the vehicle. Thus, the current center of gravity as an output quantity is made the important basis of the method and the device disclosed in the present invention.

The coefficient of influence directly indicates the existence of a critical rollover condition. Either active control interventions, e.g. by means of ESP, or a passive warning to the driver will be considered as a countermeasure to prevent rollover.

In contrast to the state of the art, the preventive procedure disclosed by the present invention includes the advantage that the vehicle remains steerable during a correcting intervention required and the driving comfort is also preserved during an active control intervention. Another advantage of the present invention over the state of the art can be seen in that the possible intervention strategies in response to detection of a critical driving situation are almost optional, not least due to the detection which offers extreme prediction. Therefore, the present invention can be realized without special effort and cost as an extension or an improvement of an existing driving stability control system, for example, the Electronic Stability Program (ESP) of the applicant.

The spring travel that can be measured at the wheel suspension, the spring pressure, the damper pressure that can be measured at the shock absorber, the inside tire pressure, or the lateral deformation of the tire can be made the basis of a first condition variable that is correlated to the wheel load.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1a is a schematic side view of a four-wheel vehicle in a cornering maneuver.

FIG. 1b is a schematic rear view of a four-wheel vehicle also in a right-curve cornering maneuver.

FIG. 2 is a rear view of a vehicle in a left curve, corresponding to FIG. 1, to represent the physical condition variables related to driving for the description of a rollover condition.

FIG. 3 is a flow chart showing a preferred sequence of operation of the method according to the present invention and the corresponding device.

FIG. 4 is a schematic view of a preferred embodiment of the device of the present invention including a spring travel switch at one wheel and a characteristic curve that illustrates the point of time of switching of the spring travel switch.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

The embodiment described hereinbelow relates to modifications in an Electronic Stability Control system (ESP). In contrast to the two-dimensional model used in the state of the art, the present invention is based on a three-dimensional vehicle model wherein beside the transverse acceleration and the yaw, also pitching and rolling of the vehicle can be taken into account, with the result that the rollover tendency is also considered a value of the critical condition variable. A principal adaption of the vehicle model in consideration of the current static center of gravity will considerably support the control success. But also a reaction only to the side-tilt behavior of the vehicle will achieve already a major improvement of stability control.

The information required for determining the center of gravity and the vehicle side-tilt may preferably be obtained from the spring travel variations. The center of gravity can be determined by a measurement of the static compression travel with respect to axles without level control device or by a measurement of the pressure in the level control device. With a known spring rigidity C of the chassis support spring, the wheel normal force F_{n_xx} (FIG. 1a) can be determined from the known force $F_{n_xx_o}$ in the construction position with spring travel $S_{_xx_o}$ as follows:

$$F_{n_xx} = F_{n_xx_o} + K \cdot \Delta S$$

wherein ΔS represents the measured spring travel variation with a direction-responsive sign (positive for compression travel, negative for rebound travel).

A level control device compensates a compression travel, thus, ΔS becomes 0. The system pressure variation Δp necessary for this compensation is proportional to the change in force, and consequently

$$F_{n_xx} = F_{n_xx_o} + K \cdot \Delta p$$

applies, wherein k is the proportionality factor.

The roll angle is determined by way of the different spring travels on the left and the right vehicle side, and the pitch angle is determined accordingly by way of the spring travels on the front and the rear end.

The height of the center of gravity can be calculated from the side-tilt angle and the associated acceleration. The center of gravity in a longitudinal and transverse direction (I_v , I_h , y_s) is known from the static analysis. The height of the center of gravity may hence be derived from the correlation described hereinabove, with the transverse acceleration known, according to

$$h_s = \frac{1}{a_{transv.}} \cdot [g \cdot (y_s^{(b)} + y_s) - F_{n_r} \cdot b/m]$$

or, with the longitudinal acceleration known, according to

$$h_s = \frac{1}{a_{along.}} \cdot \left[\frac{F_n - VA}{m} \cdot l_v - \frac{F_n - HA}{m} \cdot l_r \right]$$

A center of gravity shifted from the vehicle center can be determined by the roll angle which is different responsive to the direction, with a transverse acceleration existing. Thus, e.g. for a vehicle with a level control device at the rear axle, four sensors for determining the spring travel are required and, in the event of an axle control on each individual wheel, one or two sensors for determining the pressure in the level control device of the rear axle are required.

The flow chart shown in FIG. 3 illustrates an exemplary sequence of operation of a modified ESP control, wherein an

end position switch is used corresponding to the mentioned simplified embodiment of this invention. The end position switch responds when a predetermined spring travel threshold value in one or more of the vehicle wheels is exceeded, that means, when a predetermined wheel load is exceeded with respect to the prevailing forces. Because the mode of operation of the end position switch hence depends on the direction of cornering, it is first of all necessary to find out which wheel (the wheel on the front axle is preferred) or which wheels are disposed on the curve-outward side of the vehicle. This is because only the curve-outward wheels are able to respond in the mentioned manner corresponding to the vehicle tilt during cornering. The selection of the curve-outward wheels can be effected by means of sensors related to driving dynamics, e.g., a steering angle sensor and/or a yaw rate sensor, which are already provided in an ESP system. In general, other sensors may also be used which permit determining the direction of a cornering maneuver.

In the event that one or more of the end position switches arranged on the curve-outward side of the vehicle respond, the variation of the ESP vehicle model which has already been described in detail hereinabove will be performed. It is initially necessary, however, to check or filter the switch signals received to the end that those signals are "filtered out" which induced one or more end position switches to respond due to road conditions, e.g. due to bumps, although there was no cornering maneuver and, consequently, there was no rollover hazard.

In consideration of further quantities related to driving dynamics, for example, the steering angle and the vehicle speed, a corresponding change of the center of gravity of the vehicle is calculated, and from the change center of gravity a changed or corrected limit transverse acceleration is calculated by way of the mathematical correlations mentioned hereinabove. By way of the corrected limit transverse acceleration, the ESP vehicle model will then be changed in a manner such that the vehicle temporarily becomes "understeering" and is, hence, better adapted to the changed center-of-gravity conditions.

In the normal case, the end position switch will no longer respond after termination of the cornering maneuver. The correspondingly filtered signal cancels the temporary change of the ESP vehicle model and, thus, reestablishes the condition of the ESP prior to the cornering maneuver.

In the embodiment referred to above, the ESP vehicle model is modified only temporarily, i.e., during cornering maneuvers. It may, however, also be provided that on command of the mentioned signals of the end position switch, there is a reaction to changed center-of-gravity conditions that is followed by permanent variations of the ESP vehicle model. The above-mentioned statements will apply accordingly in this case.

Due to the knowledge about a changed center of gravity, the vehicle model prevailing in an Electronic Stability Program (ESP), for example, the program of the applicant, can be adapted to the changed center-of-gravity conditions. It is initially necessary in the adaptation of the ESP control, to predefine a critical condition variable, for example, a limit transverse acceleration a_{limit} , in consideration of the current quantity and position of the vehicle center of gravity. This limit acceleration, in the capacity of a parameter variation, can be taken into consideration within the vehicle model in the form of a variation= basic variation+ $k \times f(a_{transv.,limit})$ (4) in the ESP control, and $a_{transv.}$ designates the current value of the transverse vehicle acceleration and $a_{transv.,limit}$ designates the corresponding critical limit value. Besides, 'F' is a proportional factor and $f(x)$ is a function with a predeter-

mined top and bottom limit, wherein it applies for the bottom limit that the change must not become less than zero. The above-mentioned functional correlation can preferably be implemented as a P-controller or PI-controller, with the addend on the right side of the equation (4) being configured to either act proportionally as a P-controller, i.e., according to the relation

$$k \times [abs(a_{transv.}) - abs(a_{transv.,limit})], \quad (5)$$

or proportional-integrally as a PI-controller, i.e., according to the following equation:

$$k \times [abs(a_{transv.}) - abs(a_{transv.,limit})] + k \times \text{integral}[abs(a_{transv.}) - abs(a_{transv.,limit})], \quad (6)$$

A change to 'understeering' can be defined by way of lateral rigidities.

In the vehicle model of ESP (single-track model)

$$m \cdot v \cdot (\psi - \beta) F_{lateral\ force\ front} + F_{lateral\ force\ rear}$$

$$\Theta \psi = F_{sv} \cdot l_v - F_{sh} \cdot l_h$$

$$\text{with } F_{sv} = C_{sv} \cdot \left(\delta + \beta - \frac{l_v \cdot \psi}{v} \right)$$

$$\text{with } F_{sh} = C_{sh} \cdot \left(\beta - \frac{l_h \cdot \psi}{v} \right)$$

the lateral tire rigidities C_{sv} , C_{sh} , front and rear, are comprised.

$C_{sv,sh}$ may now be understood as C_{sv} , C_s for the normal operation and C_{1sv} , C_{1sh} may be added as variable component.

$$C_{sv} = C_{0sv} + C_{1sv}$$

$$C_{sh} = C_{0sh} + C_{1sh}$$

These C_{1sv} , C_{1sh} may now be changed in dependence on the transverse acceleration, as described on page 10 (P, PI-controller), namely

$$C_{1sv} \text{ smaller}$$

$$C_{1sh} \text{ greater}$$

so that an ESP brake intervention on the curve-outward wheel stabilizes the vehicle.

When the curve limit acceleration or speed which is predetermined by the possibly changed center of gravity or center of gravity mass is exceeded, an intervention which prevents rollover, such as a brake intervention, may be carried out. Corresponding intervention strategies are e.g. disclosed in applications 198 21 593.2 and 198 16 430.0, to which reference is made herein in full extent. Thus, values may again drop below the critical condition variable, e.g. the transverse acceleration, by a defined understeering of the vehicle, especially by a corresponding re-tuning or readjustment of the vehicle model on which the ESP control actions are based, whereby rollover is prevented.

Only the roll behavior of the vehicle is taken into consideration in a simplified embodiment. When it is further assumed that the center of gravity is shifted only insignificantly from the longitudinal vehicle axis, one spring travel information per axle is sufficient. A critical situation may also be recognized by only one single spring travel when the height of the center of gravity remains virtually the same. In this arrangement, the vehicle model is conformed to the roll behavior, and the allowed transverse acceleration is reduced while the defined inherent steering behavior is maintained. This adaptation can be effected analogously, e.g., by means of information given by sensor signal(s), and digitally, e.g., by means of information given by switches. In view of the above, the most simple embodiment is a spring travel switch on one wheel which will be described hereinbelow by way of FIG. 4.

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One preferred embodiment of an end position switch relating to a simplified way of control is represented in FIG. 4a. This way of control can be realized advantageously as an ESP that is extended by an end position switch arranged at the shock absorber or at the spring of the wheel suspension. The following procedure is advisable corresponding to the time variation shown in FIG. 4b with respect to an end position switch configured as a spring travel switch. When the switch on at least one curve-outward wheel, which is preferably the outward front wheel, responds, the critical transverse acceleration $a_{transv.,limit}$ will be exceeded corresponding to the assumed course of characteristic curve, and a defined counteraction is initiated to prevent further buildup of the transverse acceleration or to reduce it.

Understeering of the vehicle is preferred, for example, by means of a corresponding 're-tuning' of the ESP vehicle model to become 'understeering', that means by a change in the travel trajectory, for example, by a way of control corresponding to equations (4) to (6).

The variation may be defined by way of the lateral rigidities of the tires, and 'understeering' means that the lateral rigidity at the front axle of the vehicle diminishes.

The intervention into the vehicle model causes braking and, thus, reduction of the cornering force at the curve-outward wheel. With respect to the physical correlation between cornering force and brake slip (longitudinal force), reference is made in the present context in full extent to German published patent application DE-A 196 32 943 'Method for operating an automotive vehicle with brake interventions that stabilize driving', by Daimler-Benz AG.

When the switch closes again, this variation will be cancelled with time, i.e., forgotten. This control action will be terminated herewith.

Monitoring of the switch is possible also by way of a plausibility analysis. It may be provided, for example, that during straight travel the switch is only allowed to close for a minimum duration of e.g. 200 msec before countermeasures to reduce the transverse acceleration are activated. Alternatively, it may be provided that the triggering of countermeasures, beside the switch position, is linked to further conditions, for example, a threshold value for the transverse acceleration, such as $>0.5-0.7$ g.

What is claimed is:

1. A method for detecting the rollover hazard of a vehicle that includes at least two axles and at least three wheels, wherein variations in the vehicle's center of gravity are determined in the operation of the vehicle during cornering, comprising the steps of:

- detecting first condition variables which respectively correspond to each respective wheel load during cornering on at least two wheels,
- comparing each detected first condition variable to reference values representative of the respective cornering maneuver,
- calculating a corresponding change in the center of gravity from the differences between the detected first condition variables and the reference values, and
- determining a correspondingly changed, critical second condition variable representative of the rollover hazard from the calculated center of gravity variation.

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2. The method as claimed in claim 1, further including the step of using at least one of a spring travel that can be measured at the wheel suspension, a spring pressure, a damper pressure that can be measured at a shock absorber, an inside tire pressure, and a lateral deformation of the vehicle tire as a basis for a first condition variable that corresponds to the wheel load.

3. The method as claimed in claim 1, further including the step of using at least one of a transverse acceleration which acts on the center of gravity of the vehicle, a roll angle of the vehicle, a pitch angle of the vehicle, a vehicle speed, a steering angle, and a yaw rate as a basis for the second condition variable representative of the rollover hazard.

4. A device for detecting the rollover hazard of a vehicle that includes at least two axles and at least three wheels, comprising:

- a means for sensing a driving condition in a cornering maneuver,
- a means for determining variations in the center of gravity of mass,
- a means to detect first condition variables which correspond to the respective wheel load on at least two wheels,
- a means to compare the detected first condition variables with reference values representative of the respective cornering maneuver,
- a means to calculate the change in the center of gravity from the differences between the detected first condition variables and the reference values, and
- a means to determine a second condition variable representative of the rollover hazard from the calculated center of gravity variation.

5. The device as claimed in claim 4, further including means for measuring at least one of the wheel suspension spring travel, the wheel suspension spring pressure, the shock absorber damper pressure, the pressure of the air inside of the tire, the lateral deformation of the tire, and using such measurement as a basis for the first condition variable that corresponds to the wheel load.

6. The device as claimed in claim 4, further including means for measuring at least one of the transverse acceleration which acts on the center of gravity of the vehicle, the roll angle of the vehicle, the pitch angle of the vehicle, the vehicle speed, the steering angle, the yaw rate, and using such measurement as a basis for the second condition variable representative of the rollover hazard.

7. The device as claimed in claim 4, further including an end position switch provided on a shock absorber or on a spring element of the wheel suspension, the switching point of the end position switch is linked by a time correlation to the second condition variable representative of the rollover hazard.

8. The device as claimed in claim 7, wherein the switching point of the end position switch is linked to further driving state conditions.

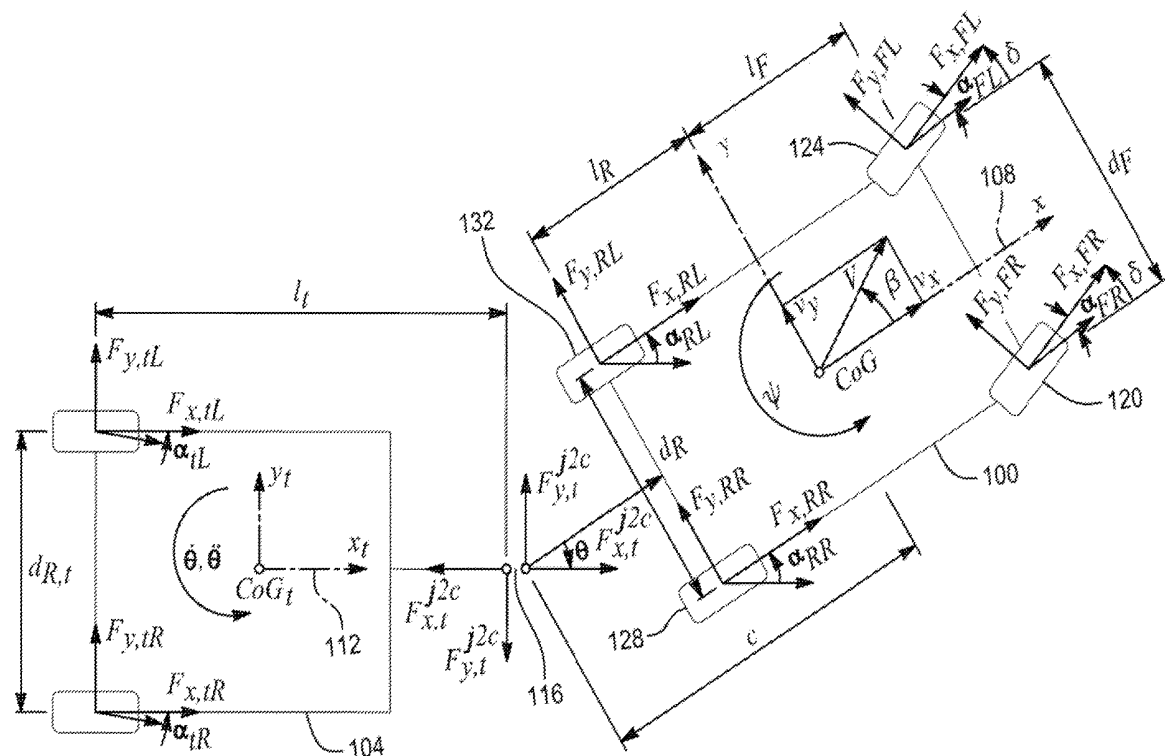
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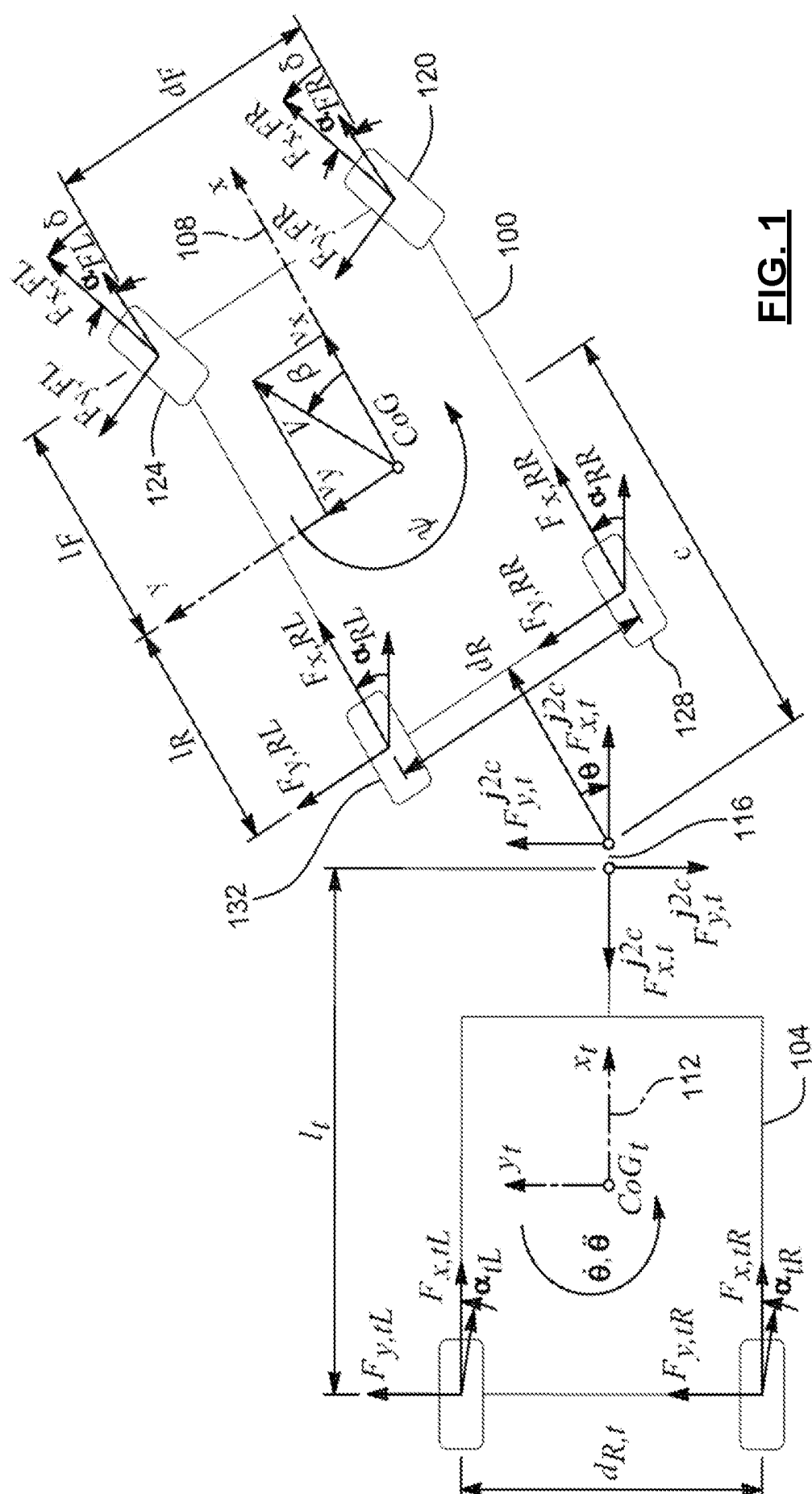


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(19) **United States**(12) **Patent Application Publication**
DHAENS et al.(10) **Pub. No.: US 2024/0010036 A1**(43) **Pub. Date: Jan. 11, 2024**(54) **VEHICLE CONTROL SYSTEMS AND
METHODS BASED ON TRAILER HITCH
JOINT FORCES**(52) **U.S. CL.**
CPC **B60D 1/30** (2013.01); **B60D 1/62** (2013.01)(71) Applicant: **DRiV Automotive Inc.**, Southfield, MI
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Sebastiaan VAN AALST, Hasselt (BE)(21) Appl. No.: **17/861,825**(22) Filed: **Jul. 11, 2022****Publication Classification**(51) **Int. Cl.**
B60D 1/30 (2006.01)
B60D 1/62 (2006.01)(57) **ABSTRACT**

A trailer sway control system for a vehicle includes: a front left active suspension actuator; a front right active suspension actuator; a rear left active suspension actuator; a rear right active suspension actuator; and an actuator control module configured to: based on (a) a first hitch force at a trailer hitch in a longitudinal direction, (b) a second hitch force at the trailer hitch in a lateral direction, and (c) a third hitch force at the trailer hitch in a vertical direction, determine target vertical forces for the front left active suspension actuator, the front right active suspension actuator, the rear left active suspension actuator, and the rear right active suspension actuator, respectively; and selectively adjust the front left active suspension actuator, the front right active suspension actuator, the rear left active suspension actuator, and the rear right active suspension actuator based on the target vertical forces, respectively.





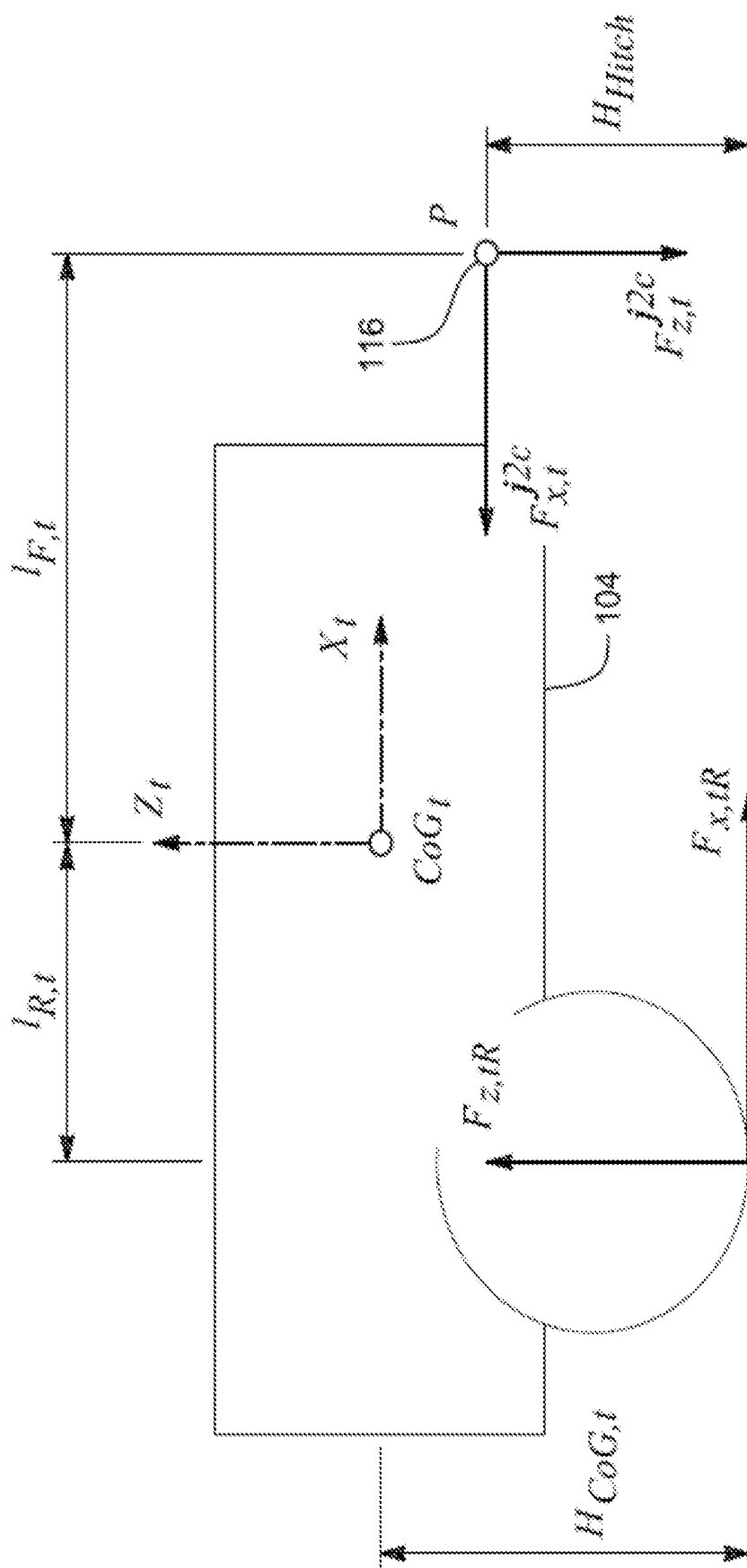


FIG. 2

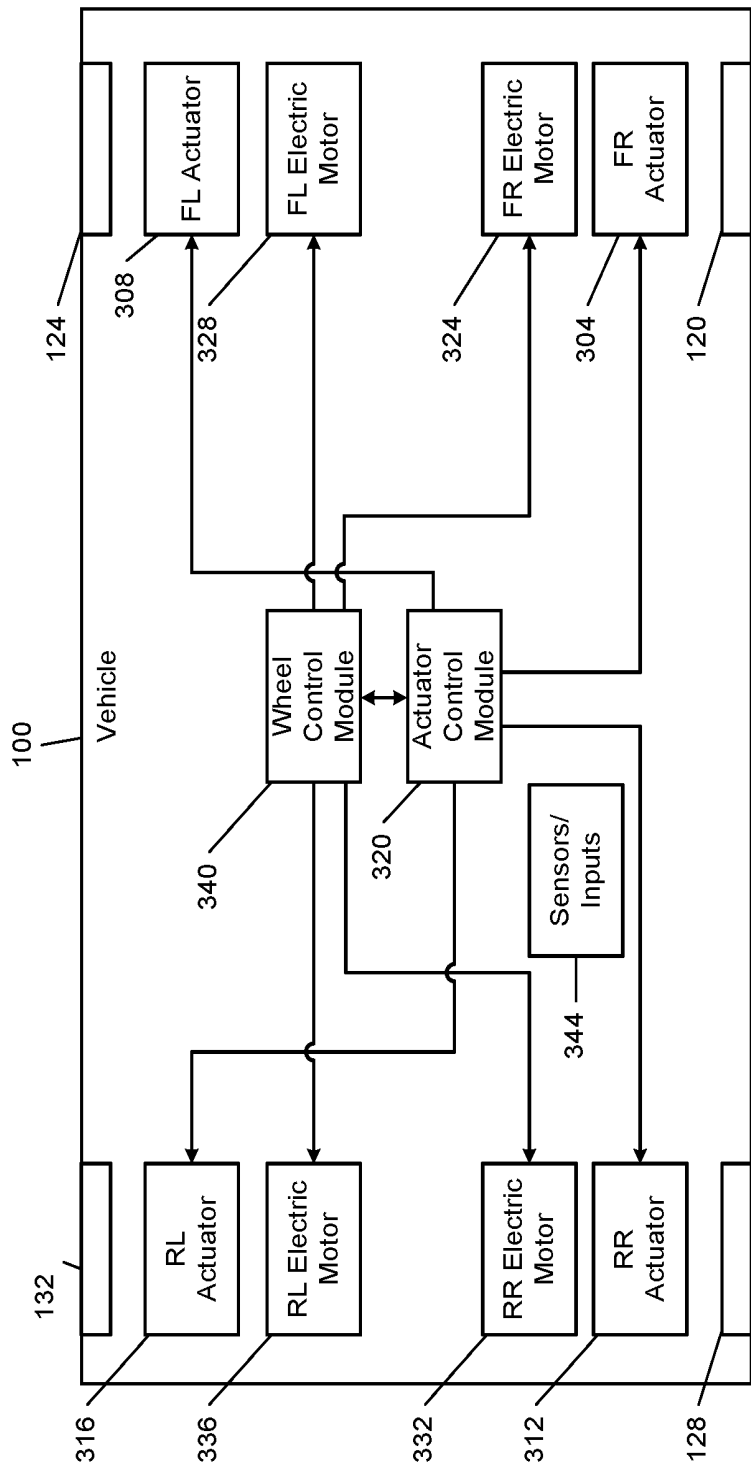


FIG. 3

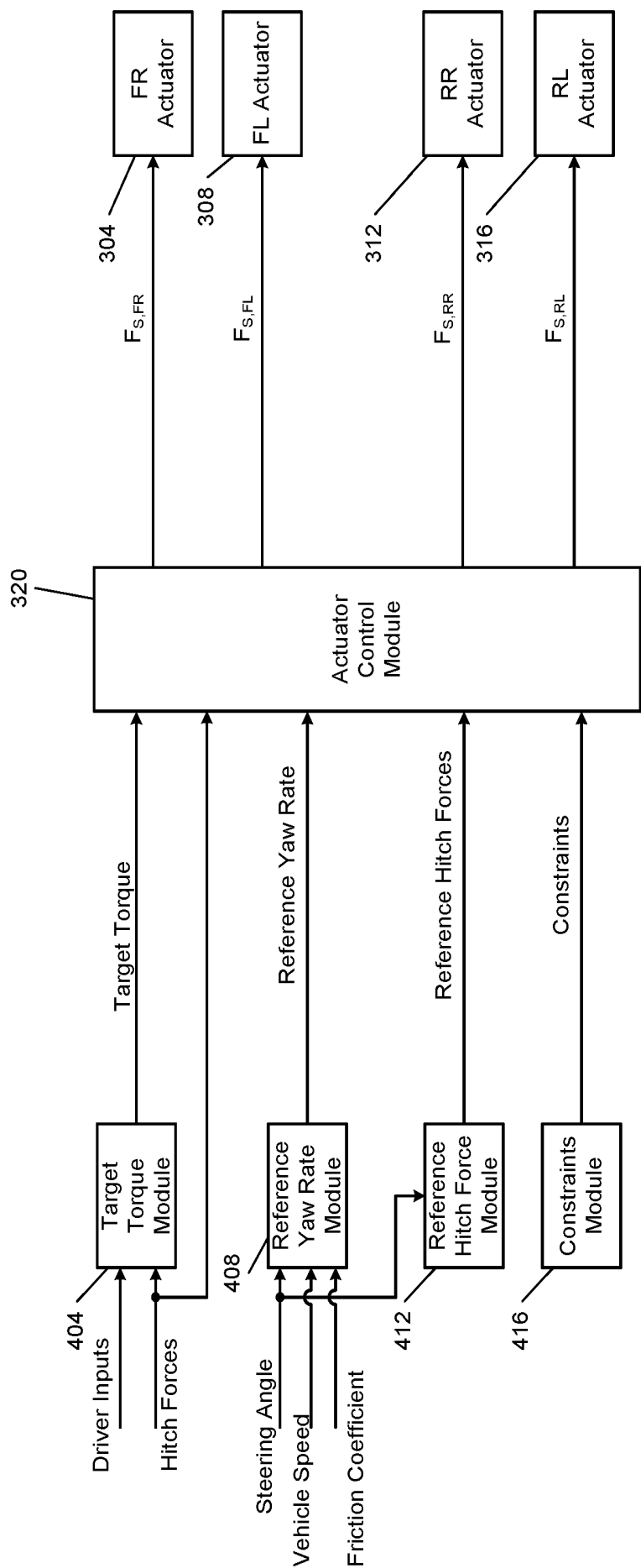


FIG. 4

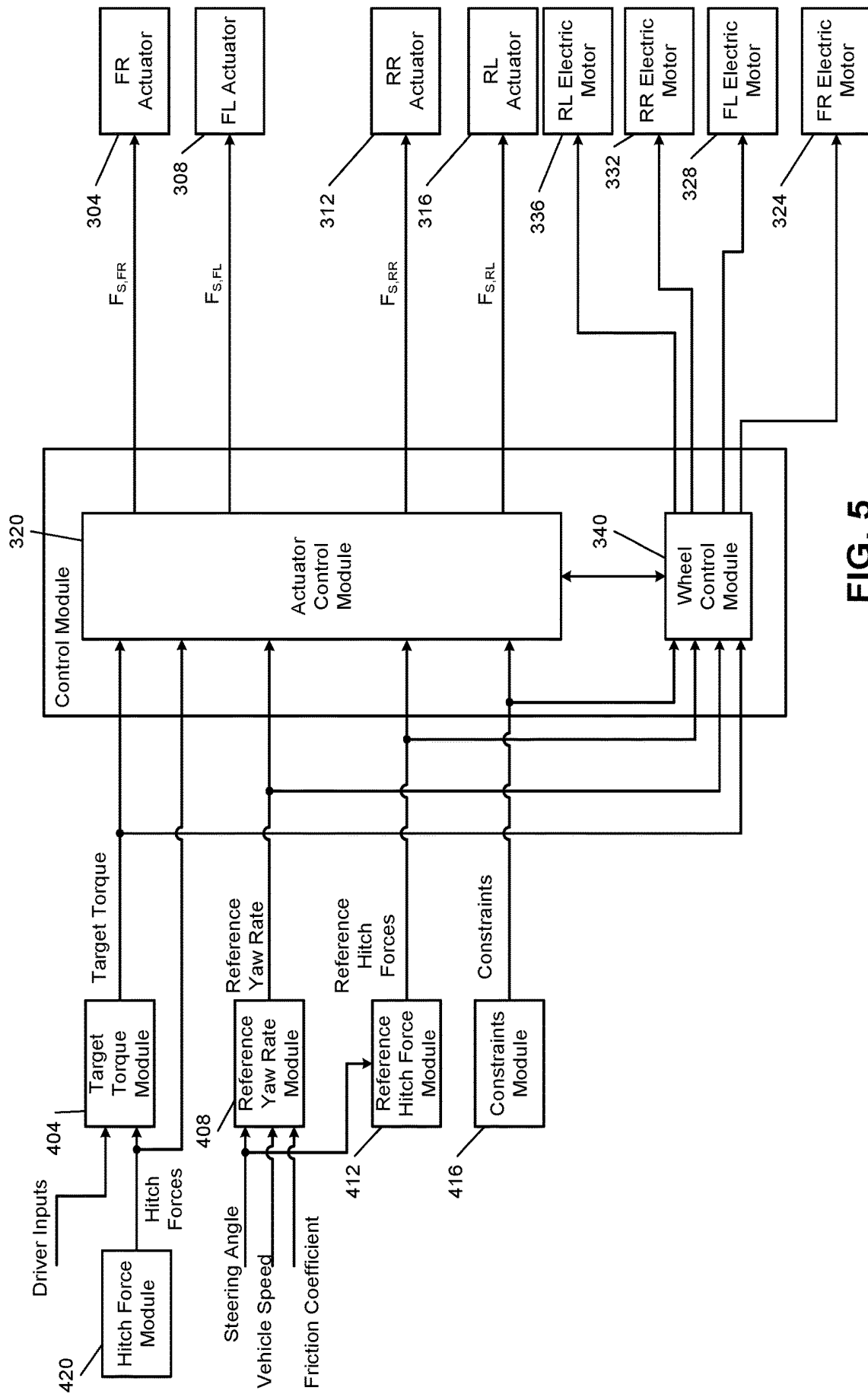


FIG. 5

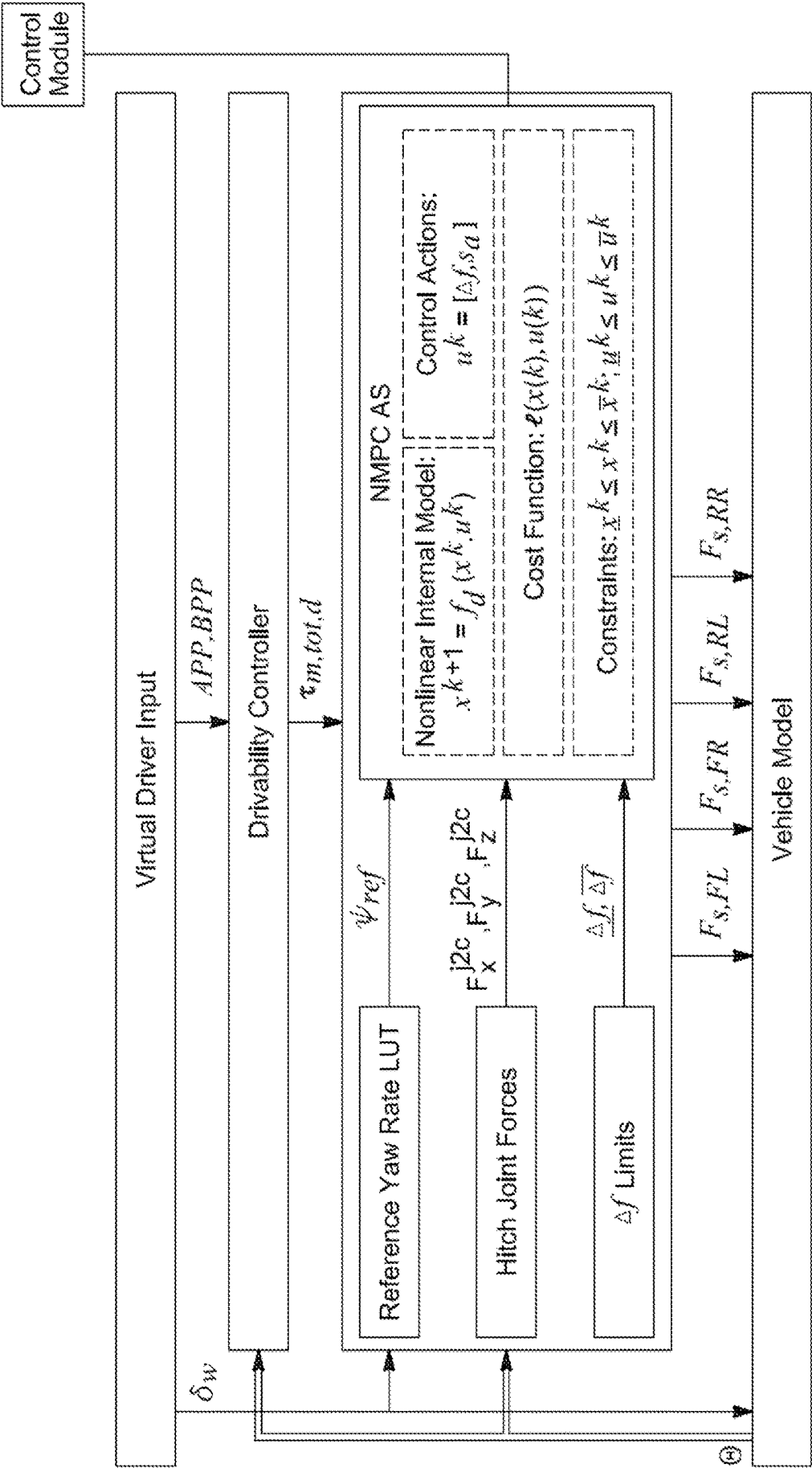


FIG. 6

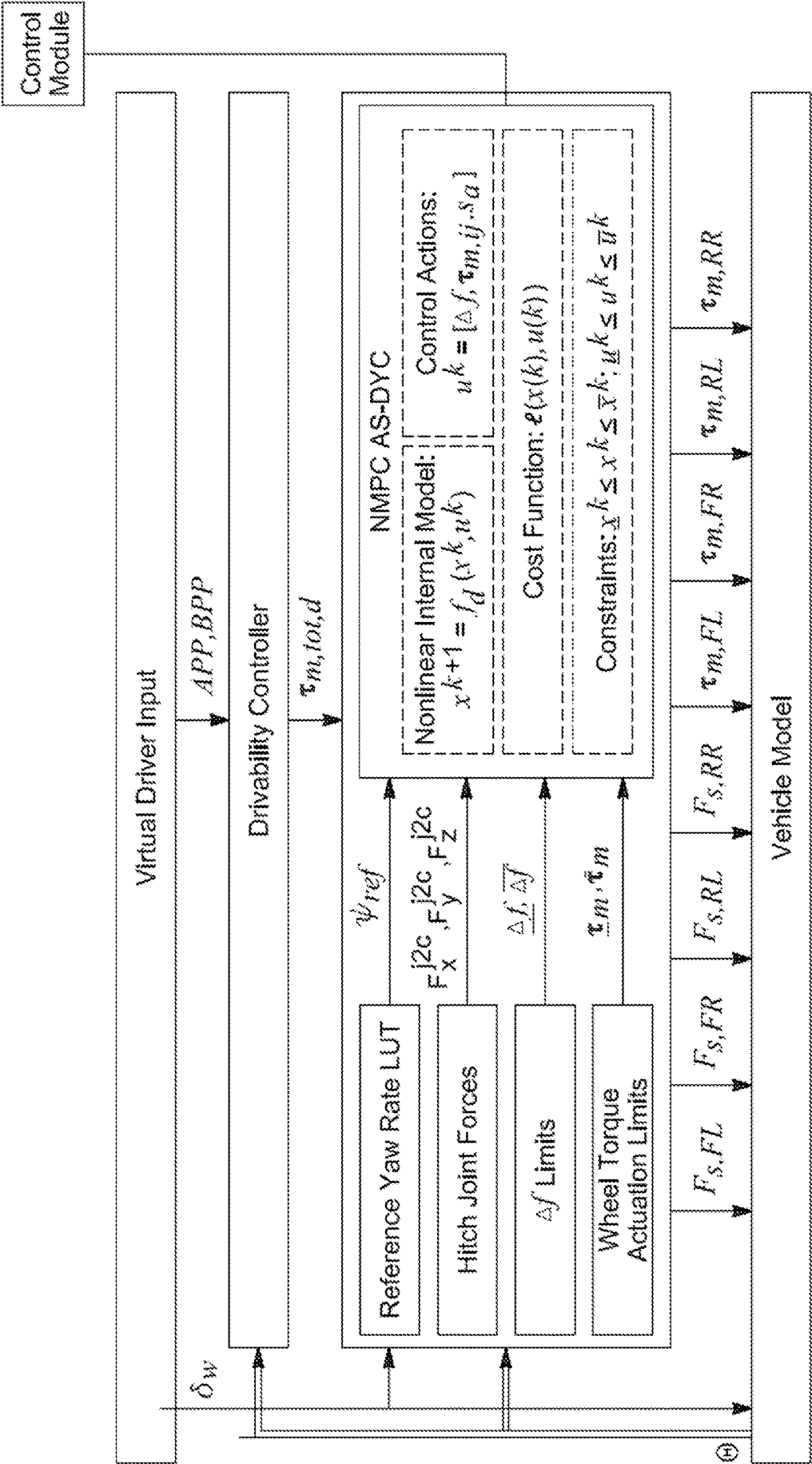


FIG. 7

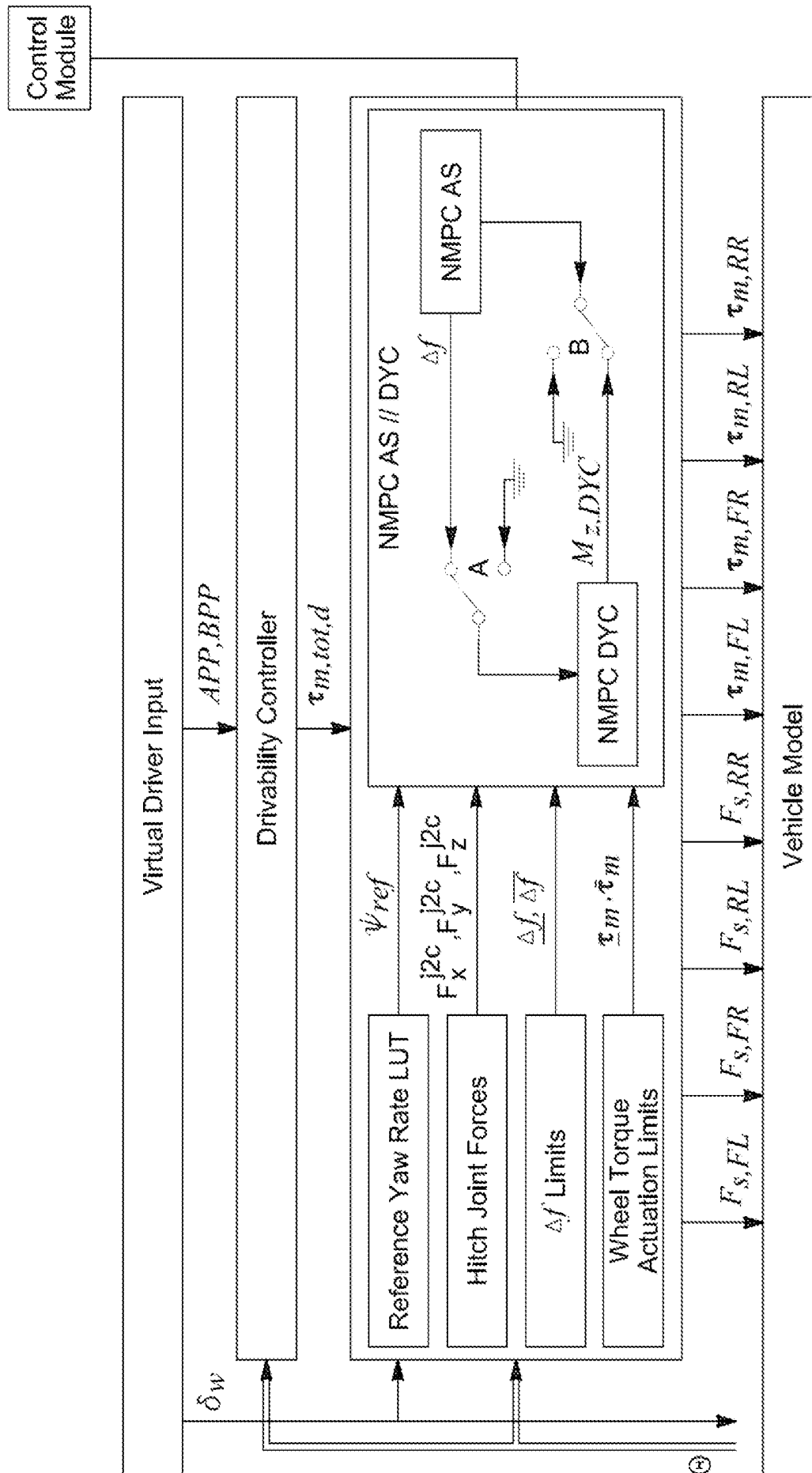


FIG. 8

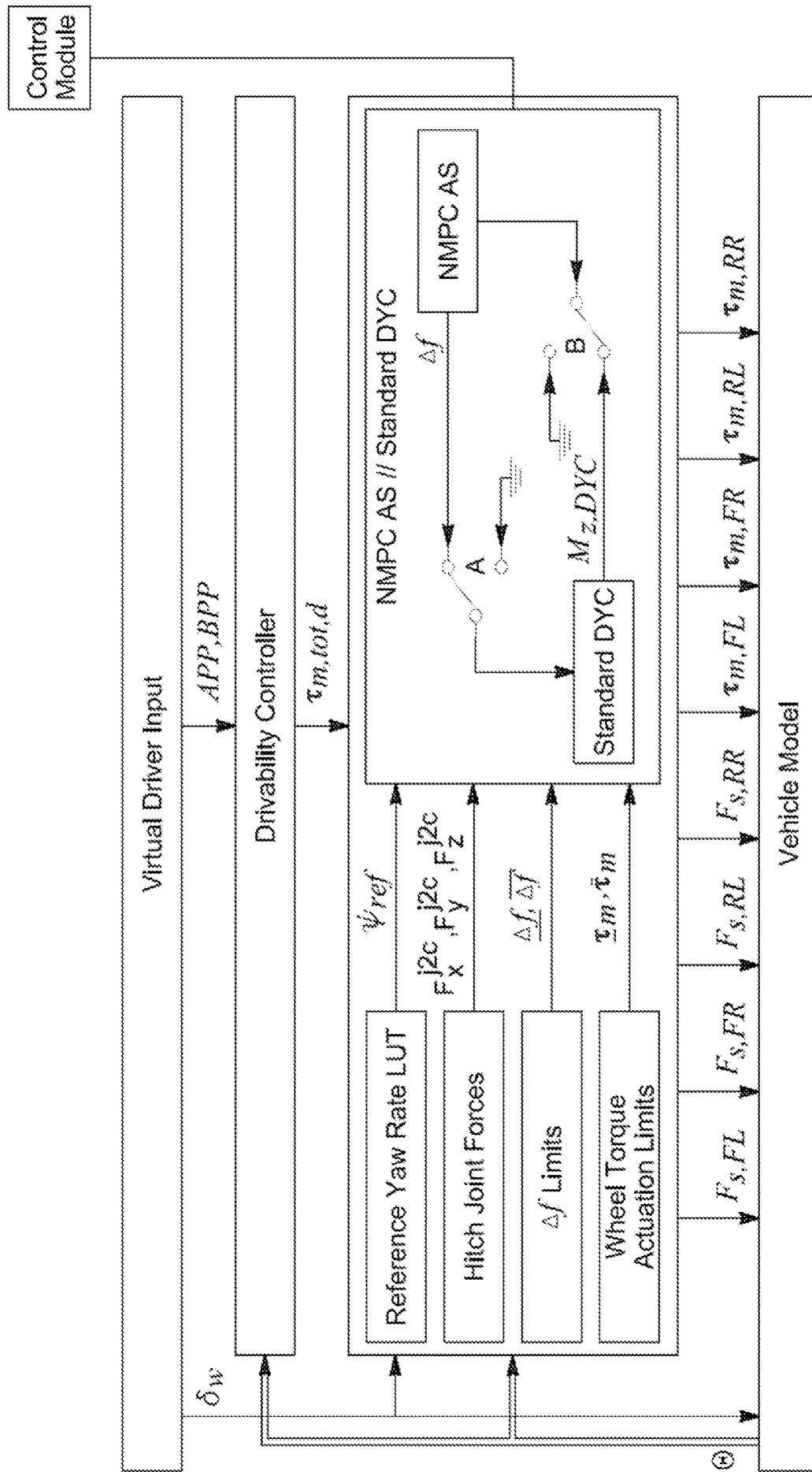
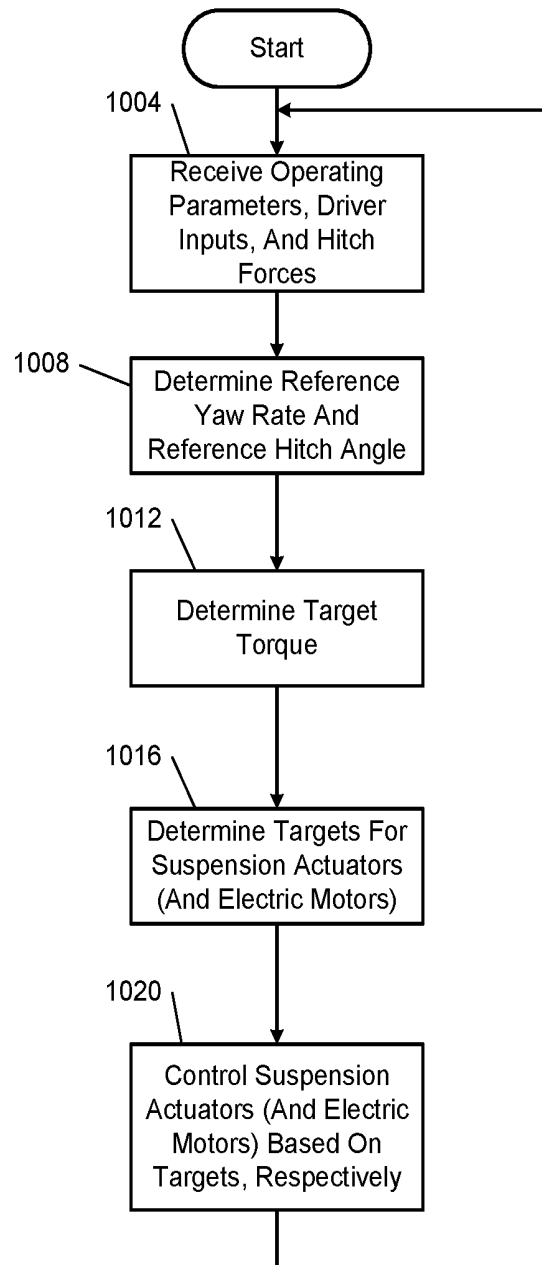


FIG. 9

**FIG. 10**

VEHICLE CONTROL SYSTEMS AND METHODS BASED ON TRAILER HITCH JOINT FORCES

FIELD

[0001] The present disclosure relates to control systems and methods of vehicles and more particularly to vehicle control based on forces at a hitch where a trailer is hitched to the vehicle.

BACKGROUND

[0002] The background description provided here is for the purpose of generally presenting the context of the disclosure. Work of the presently named inventors, to the extent it is described in this background section, as well as aspects of the description that may not otherwise qualify as prior art at the time of filing, are neither expressly nor impliedly admitted as prior art against the present disclosure.

[0003] Various different types of vehicles are available. For example, coupes, sedans, convertibles, trucks, station wagons, and utility vehicles are available. There are multiple different types of utility vehicles, such as sport utility vehicles (SUVs), crossover utility vehicles, etc.

[0004] Some types of vehicles are designed to tow trailers. For example, trucks and utility vehicles may be used to tow trailers. Various different types of trailers are available. Examples of trailers include cargo trailers, trailers for vehicles (e.g., land based vehicles, watercraft, etc.), travel trailers, and other types of trailers.

SUMMARY

[0005] In a feature, a trailer sway control system for a vehicle includes: a front left active suspension actuator associated with a front left wheel of the vehicle; a front right active suspension actuator associated with a front right wheel of the vehicle; a rear left active suspension actuator associated with a rear left wheel of the vehicle; a rear right active suspension actuator associated with a rear right wheel of the vehicle; and an actuator control module configured to: based on (a) a first hitch force at a trailer hitch of the vehicle in a longitudinal direction, (b) a second hitch force at the trailer hitch of the vehicle in a lateral direction, and (c) a third hitch force at the trailer hitch of the vehicle in a vertical direction, determine target vertical forces for the front left active suspension actuator, the front right active suspension actuator, the rear left active suspension actuator, and the rear right active suspension actuator, respectively; and selectively adjust the front left active suspension actuator, the front right active suspension actuator, the rear left active suspension actuator, and the rear right active suspension actuator based on the target vertical forces, respectively.

[0006] In further features, the actuator control module is configured to determine the target vertical forces based on minimizing (a) a first difference between the first hitch force and a first reference hitch force, (b) a second difference between the second hitch force and a second reference hitch force, and (c) a third difference between the third hitch force and a third reference hitch force.

[0007] In further features, a reference hitch forces module is configured to set the first reference hitch force, the second reference hitch force, and the third reference hitch force based on a steering angle of the vehicle.

[0008] In further features, the actuator control module is configured to determine the target vertical forces further based on a yaw rate of the vehicle.

[0009] In further features, the actuator control module is configured to determine the target vertical forces based on minimizing a difference between the yaw rate and a reference yaw rate.

[0010] In further features, a reference yaw rate module is configured to set the reference yaw rate based on a steering angle of the vehicle.

[0011] In further features, the reference yaw rate module is configured to set the reference yaw rate further based on a vehicle speed.

[0012] In further features: a front left electric motor is configured to output torque to the front left wheel of the vehicle; a front right electric motor is configured to output torque to the front right wheel of the vehicle; a rear left electric motor is configured to output torque to the rear left wheel of the vehicle; a rear right electric motor is configured to output torque to the rear right wheel of the vehicle; and a wheel control module is configured to: based on the first hitch force, the second hitch force, and the third hitch force, determine target torque outputs of the front left electric motor, the front right electric motor, the rear left electric motor, and the rear right electric motor, respectively; and selectively adjust the front left electric motor, the front right electric motor, the rear left electric motor, and the rear right electric motor based on the target torque outputs, respectively.

[0013] In further features, the wheel control module is configured to determine the target torque outputs further based on the target vertical forces.

[0014] In further features, the actuator control module is configured to determine the target vertical forces further based on the target torque outputs.

[0015] In further features: a front electric motor is configured to output torque to front wheels of the vehicle; a rear electric motor is configured to output torque to rear wheels of the vehicle; and a wheel control module is configured to: based on the hitch angle, determine target torque outputs of the front and rear electric motors, respectively; and selectively adjust the front and rear electric motors based on the target torque outputs, respectively.

[0016] In further features: an electric motor is configured to output torque two or more wheels of the vehicle; and a wheel control module is configured to: based on the hitch angle, determine a target torque of the electric motor; and selectively adjust the electric motor based on the target torque.

[0017] In a feature, a trailer sway control method for a vehicle includes: based on (a) a first hitch force at a trailer hitch of the vehicle in a longitudinal direction, (b) a second hitch force at the trailer hitch of the vehicle in a lateral direction, and (c) a third hitch force at the trailer hitch of the vehicle in a vertical direction, determining target vertical forces for a front left active suspension actuator, a front right active suspension actuator, a rear left active suspension actuator, and a rear right active suspension actuator, respectively, the front right active suspension actuator associated with a front right wheel of the vehicle, the front left active suspension actuator associated with a front left wheel of the vehicle, the rear left active suspension actuator associated with a rear left wheel of the vehicle, and the rear right active suspension actuator associated with a rear right wheel of the

vehicle; and selectively adjusting the front left active suspension actuator, the front right active suspension actuator, the rear left active suspension actuator, and the rear right active suspension actuator based on the target vertical forces, respectively.

[0018] In further features, determining the target vertical forces includes determining the target vertical forces based on minimizing (a) a first difference between the first hitch force and a first reference hitch force, (b) a second difference between the second hitch force and a second reference hitch force, and (c) a third difference between the third hitch force and a third reference hitch force.

[0019] In further features, the method further includes setting the first reference hitch force, the second reference hitch force, and the third reference hitch force based on a steering angle of the vehicle.

[0020] In further features, determining the target vertical forces includes determining the target vertical forces further based on a yaw rate of the vehicle.

[0021] In further features, determining the target vertical forces includes determining the target vertical forces based on minimizing a difference between the yaw rate and a reference yaw rate.

[0022] In further features, the method further includes setting the reference yaw rate based on a steering angle of the vehicle.

[0023] In further features, setting the reference yaw rate includes setting the reference yaw rate further based on a vehicle speed.

[0024] In further features, the method further includes: based on the hitch angle, determining a target torque of an electric motor configured to output torque two or more wheels of the vehicle; and selectively adjust the electric motor based on the target torque.

[0025] Further areas of applicability of the present disclosure will become apparent from the detailed description, the claims and the drawings. The detailed description and specific examples are intended for purposes of illustration only and are not intended to limit the scope of the disclosure.

BRIEF DESCRIPTION OF THE DRAWINGS

[0026] The present disclosure will become more fully understood from the detailed description and the accompanying drawings, wherein:

[0027] FIG. 1 is top view of a model including a vehicle and a trailer;

[0028] FIG. 2 is a side view of an example model of the trailer including various parameters;

[0029] FIG. 3 is a functional block diagram of example components of the vehicle;

[0030] FIGS. 4-9 are functional block diagrams of example implementations of a trailer sway control system; and

[0031] FIG. 10 is a flowchart depicting an example method of controlling one or more vehicle actuators based on hitch forces of the trailer.

[0032] In the drawings, reference numbers may be reused to identify similar and/or identical elements.

DETAILED DESCRIPTION

[0033] Vehicles may be used to tow trailers. A vehicle may include one or more trailer sway mitigation modules that

selectively adjust one or more operating parameters of the vehicle to decrease sway of a trailer when swaying occurs.

[0034] The present application involves control of at least one of suspension actuators of the vehicle and wheel torques of the vehicle based on a one or more hitch forces at the hitch where the trailer is hitched to the vehicle. The hitch forces include a force in a longitudinal direction of the trailer, a lateral direction of the trailer, and a vertical direction. The control enhances cornering stability of a vehicle towing a trailer and minimizes sway of the trailer.

[0035] FIG. 1 is top view of a model including a vehicle 100 and a trailer 104. CoG represents the center of gravity of a body. Longitudinal axes of the vehicle 100 and the trailer 104 are illustrated by 108 and 112, respectively. The trailer 104 is hitched to the vehicle 100 at trailer hitch 116. A hitch angle θ represents the angle between the longitudinal axes of the vehicle 100 and the trailer 104 (between 108 and 112). A hitch force in a longitudinal direction of the trailer 104 is expressed as $F_{x,t}$. A hitch force in a lateral direction of the trailer 104 is expressed as $F_{y,t}$.

[0036] The vehicle 100 includes front right and left wheels 120 and 124, respectively. The vehicle 100 also includes rear right and left wheels 128 and 132, respectively. The front wheels 120 and 124 are turnable left and right to enable vehicle turning. In various vehicles, the rear wheels 128 and 132 may also be turnable left and right or may be fixed parallel to the longitudinal axis 108 of the vehicle 100. The trailer 104 includes two or more wheels, such as illustrated. Parameters of the trailer 104 are denoted with a subscript t. FIG. 2 is a side view of an example model of the trailer 104 including various parameters. A hitch force in a vertical direction of the trailer 104 is expressed as $F_{z,t}$.

[0037] FIG. 3 is a functional block diagram of example components of the vehicle 100. As discussed above, the vehicle 100 includes the front right and left wheels 120 and 124 and the rear right and left wheels 128 and 132. The vehicle 100 may include additional wheels, such as in the example of dual rear wheel vehicles.

[0038] A front right (FR) suspension actuator 304 is an active suspension actuator and damps vertical motion near a FR corner of the vehicle 100, such as at the front right wheel 120. A front left (FL) suspension actuator 308 is an active suspension actuator and damps vertical motion near a FL corner of the vehicle 100, such as at the front left wheel 124. A rear right (RR) suspension actuator 312 is an active suspension actuator and damps vertical motion near a RR corner of the vehicle 100, such as at the rear right wheel 128. A rear left (RL) suspension actuator 316 is an active suspension actuator and damps vertical motion near a RL corner of the vehicle 100, such as at the rear left wheel 132. An actuator control module 320 controls damping forces ($F_{s,FR}$, $F_{s,FL}$, $F_{s,RR}$, $F_{s,RL}$) of the suspension actuators 304, 308, 312, and 316, respectively, based on at least one of the hitch forces ($F_{x,t}$, $F_{y,t}$, $F_{z,t}$), as discussed further below. The actuator control module 320 may control, for example, a pressure of a hydraulic fluid within one or more portions of a suspension actuator, control electric power (e.g., voltage, current) applied to a suspension actuator, or control the damping force of the suspension actuator in another suitable manner.

[0039] The vehicle 100 includes one or more torque producing devices that output torque to two or more of the wheels 120-132. For example, the vehicle 100 may include a FR electric motor 324 that outputs torque to the FR wheel

120 and a FL electric motor 328 that outputs torque to the FL wheel 124. The vehicle 100 may also include a RR electric motor 332 that outputs torque to the RR wheel 128 and a RL electric motor 336 that outputs torque to the RL wheel 132. Other motor configurations are also possible, such as one electric motor that outputs torque to the front wheels 120 and 124 and/or one electric motor that outputs torque to the rear wheels 128 and 132. Electric motors can also apply negative torque to slow/brake the vehicle 100. The vehicle 100 also includes friction brakes at each wheel that can be applied to slow that wheel. A wheel control module 340 controls the torque at each wheel. For example, the wheel control module 340 may control the torque at a wheel via controlling an electric motor and/or controlling application of the friction brake at that wheel. Additionally or alternatively to the actuator control module 320 controlling the damping forces based on the hitch force(s), the wheel control module 340 may control the torques ($\tau_{m,FR}$, $\tau_{m,FL}$, $\tau_{m,RR}$, $\tau_{m,RL}$) at the wheels, respectively, based on the hitch force(s).

[0040] The vehicle 100 includes various sensors and other input devices collectively illustrated by 344. Examples of sensors include but are not limited to an accelerator pedal position (APP) sensor, a brake pedal position (BPP) sensor, and other types of sensors. Examples of other types of input devices include, for example, cameras, radar sensors, ultrasonic sensors, and other types of sensors. As an example of a camera, a rear backup camera may be included. Images from the rear backup camera may be used to determine the hitch angle of the trailer 104. In various implementations, the hitch angle may be determined, measured, or estimated in another suitable manner.

[0041] FIG. 4 is a functional block diagram of an example implementation of a trailer sway control system. A target torque module 404 determines a target torque ($\tau_{m,tot,d}$) based on one or more driver inputs and the hitch forces of the trailer 104. The target torque module 404 may determine the target wheel torque, for example, using one or more equations and/or lookup tables that relate the driver inputs and the hitch forces to target torques. The driver inputs may include, for example, the APP measured using the APP sensor and the BPP measured using the BPP sensor.

[0042] A reference yaw rate module 408 determines a reference (target) yaw rate of the vehicle based on one or more inputs, such as steering angle δ , vehicle speed V , and tire-road coefficient of friction μ . The reference yaw rate module 408 may determine the reference yaw rate using one or more equations and/or lookup tables that relate steering angles, vehicle speeds, and coefficients to reference yaw rates. The steering angle may be determined, for example, based on measurements from a steering wheel angle (SWA) sensor, such as using one or more lookup tables or equations that relate SWAs to steering angles. The vehicle speed may be measured or estimated using one or more sensors. For example, the vehicle speed may be set based on one or more wheel speeds measured using one or more wheel speed sensors, respectively. The tire-road coefficient of friction may be determined, for example, based on wheel slip. The reference yaw rate module 408 may filter the reference yaw rate before use, such as using a first order transfer function or another suitable filter. In various implementations, the vehicle speed and tire-road coefficient of friction may not be used, and the reference yaw rate module 408 may determine

the reference yaw rate based on the steering angle. Further example detail is provided below.

[0043] A reference hitch forces module 412 determines reference (target) hitch forces. The reference hitch forces module 412 may determine the reference hitch forces based on the steering angle and one or more other parameters. The reference hitch forces module 412 may determine the reference hitch forces using one or more equations and/or lookup tables that relate the steering angle and the other parameter(s) to reference hitch forces. Further example detail is provided below.

[0044] A constraints module 416 determines one or more constraints (or limits). The constraints may be fixed predetermined values or may be variable. In the example of variables, the constraints module 416 may set one or more of the constraints based on one or more operating parameters of the vehicle 100 and/or the trailer 104.

[0045] The actuator control module 320 determines the damping forces for the suspension actuators 304-316, respectively, based on the target torque, (present) hitch forces, the reference yaw rate, the reference hitch forces, and the constraints. The actuator control module 320 may determine the damping forces further based on a present yaw rate of the vehicle 100. The present yaw rate of the vehicle 100 may be, for example, measured using a yaw rate sensor (of the sensors 344). The actuator control module 320 may determine the damping forces using one or more equations and/or lookup tables that relate target torques, reference yaw rates, yaw rates, reference hitch forces, hitch forces, and constraints to damping forces. The actuator control module 320 may determine the damping forces to minimize the difference (error) between the yaw rate and the reference yaw rate and to minimize the difference (error) between the hitch forces and the reference hitch forces. In other words, the actuator control module 320 may determine the damping forces to adjust the yaw rate toward or to the reference yaw rate and adjust the hitch forces toward or to the reference hitch forces.

[0046] The actuator control module 320 controls the suspension actuators 304-316 to achieve the damping forces, respectively. The actuator control module 320 may control, for example, hydraulic pressure and/or power applied to the suspension actuators 304-416.

[0047] In various implementations, the actuator control module 320 may initiate controlling the damping forces based on the hitch forces when the yaw rate is different than the target yaw rate by at least a predetermined amount. In various implementations, the actuator control module 320 may continuously control the damping forces based on the hitch forces, even when the yaw rate is within the predetermined amount of the target yaw rate.

[0048] In various implementations, the actuator control module 320 may set the damping forces using model predictive control (MPC), such as a nonlinear model predictive control (NMPC).

[0049] FIG. 5 is a functional block diagram of an example control system. As illustrated in FIG. 5, the wheel control module 340 may determine the torques ($\tau_{m,FR}$, $\tau_{m,FL}$, $\tau_{m,RR}$, $\tau_{m,RL}$) at the wheels, respectively, based on the target torque, (present) hitch forces, the reference yaw rate, the reference hitch forces, and the constraints. The wheel control module 340 may determine the torques further based on the present yaw rate of the vehicle 100. The wheel control module 340 may determine the torques using one or more equations

and/or lookup tables that relate target torques, reference yaw rates, yaw rates, reference hitch forces, hitch forces, and constraints to wheel torques. The wheel control module **340** may determine the torques to minimize the difference (error) between the yaw rate and the reference yaw rate and to minimize the difference (error) between the hitch forces and the reference hitch forces. In other words, the wheel control module **340** may determine the torques to adjust the yaw rate toward or to the reference yaw rate and adjust the hitch forces toward or to the reference hitch forces.

[0050] The wheel control module **340** controls the motors **324-336** to achieve the torques, respectively. The wheel control module **340** may for example control the power applied to the electric motors **324-336** based on the torques, respectively. The wheel control module **340** may additionally control application of the friction brakes of the wheels based on the torques, respectively. While the example of one motor per wheel is provided, the present application is also applicable to one electric motor for the front axle and one electric motor for the rear axle and to only one electric motor for the front or rear axle. In various implementations, an internal combustion engine may be included, and the wheel control module **340** may additionally or alternatively control torque output of the internal combustion engine based on the torques. In various implementations, the wheel control module **340** may additionally or alternatively control one or more differentials based on the torques.

[0051] In various implementations, the wheel control module **340** may initiate controlling the torques based on the hitch forces when the yaw rate is different than the target yaw rate by at least a predetermined amount. In various implementations, the wheel control module **340** may continuously control the torques based on the hitch forces, even when the yaw rate is within the predetermined amount of the target yaw rate.

[0052] In various implementations, the actuator control module **320** and the wheel control module **340** may share parameters. For example, the actuator control module **320** may set the damping forces at the wheels based on the torques, respectively, at the wheels. As another example, the wheel control module **340** may set the torques at the wheels based on the damping forces, respectively, at the wheels.

[0053] In various implementations, the wheel control module **340** may set the torques using MPC control, such as NMPC control. An example implementation of NMPC control of the wheel torques is discussed further below. An example implementation of NMPC control of the damping forces and the wheel torques is also discussed further below. In various implementations, the wheel control module **340** and the actuator control module **320** may be implemented together within a control module, such as illustrated in FIG. 5.

[0054] The driver inputs may be assumed to be constant along the NMPC prediction horizon. The driver inputs may include the steering angle θ_{sw} , the accelerator pedal position APP, and the brake pedal position or force BPP. Other inputs that may be fixed along the prediction horizon are: i) the towing vehicle **100** longitudinal and lateral accelerations, a_x and a_y that may be used to determine the load transfers (e.g., by the actuator control module); ii) a baseline front-to-total anti-roll moment distribution for; control parameters, e.g., constraints, weights, flags, etc.; iv) the trailer **104** longitudinal and lateral accelerations (which may be estimated) $\alpha_{x,t}$ and $\alpha_{y,t}$. Control of the vehicle yaw rate and hitch force

dynamics adopts the reference variables that define the steady-state cornering condition of the vehicle, for the given driver inputs.

[0055] The steady-state reference yaw rate of the vehicle-trailer system $\dot{\psi}_{ref}$ is configured to match a steady-state cornering response of the passive rigid vehicle. $\dot{\psi}_{ref}$ may be stored as nonlinear map (e.g., lookup table), and the reference yaw rate may be determined as function of steering angle δ , vehicle speed V , and tire-road friction coefficient μ . The output of the map (lookup table) may be filtered by the reference yaw rate module **408**, such as using a first order transfer function. The dynamic instability of the vehicle-trailer system may be detected by the control module when the actual yaw rate differs from the reference yaw rate by at least a predetermined amount. The map may be based on, for example, a nonlinear quasi-static or dynamic vehicle models and minimize additional key performance criteria such as tire slip power losses or total power losses. In various implementations, the reference yaw rate may be determined by the reference yaw rate module **408** based only on the steering angle.

[0056] The reference yaw rate may target the control of the vehicle cornering behavior, but may also account for the trailer presence. In various implementations example, the reference yaw rate module **408** may apply a band pass filter that outputs signals within a predetermined frequency band that are indicative of actual trailer sway. The predetermined frequency band may be, for example, 0.5 Hertz (Hz) to 1.5 Hz or another suitable frequency band. In various implementations, the control module may blend the yaw rate and hitch forces reference contributions when the trailer dynamics when one or more differences between ones of the hitch forces and respective reference hitch forces is/are greater than a predetermined value. The hitch forces may be measured using a sensor or estimated. In various implementations, the hitch forces may be determined by a hitch forces module **420** based on one or more measured operating parameters of the vehicle **100** and parameters of the trailer **104**, for example, using one or more lookup tables and/or equations.

[0057] Vehicle roll stability may be preserved by the control module during steering maneuvers by controlling the actuators **304-316** and/or the motors **316-336** to adjust a roll angle toward or to a target roll angle of the sprung masses. The target roll angle may be a predetermined value, such as 1 degree for a lateral acceleration of 5 m/s² or another suitable value. In various implementations, the control module may control the actuators **304-316** and/or the motors **316-336** to adjust a rollover prevention index RI which depends on roll dynamics and lateral acceleration to less than a predetermined value, such as 0.8 or another suitable value.

[0058] Despite being an additional reference state for the vehicle-semitrailer stability, roll angle may not be considered among the target variables in various implementations. Details of various parameters are illustrated in FIGS. 1 and 2.

[0059] FIGS. 6-9 include functional block diagrams of example implementations of the system of FIG. 4. FIG. 6 includes an example active suspension (AS) configuration. FIGS. 7-9 include example AS plus dynamic yaw control (DYC) configurations. The control inputs for the AS configuration (FIG. 6) are a front-to-total anti-roll moment distribution variation, Δf , of the active suspension system,

with respect to a predetermined nominal value. In the example of FIG. 7, in addition to the inputs of FIG. 6, the four torques on each wheel are used as control inputs. The control inputs in this example are Δf , $\tau_{m,FL}$, $\tau_{m,FR}$, $\tau_{m,RL}$, and $\tau_{m,RR}$. In the example of FIG. 8, the standalone AS and DYC for the articulated vehicle run in parallel sharing or not sharing the output control actions with each other. The control inputs for this example are Δf , $\tau_{m,FL}$, $\tau_{m,FR}$, $\tau_{m,RL}$, and $\tau_{m,RR}$. In the example of FIG. 9, the standalone AS for articulated vehicle and the standalone DYC for the isolated vehicle run in parallel sharing or not sharing the output control actions with each other. The control inputs for this example are Δf , $\tau_{m,FL}$, $\tau_{m,FR}$, $\tau_{m,RL}$, and $\tau_{m,RR}$.

[0060] In the formulations herein, the subscript i, with i=F,R indicates the axle, whereas the subscript j, with j=L,R, indicates the vehicle side. The notation ‘(·)’ indicates a time derivative, while ‘(·)’ indicates a second order time derivative. The subscript ‘t’ indicates forces applied to the vehicle according to the directions of the trailer reference frame. Other variables are the sideslip angle at the CoG, β ; the vehicle speed, V; and the hitch angle θ , namely the angle between the longitudinal axes of the trailer and the towing vehicle.

[0061] The proposed controller internal model, namely the model used by NMPC model of the control module for predicting the system response along the prediction horizon may be based on the following assumptions: motion is planar, and thus the road bank and inclination angles are negligible; heave, roll and pitch dynamics are disregarded; powertrain and braking torque on each corner is known in a first approximation; lateral and longitudinal displacements of the center of gravity caused by the roll and pitch motions are neglected;

[0062] The vehicle model used to determine the target forces for the suspension actuators may have 7 degrees of freedom, such as described by equations (1)-(4). FIGS. 1 and 2 illustrate states and parameters in accordance with their positive sign conventions.

[0063] Longitudinal force balance may be described by the equation:

$$m[\dot{v}_x - \dot{\psi}v_y] = \sum_{j=L,R} [F_{x,Fj} \cos(\delta) - F_{y,Fj} \sin(\delta)] + \sum_{j=L,R} F_{x,Rj} - F_{drag} \quad (1)$$

[0064] Lateral force balance may be described by the equation:

$$m[\dot{v}_y + \dot{\psi}v_x] = \sum_{j=L,R} [F_{x,Fj} \sin(\delta) + F_{y,Fj} \cos(\delta)] + \sum_{j=L,R} F_{y,Rj} + F_y^{j2c} \quad (2)$$

[0065] Yaw moment balance may be described by the equation:

$$J_z \ddot{\psi} = l_F [F_{x,FL} \sin(\delta) + F_{y,FL} \cos(\delta) + F_{x,FR} \sin(\delta) + F_{y,FR} \cos(\delta)] - l_R [F_{x,RL} + F_{y,RR}] - \frac{l_F}{2} [F_{x,RL} + F_{x,RR}] - \frac{l_F}{2} [F_{x,FL} \cos(\delta) -$$

-continued

$$F_{x,FR} \cos(\delta)] + \frac{l_F}{2} [F_{y,FL} \sin(\delta) - F_{y,FR} \sin(\delta)] + cF_y^{j2c} + \sum_{\substack{j=L,R \\ i=F,R}} M_{z,ij}$$

[0066] Moment balance for a wheel ij may be described by the equation

$$J_{w,i} \ddot{\theta} = \tau_{m,ij} - \tau_{b,ij} - R_{i,ij} F_{x,ij} - M_{y,ij} \quad (4)$$

[0067] In equations (1)-(4), m is the towing vehicle mass, v_x is the component of the velocity vector at the center of gravity along the longitudinal axis of the vehicle reference system, v_y is the component of the velocity vector at the center of gravity along the lateral axis of the vehicle reference system; $\dot{\psi}$ is the vehicle yaw rate, $F_{x,ij}$ is the longitudinal force of the ij-th tire/wheel, $F_{y,ij}$ is the lateral force of the ij-th wheel, F_x^{j2c} , F_y^{j2c} , F_z^{j2c} are the hitch joint force components along the longitudinal, lateral and vertical axes of the reference system of the vehicle, respectively, δ is an (e.g., average) steering angle of the front wheels. F_{drag} is an aerodynamic drag force, J_z is the yaw mass moment of inertia of the car, $M_{z,ij}$ is a self-alignment moment of the ij-th wheel, l_F and l_R are the front and rear wheelbases of the vehicle, t_F and t_R are the front and rear track widths, c is the distance of the hitch joint to the CoG of the car. $J_{w,i}^{eq}$ is the equivalent mass moment of inertia of the front wheels, which may be approximated as equal to the polar inertia, $\tau_{m,ij}$ are the wheel motor/powertrain torques, and $\tau_{b,ij}$ are the wheel braking torques. In this example, the Magic Formula (e.g., version 2002 or another nonlinear tire model) may be used to determine the rolling radius, laden radius, rolling resistance, longitudinal force, lateral force, and self-aligning moment of the tires, namely $R_{e,ij}$, $R_{l,ij}$, $M_{y,ij}$, $F_{x,ij}$, $F_{y,ij}$ and $M_{z,ij}$, as functions of the tire slip angle α_{ij} , longitudinal slip ratio $\sigma_{x,i,j}$, vertical load $F_{z,ij}$ which is expressed as a function of F_x^{j2c} , F_y^{j2c} , F_z^{j2c} , F_x^{j2c} , the longitudinal load transfer and the lateral load transfer, including the anti-roll moment contribution due to the active suspension actuation. Alternative formulations can be used in the example of a semi-active suspension system. The vertical loads may be expressed as:

$$\begin{aligned} F_{z,Fj} &= \frac{1}{2} F_z^{stat} + \frac{1}{2} \Delta F_{z,F}^x + k_1 \Delta F_{z,F}^y \\ F_{z,Rj} &= \frac{1}{2} F_z^{stat} + \frac{1}{2} \Delta F_{z,R}^x + k_1 \Delta F_{z,R}^y \\ F_{z,F}^{stat} &= \frac{mg l_R}{l} + F_z^{j2c,stat} \frac{c - l_R}{l} \\ F_{z,R}^{stat} &= \frac{mg l_F}{l} - F_z^{j2c,stat} \frac{c + l_F}{l} \\ \Delta F_{z,F}^x &= -ma_x \frac{H_{CoG}}{l} + F_x^{j2c} \frac{H_{Hitch}}{l} + \Delta F_{z,t}^x \frac{c - l_R}{l} \\ \Delta F_{z,R}^x &= ma_x \frac{H_{CoG}}{l} - F_x^{j2c} \frac{H_{Hitch}}{l} - \Delta F_{z,t}^x \frac{c + l_F}{l} \\ \Delta F_{z,F}^y &= ma_x \frac{H_{CoG}}{l} - F_x^{j2c} \frac{H_{Hitch}}{l} - \Delta F_{z,t}^y \frac{c + l_F}{l} \\ \Delta F_{z,R}^y &= -ma_x \frac{H_{CoG}}{l} + F_x^{j2c} \frac{H_{Hitch}}{l} + \Delta F_{z,t}^y \frac{c - l_R}{l} \\ \Delta F_{z,f}^y &= \left[ma_y \frac{l_R}{l} + F_y^{j2c} \frac{c - l_R}{l} \right] \frac{H_{RC}}{d_F} + \frac{M_{AR,F}^{tot}}{d_F} \\ \Delta F_{z,R}^y &= \left[ma_y \frac{l_F}{l} - F_y^{j2c} \frac{c + l_F}{l} \right] \frac{H_{RC}}{d_R} + \frac{M_{AR,R}^{tot}}{d_R} \end{aligned} \quad (5)$$

where H_{CoG} is the height of the CoG, H_{RC} is the roll axis height at the longitudinal coordinate of the CoG, $F_z^{j2c,stat}$ is a static contribution of the vertical component of the hitch joint force, and ΔF_z^x is the longitudinal load transfer contribution of the vertical component of the hitch joint force due to the trailer longitudinal acceleration. In this example, the total active suspension anti-roll moment on the front and rear axles, $M_{AR,F}^{tot}$ and $M_{AR,R}^{tot}$, may be determined using the equations:

$$\begin{aligned} M_{AR,F}^{tot} &= M_{AR,F}^{passive} + M_{AR,F}^{AS} \\ M_{AR,R}^{tot} &= M_{AR,R}^{passive} + M_{AR,R}^{AS} \end{aligned} \quad (6)$$

where the passive vehicle anti-roll moment on the i -th axle, $M_{AR,i}^{passive}$, and the active suspension anti-roll moments $M_{AR,F}$ include the hitch joint force components contributions, and may be determined using the equations:

$$\begin{aligned} M_{AR,F}^{passive} &= \frac{K_{\phi,F}}{K_{\phi,F} + K_{\phi,R}} \left([1 - K_{comp}] m_{a_y} [H_{CoG} - H_{RC}] - F_y^{j2c} [H_{Hitch} - H_{RC}] \right) \\ M_{AR,R}^{passive} &= \frac{K_{\phi,R}}{K_{\phi,F} + K_{\phi,R}} \left([1 - K_{comp}] m_{a_y} [H_{CoG} - H_{RC}] - F_y^{j2c} [H_{Hitch} - H_{RC}] \right) \\ M_{AR,F}^{AS} &= K_{comp} f m_{a_y} [H_{CoG} - H_{RC}] \\ M_{AR,R}^{AS} &= K_{comp} [1 - f] m_{a_y} [H_{CoG} - H_{RC}] \end{aligned}$$

where $K_{\phi,i}$ is the roll stiffness of the considered axle, K_{comp} is a factor expressing the level of roll angle compensation of the active suspension actuation, and $f = f_{AR,0} + \Delta f$ is the controllable front-to-total anti-roll moment distribution. The total active suspension anti-roll moment may have different formulations, depending on the control implementation.

[0068] Suspension stiffness and damping characteristic may be known predetermined values. The total passive anti-roll moment may depend on the roll dynamics of the rigid vehicle and may be determined by the control module using the equations:

$$\begin{aligned} M_{AR,\phi,i}^{passive} &= K_{\phi,i} \dot{\phi} \\ M_{AR,\phi,i}^{passive} &= K_{\phi,i} \phi \end{aligned} \quad (8)$$

where ϕ is the roll angle, $\dot{\phi}$ is the roll rate, $K_{\phi,i}$ is a roll damping coefficient, $M_{AR,\phi,i}^{passive}$ is the anti-roll moment due to the passive suspension roll stiffness, and $M_{AR,\phi,i}^{passive}$ is the anti-roll moment due to passive suspension roll damping properties.

[0069] This example may involve extension of the internal model to the equations of motion by an additional degree of freedom, namely a roll dynamics contribution:

$$\begin{aligned} I_x \ddot{\phi} &= m_{a_y} [H_{CoG} - H_{RC}] \cos(\phi) + m_g [H_{CoG} - H_{RC}] \sin(\phi) - \\ &M_{AR,\phi,F}^{passive} - M_{AR,\phi,R}^{passive} - M_{AR,\phi,F}^{passive} - M_{AR,\phi,R}^{passive} - M_{AR,F}^{AS} - M_{AR,R}^{AS} + \\ &\frac{m_t g l_{tF}}{l_t} [H_{Hitch} - H_{RC}] \sin(\phi) + F_y^{j2c} [H_{Hitch} - H_{RC}] \cos(\phi) \end{aligned} \quad (9)$$

where m_t is the mass of the trailer, l_{tF} is the hitch joint to trailer CoG distance, $\alpha_{y,t}$ and $\alpha_{x,t}$ are the acceleration components of the trailer, and F_y^{j2c} depends on the trailer acceleration components, trailer parameters, and trailer hitch

angle. The roll dynamics may be disregarded, so sensing devices on the trailer may not be included, such as to measure θ , $\alpha_{y,t}$ and $\alpha_{x,t}$.

[0070] The longitudinal and lateral slips $\sigma_{x,i,j}$ and α_{ij} may be determined by the control module using the equations:

$$\alpha_{ij} = \tan^{-1} \left(\frac{v_y \pm \dot{\psi} l_i}{v_x \mp \frac{d_i}{2} \dot{\psi}} \right) - k_2 \delta \quad \begin{cases} k_2 = 1 & \text{if } i = F \\ k_2 = 0 & \text{if } i = R \end{cases} \quad (10)$$

$$\sigma_{x,ij} = \frac{\omega_{ij} R_{e,ij} - v_{x,ij}}{v_{x,ij}} \quad (11)$$

where $v_{x,ij}$ is the longitudinal component of the peripheral wheel speed.

[0071] The prediction model could have different formulations. For example, the prediction model may not involve wheel dynamics.

[0072] In various implementations, the hitch force module 420 may determine hitch joint force components (or hitch forces) based on the following assumptions: the yaw and pitch acceleration of the trailer 104 are disregarded; and the longitudinal and lateral accelerations of the trailer $\alpha_{x,t}$ and $\alpha_{y,t}$ are approximated by the hitch force module 420 based on the acceleration components of the vehicle 100.

[0073] FIGS. 1 and 2 show the orientation of the hitch joint force components together with the sign conventions for the trailer variables. The longitudinal component $F_{x,t}^{j2c}$ is determined by the hitch force module 420 based on the longitudinal forces balance, such as using the equation:

$$F_{x,t}^{j2c} = -m_t \alpha_{x,t} \approx -m_t \alpha_x \quad (12)$$

where m_t is the mass of the trailer 104. Considering the example of FIG. 1, the rotation balance with respect to the CoG of the trailer axle may be used by the hitch force module 420 to determine the latitudinal force, such as using the equation:

$$F_{y,t}^{j2c} = -\frac{m_t a_{y,t} l_{tR}}{l_t} \approx -\frac{m_t a_y l_{tR}}{l_t} \quad (13)$$

where l_{tR} is the trailer CoG to the trailer axle distance, and l_t is the hitch joint to trailer axle distance. $F_{x,t}^{j2c}$ and $F_{y,t}^{j2c}$ may be transposed by the hitch force module 420 into the towing vehicle coordinate system and defined as:

$$F_x^{j2c} = -m_t a_x \cos(\theta_{kin}) - \frac{m_t a_y l_{tR}}{l_t} \sin(\theta_{kin}) \quad (14)$$

$$F_y^{j2c} = m_t a_x \sin(\theta_{kin}) - \frac{m_t a_y l_{tR}}{l_t} \cos(\theta_{kin}) \quad (15)$$

where θ_{kin} is the kinematic hitch angle, the hitch angle in absence of slip angles. An example equation describing the evolution of the kinematic hitch angle for a given vehicle speed is:

$$\theta_{kin} = \tan^{-1} \left(\frac{\tan(\delta) [l^2 l_t + [c - l_R] \sqrt{\tan(\delta)^2 l^2 [c - l_R]^2 - \tan(\delta)^2 l^2 l_t^2 + l^4}]}{l [-\tan(\delta)^2 l_t [c - l_R] + \sqrt{\tan(\delta)^2 l^2 [c - l_R]^2 - \tan(\delta)^2 l^2 l_t^2 + l^4}]} \right) \quad (16)$$

[0074] Alternative formulations for computing hitch angle may be based on its value in quasi-steady-state cornering conditions, such as mapped in look-up tables as functions of steering angle, vehicle speed, and tire-road friction coefficient. Alternatively, a state observer including the trailer dynamics may be used to determine the hitch angle.

[0075] Considering the example in FIG. 2, the rotation balance with respect to the CoG of the trailer axle can be used to determine the vertical hitch force by the hitch force module 420, such as using the equation:

$$F_{z,t}^{j2c} = F_{z,t}^{j2c,stat} + \Delta F_{z,t}^x = -\frac{m_t g l_{R,t}}{l_t} + \frac{m_t a_{x,t} [H_{CoG,t} - H_{Hitch}]}{l_t} \cong -\frac{m_t g l_{R,t}}{l_t} + \frac{m_t a_x [H_{CoG,t} - H_{Hitch}]}{l_t} \quad (17)$$

$F_{z,t}^{j2c}$ assumes the same value when transposed to the towing car coordinate system.

[0076] The NMPC model may be expressed by the following continuous formulation

$$\dot{x}(t) = f(x(t), u(t)) \quad (18)$$

where the state vector for the articulated vehicle model and the state vector for the rigid vehicle model x is:

$$x = [v_x, v_y, \dot{\psi}, \omega_{FL}, \omega_{FR}, \omega_{RL}, \omega_{RR}] \quad (19)$$

and the control vector u of the control module is:

$$u = [\Delta f, \tau_{m,FL}, \tau_{m,FR}, \tau_{m,RL}, \tau_{m,RR}] \quad (20)$$

[0077] In example of the roll dynamics being included in the NMPC model formulation, the following state vector may be considered:

$$x_\phi = [v_x, v_y, \dot{\psi}, \omega_{FL}, \omega_{FR}, \omega_{RL}, \omega_{RR}] \quad (21)$$

[0078] The NMPC control law may minimize a cost function J , subject to equality and inequality constraints. The nonlinear optimal control problem may be defined in discrete time as:

$$\min_u J(x(0), u(\cdot) := \ell_N(x(N)) + \sum_{k=0}^{N-1} \ell(x(k), u(k)) \quad (22)$$

s.t.

$$x(0) = x_{in}$$

$$x(k+1) = f_a(x(k), u(k))$$

$$\underline{x} \leq x(k) \leq \bar{x}$$

$$\underline{x} \leq x(N) \leq \bar{x}$$

$$\underline{u} \leq u(k) \leq \bar{u}$$

$$u(\cdot) : [0, N-1]$$

where $\ell_N(x(N))$ is a terminal cost, N is the number of steps of the prediction horizon H_p (e.g., equal to the control

horizon H_c , i.e. $H_p = H_c = N T_s$ with T_s being the sampling time, but could be a different value), k indicates the discretization step, $\underline{x}$ and $\bar{x}$ are the lower and upper limits for x , $\underline{u}$ and $\bar{u}$ are the lower and upper limits for u , $x(k+1) = f_a(x(k), u(k))$ is a discretized model defined in equation (18). The stage cost functions may be defined as a least-squares functions and are given by equation (23) for configurations with the articulated and the isolated vehicle model, respectively.

$$\ell(x(k), u(k)) = W_{Q,\Delta\psi} \Delta\dot{\psi}^2 + W_{Q,\Delta\varphi} \Delta\varphi^2 + \quad (24)$$

$$W_{Q,\Delta\tau_{m,tot}} \Delta\tau_{m,tot}^2 + W_{R,\tau_{m,ij}} \sum \tau_{m,ij}^2 + W_{R,\Delta f} \Delta f^2 + W_{R,s_{\alpha_R}} s_{\alpha_R}^2$$

where $W_{Q,\dot{\psi}}$, $W_{Q,\Delta\varphi}$, $W_{Q,\Delta\tau_{m,tot}}$ and $W_{Q,\theta}$ are the cost function weights, respectively, tracking of the rigid vehicle reference yaw rate of the rigid vehicle roll angle, of the torque demand (target) or of the articulation hitch angle,

$$W_{R,s_{\alpha_R}}$$

is a weighting coefficient referred to as a rear tire slip angle slack variable, $W_{R,\tau_{m,ij}}$ and $W_{R,\Delta f}$ are the cost function weights referred to as available actuators. The contributions in equation (23) are based on:

$$\Delta\dot{\psi} = \dot{\psi}_{ref} - \dot{\psi}, \text{ the yaw rate tracking error;}$$

$$\Delta\varphi = \varphi_{ref} - \varphi,$$

[0079] $\Delta\varphi = \text{RI}_{ref} - \text{RI}$, roll angle tracking error and rollover index (RI) minimization, respectively.

Alternatively, rollover prevention may be accounted for by adding a soft constraint on the rollover index through a slack variable, $\text{RI} \leq \text{RI}^{max}(1 + s_{RI})$, where $s_{RI} \geq 0$ is the slack variable. In the following application, the contribution of tracking the roll angle error may be disregarded.

[0080] $\Delta\tau_{m,tot}$ is a torque demand tracking error at the vehicle level between the driver demands and a reference total torque. Slack variables may be used to set the soft constraints.

[0081] The following state and control action bounds have been implemented in equation 22 as constraints (lower and upper limits):

[0082] Wheel torque constraints, i.e. $\underline{\tau}_m \leq \tau_{m,ij} \leq \bar{\tau}_m$, where $\underline{\tau}_m$ and $\bar{\tau}_m$ are the predetermined minimum and maximum values of the wheel torques, used to generate the direct yaw moment control action, in addition to the total traction or braking torques;

[0083] The front-to-total anti-roll moment distribution deviation constraint, $\underline{\Delta f} \leq \Delta f \leq \bar{\Delta f}$, where $\underline{\Delta f}$ and $\bar{\Delta f}$ are predetermined minimum and maximum anti-roll front-to-total moment distribution deviations from a nominal value;

[0084] Rear tire slip angle constraint to account for vehicle lateral stability, $|\alpha_R| \leq \alpha_R^{max}(1 + s_{\alpha_R})$, where $s_{\alpha_R} \geq 0$ is the slack variable; and α_R^{max} is the maximum admissible tire slip angle, for example, $\alpha_R^{max} = 3$ degrees or another suitable value.

[0085] Further constraints are possible, such as to prevent the activation of a stability controller. Constraints on the front slip angle of the vehicle, on the vehicle sideslip angle,

and on the tire slip ratios of the vehicle may additionally or alternatively be implemented.

[0086] FIG. 6 shows the schematics of a standalone AS implementation. Here, the driver inputs are APP, BPP and δ_{in} . Θ represents the vehicle operational parameters measured by sensors and received via a network bus of the vehicle. The driver inputs are converted into the reference yaw rate and torque demand, which together with the reference hitch forces and the actuator physical constraints, are conveyed to the control module as constant inputs in the prediction horizon. In various implementations, NMPC (nonlinear model predictive control) may be used by the control module. The NMPC includes a nonlinear optimal control problem, where the inputs undergo a nonlinear internal model to define the optimal control actions along the prediction horizon. The control actions are defined by minimizing the constrained objective function. The final outputs are the forces of the actuators 304-316 on each corner of the vehicle.

[0087] FIG. 7 is an example of an NMPC AS-DYC implementation, where DYC is a torque-vectoring actuation strategy involving control of the electric motors 324-336. Here, physical constraints of the motors are an additional input with respect to the standalone AS implementation of FIG. 6. Moreover, the final outputs of the nonlinear optimal control problem of the example of FIG. 7 are the vertical forces for the actuators 304-316 and the wheel torques for the electric motors 324-336.

[0088] Parallel or independent control of the actuators 304-316 and the wheel torques for the electric motors 324-336 may be performed. For example, an NMPC AS//DYC implementation with and without communication between the actuator and wheel control modules is illustrated in FIG. 8. In this example, a standalone AS NMPC implementation (e.g., FIG. 6) with an articulated vehicle internal model runs in parallel to a NMPC DYC implementation with an internal model formulation resembling NMPC AS, namely with the hitch angle dynamics to account for the trailer, represented by equations 19-21 and 23. The two subsystems (actuator and motor controls) run independently either sharing the information of their respective control actions which are taken into account as additional inputs by the other (with communication) or without sharing the information of their respective control actions with each other.

[0089] FIG. 9 includes an example NMPC AS//Standard DYC implementation with and without communication between the actuator and motor controls, where a standalone AS NMPC implementation (e.g., FIG. 6) with an articulated vehicle internal model in parallel to a standard NMPC DYC implementation with a rigid vehicle internal model, calibrated for an isolated vehicle. Here, the same communication options may be enabled or disabled. In FIGS. 8 and 9, where the switches A and B enable or disable the communication between the two controls running in parallel.

[0090] The NMPC can be implemented on a software environment for automatic control and dynamic optimization. The following may be used: Gauss Newton Hessian approximation, multiple shooting discretization, a qpOASES solver, a number of optimization steps may be set equal to 2, a discretization time of the internal models may be set to 5 ms (this may ensures their numerical stability without significantly affecting the computational time), for the integrator types and orders 2nd order Implicit Runge Kutta

(IRK-GL2) for the controls based on the articulated vehicle and 4th order Implicit Runge Kutta (IRK-GL4) for DYC based on the rigid vehicle model, sampling time for each articulated vehicle model may be 35 ms (milliseconds), while the sampling time for DYC implementation may be 15 ms. Example implementation settings are summarized in the following table with measured variables coming from the vehicle.

TABLE I

Example of Implemented Control Module Settings			
Actuation	Integrator	t_d (ms)	t_{sa} (ms)
AS	IRK-GL4	5	15
AS-DYC	IRK-GL4	5	15
AS//DYC	IRK-GL4	5	15
AS//Standard DYC	IRK-GL4	5	15

[0091] Testing proved that these parametrizations are a good compromise between performance and computational efficiency. The present application involves use of active suspension system on the towing vehicle to limit the trailer sway with, for example, a nonlinear model predictive control (NMPC) on an articulated vehicle model that includes the trailer parameters and dynamics. Additionally, the present application involves use of the active suspension and direct yaw control on the towing vehicle to preserve the articulated vehicle stability and to limit the trailer sway with nonlinear model predictive control coordination based on an articulated vehicle model which includes the trailer parameters and dynamics. Additionally, the present application involves use of active suspension and direct yaw moment control coordination with two parallel NMPC configuration based on an articulated vehicle model (or isolated vehicle model for the case of the Standard DYC) as an alternative in preserving the articulated vehicle stability and to limit the trailer sway.

[0092] The control described herein stabilizes the vehicle-trailer system as opposed to the passive vehicle. The active suspension control NMPC AS effectively tracks the towing vehicle reference yaw rate and tracks and limits the hitch force errors (differences between the hitch forces and reference hitch forces, respectively) and the trailer roll angle. AS plus DYC implementation can further improve the yaw and hitch damping capability relative to only AS control. Moreover, the trailer roll angle is further limited.

[0093] FIG. 10 is a flowchart depicting an example method of controlling one or more vehicle actuators based on hitch forces. Control begins with 1004 where the driver inputs, the hitch forces, and other vehicle operating parameters are received.

[0094] At 1008, the reference yaw rate module 408 determines the reference yaw rate based on the steering angle, the vehicle speed, and the tire-road coefficient of friction. The reference hitch forces module 412 also determines the reference hitch forces at 1012. At 1012, the target torque module 404 determines the target torque based on the driver inputs and the hitch forces.

[0095] At 1016, the actuator control module 320 determines target forces for the suspension actuators 304-316, respectively, based on the hitch forces, the reference hitch forces, the target torque, the reference yaw rate, and the yaw rate, such as using NMPC control. The actuator control module 320 may set the target forces, for example, to

minimize the difference (error) between the hitch forces and the reference hitch forces and to minimize the difference (error) between the yaw rate and the reference yaw rate.

[0096] Also at 1016, the wheel control module 340 may determine target torques for the motors 324-336, respectively, based on the hitch forces, the reference hitch forces, the target torque, the reference yaw rate, and the yaw rate, such as using NMPC control. The wheel control module 340 may set the target torques, for example, to minimize the difference (error) between the hitch forces and the reference hitch forces and to minimize the difference (error) between the yaw rate and the reference yaw rate. In various implementations, the wheel control module 340 may determine the target torques based on the target forces and/or determine the target forces based on the target torques.

[0097] At 1020, the actuator control module 320 controls the suspension actuators 304-316 to achieve the target (vertical) forces, respectively. The actuator control module 320 may, for example, apply power or hydraulic fluid to the suspension actuators 304-326 based on the target forces, respectively. The wheel control module 340 may also apply power to the electric motors 328-336 to achieve the target torques, respectively. Control returns to 1004 for a next control loop.

[0098] The foregoing description is merely illustrative in nature and is in no way intended to limit the disclosure, its application, or uses. The broad teachings of the disclosure can be implemented in a variety of forms. Therefore, while this disclosure includes particular examples, the true scope of the disclosure should not be so limited since other modifications will become apparent upon a study of the drawings, the specification, and the following claims. It should be understood that one or more steps within a method may be executed in different order (or concurrently) without altering the principles of the present disclosure. Further, although each of the embodiments is described above as having certain features, any one or more of those features described with respect to any embodiment of the disclosure can be implemented in and/or combined with features of any of the other embodiments, even if that combination is not explicitly described. In other words, the described embodiments are not mutually exclusive, and permutations of one or more embodiments with one another remain within the scope of this disclosure.

[0099] Spatial and functional relationships between elements (for example, between modules, circuit elements, semiconductor layers, etc.) are described using various terms, including “connected,” “engaged,” “coupled,” “adjacent,” “next to,” “on top of,” “above,” “below,” and “disposed.” Unless explicitly described as being “direct,” when a relationship between first and second elements is described in the above disclosure, that relationship can be a direct relationship where no other intervening elements are present between the first and second elements, but can also be an indirect relationship where one or more intervening elements are present (either spatially or functionally) between the first and second elements. As used herein, the phrase at least one of A, B, and C should be construed to mean a logical (A OR B OR C), using a non-exclusive logical OR, and should not be construed to mean “at least one of A, at least one of B, and at least one of C.”

[0100] In the figures, the direction of an arrow, as indicated by the arrowhead, generally demonstrates the flow of information (such as data or instructions) that is of interest

to the illustration. For example, when element A and element B exchange a variety of information but information transmitted from element A to element B is relevant to the illustration, the arrow may point from element A to element B. This unidirectional arrow does not imply that no other information is transmitted from element B to element A. Further, for information sent from element A to element B, element B may send requests for, or receipt acknowledgements of, the information to element A.

[0101] In this application, including the definitions below, the term “module” or the term “controller” may be replaced with the term “circuit.” The term “module” may refer to, be part of, or include: an Application Specific Integrated Circuit (ASIC); a digital, analog, or mixed analog/digital discrete circuit; a digital, analog, or mixed analog/digital integrated circuit; a combinational logic circuit; a field programmable gate array (FPGA); a processor circuit (shared, dedicated, or group) that executes code; a memory circuit (shared, dedicated, or group) that stores code executed by the processor circuit; other suitable hardware components that provide the described functionality; or a combination of some or all of the above, such as in a system-on-chip.

[0102] The module may include one or more interface circuits. In some examples, the interface circuits may include wired or wireless interfaces that are connected to a local area network (LAN), the Internet, a wide area network (WAN), or combinations thereof. The functionality of any given module of the present disclosure may be distributed among multiple modules that are connected via interface circuits. For example, multiple modules may allow load balancing. In a further example, a server (also known as remote, or cloud) module may accomplish some functionality on behalf of a client module.

[0103] The term code, as used above, may include software, firmware, and/or microcode, and may refer to programs, routines, functions, classes, data structures, and/or objects. The term shared processor circuit encompasses a single processor circuit that executes some or all code from multiple modules. The term group processor circuit encompasses a processor circuit that, in combination with additional processor circuits, executes some or all code from one or more modules. References to multiple processor circuits encompass multiple processor circuits on discrete dies, multiple processor circuits on a single die, multiple cores of a single processor circuit, multiple threads of a single processor circuit, or a combination of the above. The term shared memory circuit encompasses a single memory circuit that stores some or all code from multiple modules. The term group memory circuit encompasses a memory circuit that, in combination with additional memories, stores some or all code from one or more modules.

[0104] The term memory circuit is a subset of the term computer-readable medium. The term computer-readable medium, as used herein, does not encompass transitory electrical or electromagnetic signals propagating through a medium (such as on a carrier wave); the term computer-readable medium may therefore be considered tangible and non-transitory. Non-limiting examples of a non-transitory, tangible computer-readable medium are nonvolatile memory circuits (such as a flash memory circuit, an erasable programmable read-only memory circuit, or a mask read-only memory circuit), volatile memory circuits (such as a static random access memory circuit or a dynamic random access memory circuit), magnetic storage media (such as an

analog or digital magnetic tape or a hard disk drive), and optical storage media (such as a CD, a DVD, or a Blu-ray Disc).

[0105] The apparatuses and methods described in this application may be partially or fully implemented by a special purpose computer created by configuring a general purpose computer to execute one or more particular functions embodied in computer programs. The functional blocks, flowchart components, and other elements described above serve as software specifications, which can be translated into the computer programs by the routine work of a skilled technician or programmer.

[0106] The computer programs include processor-executable instructions that are stored on at least one non-transitory, tangible computer-readable medium. The computer programs may also include or rely on stored data. The computer programs may encompass a basic input/output system (BIOS) that interacts with hardware of the special purpose computer, device drivers that interact with particular devices of the special purpose computer, one or more operating systems, user applications, background services, background applications, etc.

[0107] The computer programs may include: (i) descriptive text to be parsed, such as HTML (hypertext markup language), XML (extensible markup language), or JSON (JavaScript Object Notation) (ii) assembly code, (iii) object code generated from source code by a compiler, (iv) source code for execution by an interpreter, (v) source code for compilation and execution by a just-in-time compiler, etc. As examples only, source code may be written using syntax from languages including C, C++, C #, Objective-C, Swift, Haskell, Go, SQL, R, Lisp, Java®, Fortran, Perl, Pascal, Curl, OCaml, Javascript®, HTML5 (Hypertext Markup Language 5th revision), Ada, ASP (Active Server Pages), PHP (PHP: Hypertext Preprocessor), Scala, Eiffel, Smalltalk, Erlang, Ruby, Flash®, Visual Basic®, Lua, MATLAB, SIMULINK, and Python®.

What is claimed is:

1. A trailer sway control system for a vehicle, comprising:
 - a front left active suspension actuator associated with a front left wheel of the vehicle;
 - a front right active suspension actuator associated with a front right wheel of the vehicle;
 - a rear left active suspension actuator associated with a rear left wheel of the vehicle;
 - a rear right active suspension actuator associated with a rear right wheel of the vehicle; and
 - an actuator control module configured to:
 - based on (a) a first hitch force at a trailer hitch of the vehicle in a longitudinal direction, (b) a second hitch force at the trailer hitch of the vehicle in a lateral direction, and (c) a third hitch force at the trailer hitch of the vehicle in a vertical direction, determine target vertical forces for the front left active suspension actuator, the front right active suspension actuator, the rear left active suspension actuator, and the rear right active suspension actuator, respectively; and
 - selectively adjust the front left active suspension actuator, the front right active suspension actuator, the rear left active suspension actuator, and the rear right active suspension actuator based on the target vertical forces, respectively.

2. The trailer sway control system of claim 1 wherein the actuator control module is configured to determine the target vertical forces based on minimizing (a) a first difference between the first hitch force and a first reference hitch force, (b) a second difference between the second hitch force and a second reference hitch force, and (c) a third difference between the third hitch force and a third reference hitch force.

3. The trailer sway control system of claim 2 further comprising a reference hitch forces module configured to set the first reference hitch force, the second reference hitch force, and the third reference hitch force based on a steering angle of the vehicle.

4. The trailer sway control system of claim 1 wherein the actuator control module is configured to determine the target vertical forces further based on a yaw rate of the vehicle.

5. The trailer sway control system of claim 4 wherein the actuator control module is configured to determine the target vertical forces based on minimizing a difference between the yaw rate and a reference yaw rate.

6. The trailer sway control system of claim 5 further comprising a reference yaw rate module configured to set the reference yaw rate based on a steering angle of the vehicle.

7. The trailer sway control system of claim 6 wherein the reference yaw rate module is configured to set the reference yaw rate further based on a vehicle speed.

8. The trailer sway control system of claim 1 further comprising:

- a front left electric motor configured to output torque to the front left wheel of the vehicle;
- a front right electric motor configured to output torque to the front right wheel of the vehicle;
- a rear left electric motor configured to output torque to the rear left wheel of the vehicle;
- a rear right electric motor configured to output torque to the rear right wheel of the vehicle; and
- a wheel control module configured to:

- based on the first hitch force, the second hitch force, and the third hitch force, determine target torque outputs of the front left electric motor, the front right electric motor, the rear left electric motor, and the rear right electric motor, respectively; and
- selectively adjust the front left electric motor, the front right electric motor, the rear left electric motor, and the rear right electric motor based on the target torque outputs, respectively.

9. The trailer sway control system of claim 8 wherein the wheel control module is configured to determine the target torque outputs further based on the target vertical forces.

10. The trailer sway control system of claim 8 wherein the actuator control module is configured to determine the target vertical forces further based on the target torque outputs.

11. The trailer sway control system of claim 1 further comprising:

- a front electric motor configured to output torque to front wheels of the vehicle;
- a rear electric motor configured to output torque to front wheels of the vehicle; and
- a wheel control module configured to:
 - based on the hitch angle, determine target torque outputs of the front and rear electric motors, respectively; and

selectively adjust the front and rear electric motors based on the target torque outputs, respectively.

12. The trailer sway control system of claim **1** further comprising:

an electric motor configured to output torque two or more wheels of the vehicle; and

a wheel control module configured to:

based on the hitch angle, determine a target torque of the electric motor; and

selectively adjust the electric motor based on the target torque.

13. A trailer sway control method for a vehicle, comprising:

based on (a) a first hitch force at a trailer hitch of the vehicle in a longitudinal direction, (b) a second hitch force at the trailer hitch of the vehicle in a lateral direction, and (c) a third hitch force at the trailer hitch of the vehicle in a vertical direction, determining target vertical forces for a front left active suspension actuator, a front right active suspension actuator, a rear left active suspension actuator, and a rear right active suspension actuator, respectively,

the front right active suspension actuator associated with a front right wheel of the vehicle, the front left active suspension actuator associated with a front left wheel of the vehicle, the rear left active suspension actuator associated with a rear left wheel of the vehicle, and the rear right active suspension actuator associated with a rear right wheel of the vehicle; and

selectively adjusting the front left active suspension actuator, the front right active suspension actuator, the rear left active suspension actuator, and the rear right active suspension actuator based on the target vertical forces, respectively.

14. The trailer sway control method of claim **13** wherein determining the target vertical forces includes determining the target vertical forces based on minimizing (a) a first difference between the first hitch force and a first reference hitch force, (b) a second difference between the second hitch force and a second reference hitch force, and (c) a third difference between the third hitch force and a third reference hitch force.

15. The trailer sway control method of claim **14** further comprising setting the first reference hitch force, the second reference hitch force, and the third reference hitch force based on a steering angle of the vehicle.

16. The trailer sway control method of claim **13** wherein determining the target vertical forces includes determining the target vertical forces further based on a yaw rate of the vehicle.

17. The trailer sway control method of claim **16** wherein determining the target vertical forces includes determining the target vertical forces based on minimizing a difference between the yaw rate and a reference yaw rate.

18. The trailer sway control method of claim **17** further comprising setting the reference yaw rate based on a steering angle of the vehicle.

19. The trailer sway control method of claim **18** wherein setting the reference yaw rate includes setting the reference yaw rate further based on a vehicle speed.

20. The trailer sway control method of claim **13** further comprising:

based on the hitch angle, determining a target torque of an electric motor configured to output torque two or more wheels of the vehicle; and

selectively adjust the electric motor based on the target torque.

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