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UNITED STATES DISTRICT COURT

CENTRAL DISTRICT OF CALIFORNIA

CV 13 - 00801 PA SPx

dw-link Incorporated, Plaintiff, v. Giant Bicycle, Inc., and Giant Manufacturing Co., Ltd., Defendants.	Case No. **COMPLAINT FOR:** **(1) PATENT INFRINGEMENT** **(2) BREACH OF CONTRACT** **(3) UNJUST ENRICHMENT** **DEMAND FOR JURY TRIAL** By Fax

Plaintiff dw-link Incorporated ("dw-link"), for its Complaint against Defendants Giant Bicycle, Inc. "(Giant-USA") and Giant Manufacturing Co., Ltd. ("Giant-Taiwan"), alleges and states as follows:

JURISDICTION AND VENUE

1. This Court has jurisdiction over this action pursuant to 28 U.S.C. §§ 1331 and 1338 in that this is an action for patent infringement in violation of the patent laws of the United States, 35 U.S.C. §§ 1, *et seq.*, as hereinafter more fully appears. This Court also has jurisdiction over this action pursuant to 28 U.S.C. § 1332 in that the parties are citizens of different states and of a foreign state and the amount in controversy exceeds the sum of $75,000.00 exclusive of interest and costs as hereinafter more fully appears. This Court further has supplemental jurisdiction over certain claims asserted in this action pursuant to 28 U.S.C. § 1367.

2. Venue is proper in this Court pursuant to 28 U.S.C. §§ 1391 and 1400, in that Giant-USA and Giant-Taiwan reside in this judicial district and/or have committed acts of infringement in this judicial district.

PARTIES

3. dw-link is a Massachusetts corporation with its principle place of business located at 11 Boldt Farms Road, Edgartown, MA 02539.

4. Upon information and belief, Giant-USA is a Virginia corporation with its principle place of business located at 3587 Old Conejo Road, Newbury Park, CA 91320.

5. Upon information and belief, Giant-Taiwan is a foreign corporation with its principle place of business located at No. 19, Shun-Farn Road, Tachia, Taichung, Taiwan 43774 R.O.C.

6. Upon information and belief, Giant-USA is a wholly-owned subsidiary, either directly or indirectly, of Giant-Taiwan. Upon information and belief, Giant-Taiwan is the largest manufacturer of bicycles in the world.

FACTS

7. In 2003 David Weagle ("Weagle") filed a patent application on a new and innovative bicycle suspension system. His patent application was duly and validly issued by the Unites States Patent and Trademark Office on May 23, 2006, as U.S. Patent No. 7,048,292 B2 ("the '292 Patent"). A true and correct copy of the '292 Patent is attached as Exhibit A. The '292 Patent has been properly and legally assigned to dw-link.

8. In 2004, Weagle filed a continuation-in-part patent application based upon the original filing of the '292 Patent. This continuation-in-part application was duly and validly issued by the United States Patent and Trademark Office on October 31, 2006, as U.S. Patent No. 7,128,329 B2. This patent has been properly and legally assigned to dw-link.

9. On July 27, 2007, dw-link filed a request for reexamination of U.S. Patent No. 7,128,329 B2. On October 14, 2008, the United States Patent Office confirmed the patentability of the inventions disclosed in U.S. Patent No. 7,128,329 B2 and issued Ex Parte Reexamination Certificate No. 7,128,329 C1 ("the '329 Patent"). A true and correct copy of the '329 Patent, as originally issued and including the reexamination certificate, is attached as Exhibit B.

10. On September 22, 2006, Weagle filed a division of the application which issued as the '329 Patent. This divisional application was duly and validly issued by the United States Patent Office on November 9, 2010, as U.S. Patent No. 7,828,314 B2 ("the '314 Patent"). A true and correct copy of the '314 Patent is attached as Exhibit C. The '314 Patent has been properly and legally assigned to dw-link.

11. The '292 Patent, '329 Patent, and '314 Patent (collectively, "the dw-link Patents") have been recognized throughout the bicycle industry as innovative, ground-breaking, and significant advancements in bicycle suspension technology.

The dw-link Patents have been licensed to, and are lawfully practiced by, numerous companies around the world.

12. dw-link has complied with, and ensured compliance by its licensees with, the patent marking requirements of the United States patent laws as required by 35 U.S.C. § 287.

13. In July 2004, Weagle attended the H3 Publications Symposium in Whistler, British Columbia, Canada. He had at the Symposium a prototype of his new, patent pending, bicycle suspension system.

14. At the Symposium, Weagle spoke with Jeff Menown and Dennis Lane of Giant-USA and explained his dw-link bicycle suspension system to them. Weagle also advised them that the dw-link bicycle suspension system was patent pending world-wide. Menown and Lane discussed Giant's pre-existing suspension system and their belief that the pre-existing design needed to be changed for production due to likely infringement of another entity's patents.

15. Thereafter, Giant-Taiwan and Giant-USA changed their pre-existing bicycle suspension system to be essentially a copy of the patent-pending dw-link bicycle suspension system.

16. In 2005, Giant-USA and Giant-Taiwan launched their revised design as the "Maestro" suspension system on nearly all of their bicycles. The bicycle models manufactured and sold by Giant-USA and Giant-Taiwan in the United States which include their "Maestro" suspension system include, but are not limited to, the Anthem, Trance, Trance X, Reign, Reign X, Anthem X 29, Trance X 29, Faith, and Glory product models and submodels of those models, such as Anthem Advanced, Anthem 0, Trance Advanced, and Reign X1.

17. Giant-Taiwan manufactures, exports, and ships bicycles utilizing the "Maestro" suspension system to Giant-USA. Giant-USA imports bicycles utilizing the "Maestro" suspension system into the United States and then distributes, offers to sell, and sells them throughout the United States.

18. On February 7, 2007, dw-link sent a letter to Giant-USA notifying Giant-USA, *inter alia*, that all bicycles which utilized their "Maestro" suspension system infringed all claims of the '329 Patent.

19. On May 4, 2007, Weagle and counsel for Giant-USA had a telephone conversation in which Giant-USA's counsel verified that Giant-USA was interested in "entering into a business arrangement with dw-link that may include Giant acquiring certain rights in dw-link's patent portfolio," which included at that time the dw-link Patents and their foreign equivalents.

20. On June 1, 2007, counsel for Giant-USA confirmed that Giant-USA was interested in acquiring the dw-link Patents and/or at a minimum a license under the '329 Patent.

21. On or about June 21, 2007, Giant-USA communicated an offer to purchase the '329 Patent from dw-link.

22. dw-link found the offer from Giant-USA to be inadequate. Accordingly, on June 21, 2007, dw-link reminded Giant-USA that it was infringing at least the '329 Patent and that Giant-USA needed to cease all infringing activities or acquire rights to permit it to lawfully practice the inventions disclosed in at least the '329 Patent.

23. In an email dated October 30, 2007, counsel for Giant-USA stated that, "Giant remains interested in your patent portfolio."

24. In an email dated November 19, 2007, counsel for Giant-USA advised that it wished to await the results of the reexamination of the '329 Patent, which had been filed several months earlier. Counsel for Giant USA reaffirmed, however, that, "Giant is interested in the '329 Patent if it survives re-exam. Remember that you previously asserted that Giant infringed the '329 Patent."

25. On November 20, 2007, dw-link sent an offer to license the '329 Patent to Giant-USA on certain terms and conditions.

26. On November 27, 2007, counsel for Giant-USA responded by email

stating that, "Giant believes that it would be best to wait until the results of the Re-examination proceeding in order to enter into a license agreement for the '329 Patent."

27. Throughout the communications between dw-link and Giant-USA in 2007 and thereafter, Giant-USA never denied its infringement of the '329 Patent.

28. The reexamination certificate of the '329 Patent, confirming its validity, did not materially alter the claims of the '329 Patent and did not change the fact that Giant-USA and Giant-Taiwan's bicycles, which use the "Maestro" suspension system, infringe the '329 Patent.

29. From 2007 until early 2009, Giant-USA's counsel continued to negotiate with dw-link. Finally, on March 17, 2009, Weagle, of dw-link, met with Owen Chang, Teddy Chang, and Patrick Wu of Giant-Taiwan for the purpose of negotiating an agreement between dw-link and Giant-Taiwan. Giant-Taiwan understood the infringement by its bicycles, which utilized the "Maestro" suspension system, and indicated it would rather enter into an agreement with dw-link.

30. Negotiations between dw-link and Giant-Taiwan continued throughout 2009 and 2010. At no time during these negotiations did Giant-Taiwan deny infringement of the dw-link Patents or challenge the validity of the dw-link Patents.

31. On April 15, 2010, dw-link and Giant-Taiwan entered into a Non-Disclosure and Restricted Use Agreement.

32. On April 18, 2010, dw-link and Giant-Taiwan entered into a Letter of Intent regarding certain terms for an agreement between them.

33. Finally, as of November 17, 2010, dw-link and Giant-Taiwan entered into a Joint Development Agreement ("the JDA").

34. Among other terms, pursuant to the JDA, dw-link agreed to work with Giant-Taiwan to design a next generation mountain bike suspension system called "G+ Technology."

35. Among other terms, pursuant to the JDA, the Non-Disclosure and Restricted Use Agreement was nullified and a new non-disclosure and confidentiality section within the JDA was substituted.

36. In exchange for dw-link's work in connection with the JDA, Giant-Taiwan agreed, *inter alia*, to pay dw-link: (1) an initial payment upon execution of the JDA in the amount of US$600,000.00; (2) a payment of US$400,000.00 upon achievement of the "fifth milestone" as set forth in the JDA; (3) all expenses and costs incurred by dw-link in connection with the JDA; (4) royalties on future sales of bicycles utilizing the to-be-developed "G+ Technology"; and an advance royalty payment of US$30,000.00 for "the first three quarters of each fiscal year during the Term of Agreement [the JDA]."

37. Following execution of the JDA, Giant-Taiwan made the initial payment of US$600,000.00.

38. After execution of the JDA, dw-link began concentrated and significant work with Giant-Taiwan to design the new "G+ Technology."

39. dw-link expended considerable time and effort in designing and testing the new "G+ Technology." dw-link worked tirelessly to develop the new "G+ Technology." Such work, time and efforts considerably benefited Giant-Taiwan.

40. dw-link met and complied with all of its obligations and requirements under the JDA.

41. As dw-link's work under the JDA neared completion, however, it became clear that Giant-Taiwan intended to refuse acceptance of the new "G+ Technology." Giant-Taiwan delayed payments of expenses and costs incurred by dw-link pursuant to the terms of the JDA and failed to pay any of the advance royalty payments required under the JDA.

42. In early 2012, dw-link achieved the "fifth milestone" under the JDA, thereby obligating Giant-Taiwan to pay dw-link US$400,000.00.

43. Giant-Taiwan, however, claimed that the "Fifth Milestone" had not

been reached and refused to make the required US$400,000.00 payment to dw-link.

44. Giant-Taiwan and dw-link thereafter engaged in months of discussions to try to resolve the issue. Giant-Taiwan sought to re-write the JDA to add new requirements for achievement of the "Fifth Milestone," including requirements which would violate the laws of physics.

45. By letter dated October 16, 2012, dw-link provided a notice of various breaches of the JDA to Giant-USA and Giant-Taiwan. dw-link also reminded Giant-USA and Giant-Taiwan of their continuing infringement of at least the '329 Patent and the need to resolve this continuing infringement.

46. Giant-Taiwan thereafter responded in several communications, *inter alia*, denying breaches of the JDA. Neither Giant-USA nor Giant-Taiwan denied their continuing infringement of at least the '329 Patent.

47. The JDA provides that it shall be governed by and construed under the laws of the United States and the State of California. The JDA further provides that any lawsuits to enforce it or that relate to it shall be brought only in courts in the State of California.

COUNT I: INFRINGEMENT OF THE '329 PATENT

(Against Giant-USA and Giant-Taiwan)

48. dw-link restates and realleges the allegations contained in paragraphs 1 through 47 and incorporates them by reference.

49. All bicycles manufactured, sold, shipped, exported to, and imported into the United States by Giant-Taiwan and Giant-USA, which include or utilize the "Maestro" bicycle suspension system, infringe at least claim 1 of the '329 Patent.

50. Giant-USA and Giant-Taiwan have each acted with knowledge of their infringement of the '329 Patent and acted with willful blindness to such acts of infringement.

51. Giant-USA has thereby infringed at least claim 1 of the '329 Patent in violation of 35 U.S.C. § 271(a), (b), and (c).

52. Giant-Taiwan has thereby infringed at least claim 1 of the '329 Patent in violation of 35 U.S.C. § 271(a), (b), and (c).

53. Giant-USA and Giant-Taiwan each have acted despite an objectively high likelihood that their actions constituted infringement of a valid patent and this objectively defined risk was either known or so obvious that it should have been known to Giant-USA and Giant-Taiwan.

54. Giant-USA and Giant-Taiwan, and each of them, therefore have willfully infringed the '329 Patent. dw-link therefore is entitled to an award of enhanced damages and attorneys' fees pursuant to 35 U.S.C. §§ 284 and 285.

55. As a result of Giant-USA's and Giant-Taiwan's infringement of the '329 Patent, dw-link has suffered losses and damages, the precise amounts to be determined at trial. dw-link is entitled to an award of damages, but not less than a reasonable royalty, to adequately compensate it for Giant-USA and Giant-Taiwan's infringement of the '329 Patent, plus pre- and post-judgment interest.

56. dw-link further is entitled to entry of an injunction enjoining Giant-USA and Giant-Taiwan from continuing to infringe the '329 Patent, in accordance with 35 U.S.C. § 283.

COUNT II: INFRINGEMENT OF THE '314 PATENT

(Against Giant-USA and Giant-Taiwan)

57. dw-link restates and realleges the allegations contained in paragraphs 1 through 56 and incorporates them by reference.

58. All bicycles manufactured, sold, shipped, exported to, and imported into the United States by Giant-Taiwan and Giant-USA, which include or utilize the "Maestro" bicycle suspension system, infringe at least claim 1 of the '314 Patent.

59. Giant-USA and Giant-Taiwan have each acted with knowledge of their infringement of the '314 Patent and acted with willful blindness to such acts of infringement.

60. Giant-USA has thereby infringed at least claim 1 of the '314 Patent in

violation of 35 U.S.C. § 271(a), (b), and (c).

61. Giant-Taiwan has thereby infringed at least claim 1 of the '314 Patent in violation of 35 U.S.C. § 271(a), (b), and (c).

62. Giant-USA and Giant-Taiwan each have acted despite an objectively high likelihood that their actions constituted infringement of a valid patent and this objectively defined risk was either known or so obvious that it should have been known to Giant-USA and Giant-Taiwan.

63. Giant-USA and Giant-Taiwan, and each of them, therefore have willfully infringed the '314 Patent. dw-link therefore is entitled to an award of enhanced damages and attorneys' fees pursuant to 35 U.S.C. §§ 284 and 285.

64. As a result of Giant-USA's and Giant-Taiwan's infringement of the '314 Patent, dw-link has suffered losses and damages, the precise amounts to be determined at trial. dw-link is entitled to an award of damages, but not less than a reasonable royalty, to adequately compensate it for Giant-USA and Giant-Taiwan's infringement of the '314 Patent, plus pre- and post-judgment interest.

65. dw-link further is entitled to entry of an injunction enjoining Giant-USA and Giant-Taiwan from continuing to infringe the '314 Patent, in accordance with 35 U.S.C. § 283.

<u>COUNT III: BREACH OF CONTRACT</u>
<u>(Against Giant-Taiwan)</u>

66. dw-link restates and realleges the allegations contained in paragraphs 1 through 65 and incorporates them by reference.

67. Giant-Taiwan has breached the JDA, including, but not limited to, Article 7, sections 7.1 and 7.2(e).

68. Giant-Taiwan has breached its duty of good faith and fair dealing under the JDA.

69. As a result of Giant-Taiwan's breaches of the JDA, dw-link has suffered losses and damages of at least US$580,000.00, the precise amount to be

determined at trial.

70. dw-link therefore is entitled to an award of damages against Giant-Taiwan in the amount of at least US$580,000.00, plus pre- and post-judgment interest, the precise amounts to be determined at trial.

71. In addition, dw-link is entitled to an injunction enjoining Giant-Taiwan from violating its non-disclosure and restrictions of use confidential information obligations under the terms of the JDA. dw-link is further entitled to an injunction enjoining Giant-Taiwan from any use of the "G+ Technology" designed and developed by dw-link pursuant to the terms of the JDA.

COUNT IV: UNJUST ENRICHMENT

(Against Giant-Taiwan)

72. dw-link restates and realleges the allegations contained in paragraphs 1 through 71 and incorporates them by reference.

73. Giant-Taiwan has benefited by the design and development work performed by dw-link pursuant to the terms of the JDA and for which Giant-Taiwan has not compensated dw-link

74. Giant-Taiwan thereby has been unjustly enriched as a result of receiving and accepting the valuable design and development work of dw-link in an amount of at least US$400,000.00, the precise amount to be determined at trial.

75. dw-link therefore is entitled to an award of damages against Giant-Taiwan in the amount of at least US$400,000.00, plus pre- and post-judgment interests, the precise amounts to be determined at trial.

DEMAND FOR JURY TRIAL

dw-link hereby demands a trial by jury on all issues triable to a jury.

PRAYER FOR RELIEF

WHEREFORE, Plaintiff dw-link Incorporated prays for entry of judgment against Defendants Giant Bicycle, Inc. and Giant Manufacturing Co., Ltd. as follows:

1. On Count I, awarding damages against Giant-USA and Giant-Taiwan for infringement of the '329 Patent in an amount to be determined at trial, but in no event less than a reasonable royalty.

2. On Count II, awarding damages against Giant-USA and Giant-Taiwan for infringement of the '314 Patent in an amount to be determined at trial, but in no event less than a reasonable royalty.

3. On Counts I and II, awarding increased damages pursuant to 35 U.S.C. § 284 for Giant-USA and Giant-Taiwan's willful infringement of the '329 and/or '314 Patents.

4. On Counts I and II, finding this case to be exceptional and awarding dw-link its reasonable attorneys' fees and costs against Giant-USA and Giant-Taiwan pursuant to 35 U.S.C. § 285.

5. On Counts I and II, for an injunction against Giant-USA and Giant-Taiwan enjoining them, their officers, directors, managing agents, attorneys, and all persons in active concert with them, from any and all acts of infringement of the '329 and '314 Patents.

6. On Count III, awarding damages against Giant-Taiwan for breach of contract in an amount of at least US$580,000.00, the precise amount to be determined at trial.

7. On Count III, awarding dw-link its reasonable attorneys' fees and costs incurred in accordance with the terms of the JDA.

8. On Count III, for an injunction against Giant-Taiwan, enjoining it, its officers, directors, managing agents, attorneys, and all persons in active concert with them, from violating its non-disclosure and confidentiality obligations under the JDA and further enjoining it from any use of the "G+ Technology" designed and developed by dw-link pursuant to the terms of the JDA.

9. On Count IV, awarding damages against Giant-Taiwan for unjust enrichment in an amount of at least US$400,000.00, the precise amount to be determined at trial.

10. Awarding dw-link pre- and post-judgment interest on all sums awarded.

11. Awarding dw-link its costs incurred in this action.

12. Awarding dw-link such other and further relief as the Court may deem just and equitable.

Dated: February 5, 2013

Respectfully submitted,

SNR DENTON US LLP

By Arthur Beeman
Arthur Beeman

Attorneys for Plaintiff
Dw-link Incorporated

Exhibit A

US007048292B2

(12) **United States Patent**
Weagle

(10) **Patent No.: US 7,048,292 B2**
(45) **Date of Patent: May 23, 2006**

(54) **BICYCLE SUSPENSION SYSTEMS**

(76) Inventor: **David Weagle**, P.O. Box 1184, Edgartown, MA (US) 02539

(*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 0 days.

(21) Appl. No.: **10/669,412**

(22) Filed: **Sep. 25, 2003**

(65) **Prior Publication Data**

US 2005/0067810 A1 Mar. 31, 2005

(51) **Int. Cl.**
B62K 25/28 (2006.01)

(52) **U.S. Cl.** **280/285**; 280/284

(58) **Field of Classification Search** 280/284, 280/285

See application file for complete search history.

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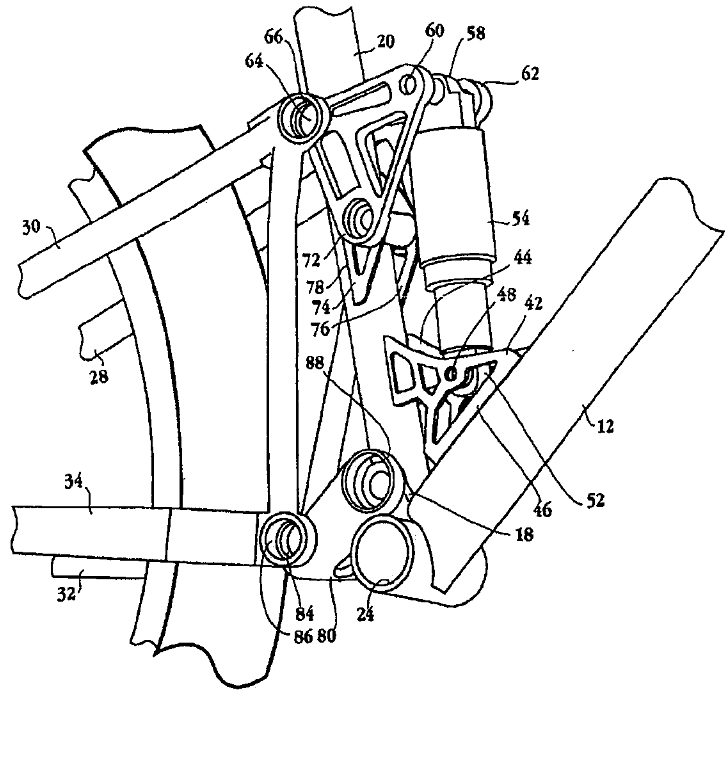
Primary Examiner—Lesley D. Morris
Assistant Examiner—Matthew Luby
(74) *Attorney, Agent, or Firm*—Perkins Coie LLP

(57) **ABSTRACT**

A bicycle rear wheel suspension system in which plural interconnections are provided of rear wheel-supporting components, at which interconnections there occurs pivotal traverses contributing to urging the axle of the rear wheel along a path providing positions of movement therealong to achieve a desired extent of pressure feedback to the pedals, an easing of suspension reaction to bumps, and multiple chainstay lengths, all parameters to better suit the bicycle to its end uses and the terrain on which it is used.

8 Claims, 3 Drawing Sheets



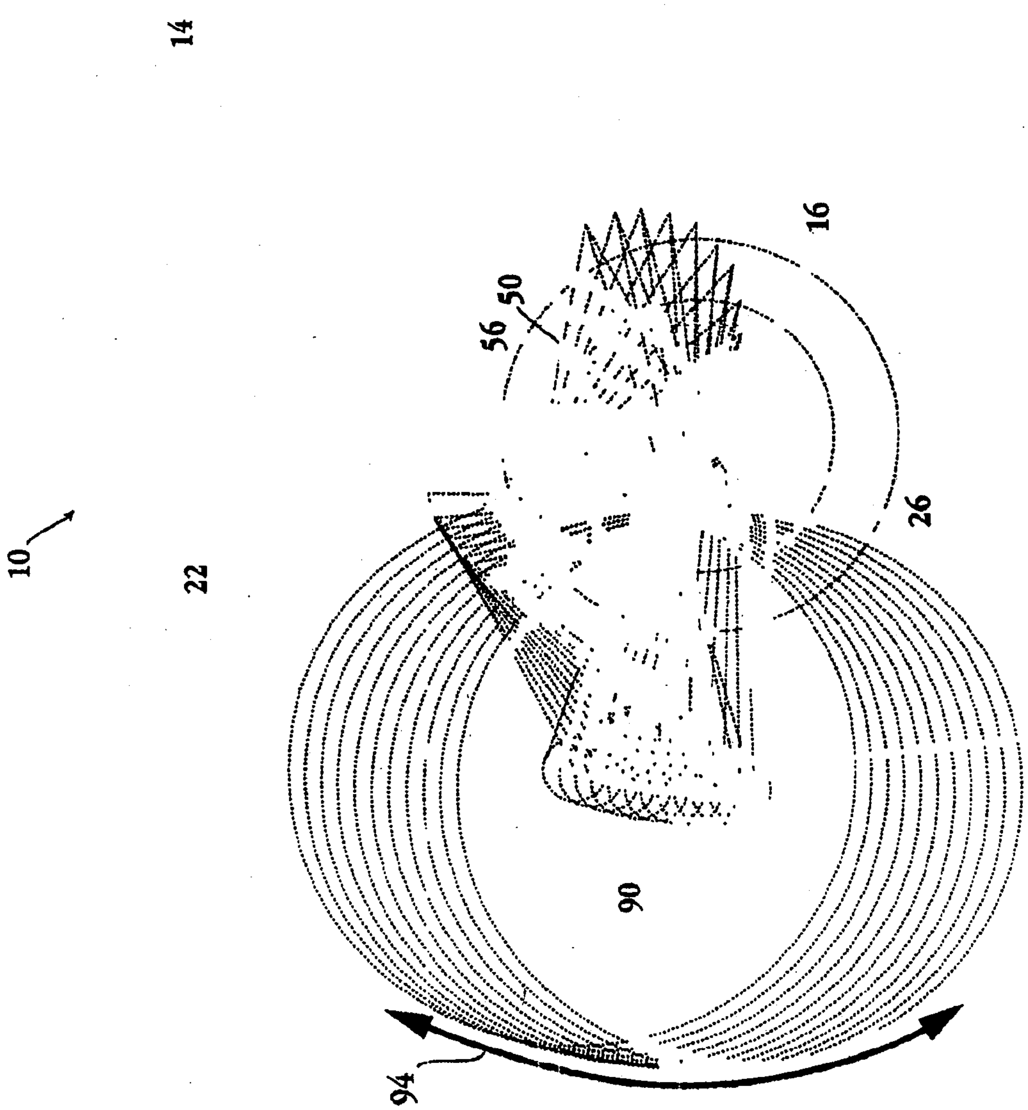


Fig. 1

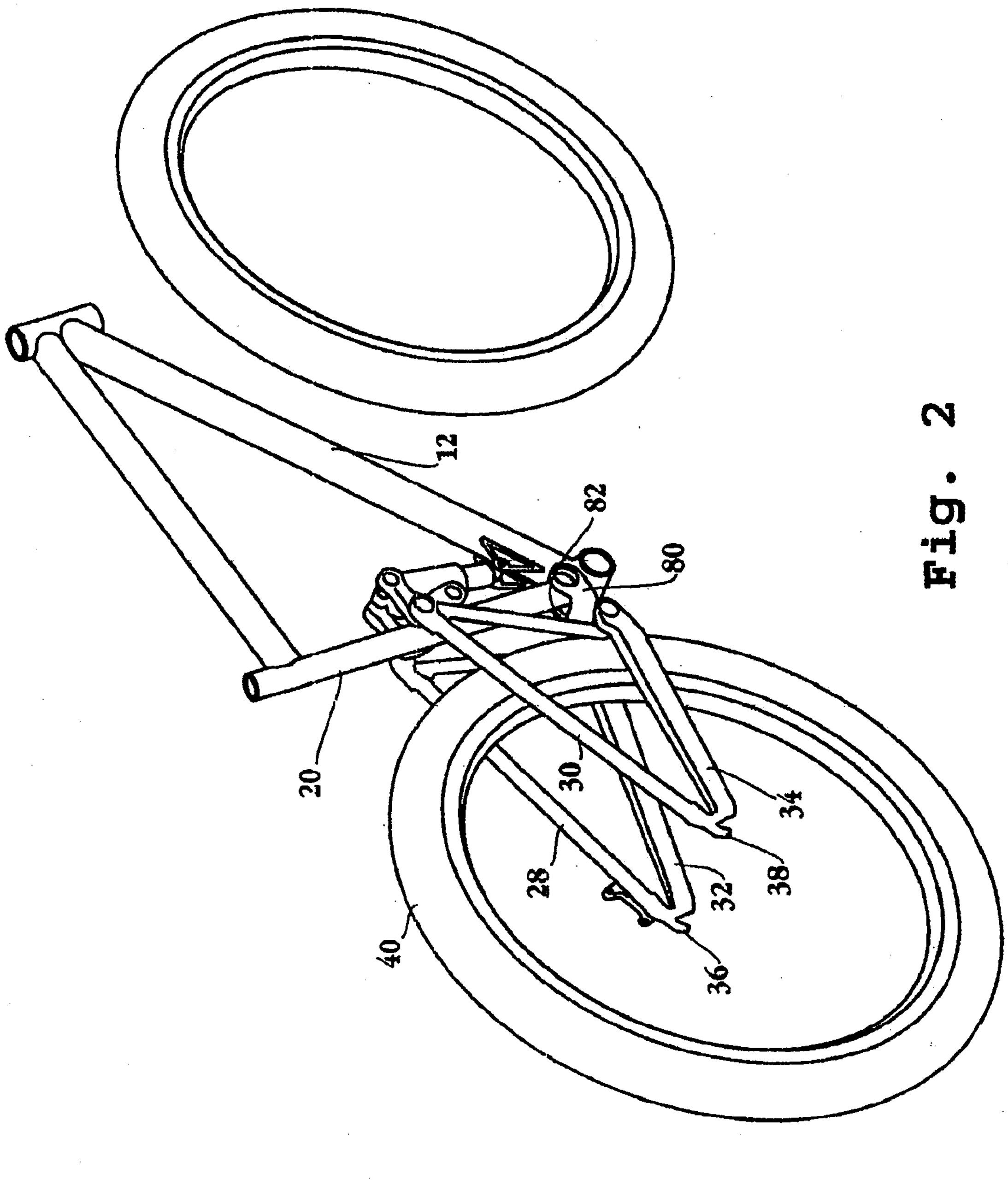


Fig. 2

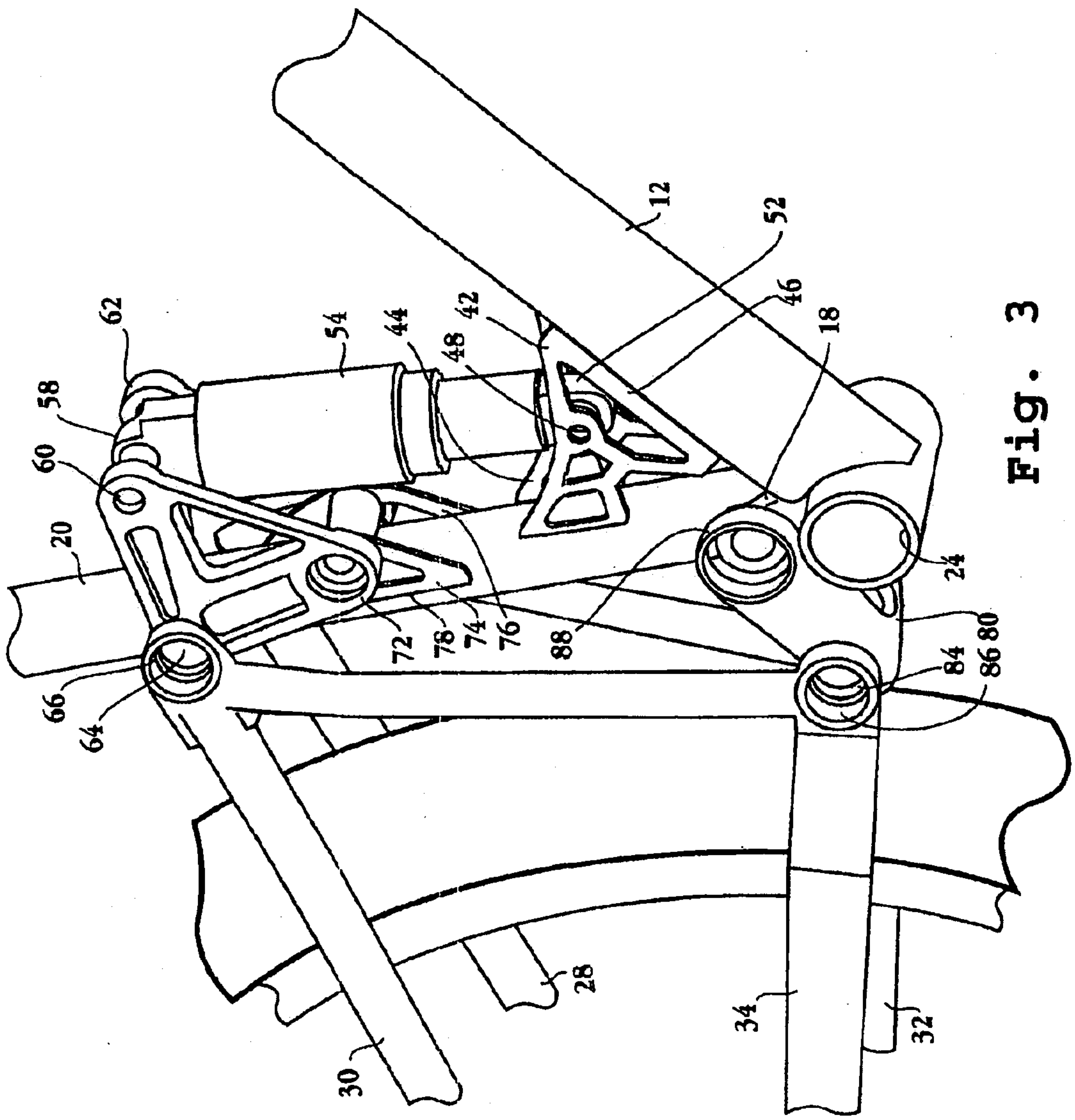

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Fig. 3

BICYCLE SUSPENSION SYSTEMS

The present invention relates generally to improvements for bicycles, the improvements more particularly residing in a link suspension system that can more effectively be tuned to balance forces in the rear suspension of the bicycle, all as will be better understood as the description proceeds.

EXAMPLE OF THE PRIOR ART

Take into consideration the system described by U.S. Pat. No. 6,206,397 B1. This link suspension system claims linkage arrangement and a defined range of rear wheel axle paths for a suspension bicycle. The axle path claimed and shown in the patent art can be manipulated into an S shape, or a converging C shape. The theory behind this is that during the lower part of the suspension travel, the wheel axle will travel at an increasing rate, away from the bottom bracket center. By achieving this, the designers hope to increase the resistance to rear suspension compression during the beginning of the travel. This resistance to suspension compression is called anti-squat in popular engineering text. As the '397 patent is examined, it becomes obvious that the inventors overlooked several key factors that must be evaluated in order to obtain a clear understanding of anti-squat and how it pertains to a suspension system. The system as described in the '397 patent feature pro-squat in the beginning of the suspension travel, and a rising rate of anti-squat as the suspension cycles through the end of its travel. In practice bicycles designed using the system described in the '397 patent feature inefficient acceleration in the beginning of the suspension travel, where efficient acceleration is needed most.

As background to understanding the present invention it is to be noted that a link bicycle suspension system is a defined specific range of kinematical linkages which can be used to produce a tactical rear axle path. Each combination of linkages can be tuned to balance forces in the rear suspension of the bicycle in ways that no previous system has been able to. Variations of the linkage layout can shift the balance of forces to give distinct advantages for suspension systems used for differing applications. The suspension system allows a designer to manipulate the rear axle path in relation to the bicycle frame. Manipulating axle paths has a huge impact on the performance of the rear suspension, since axle path governs several key aspects of suspension performance.

It is an object of the present invention to achieve a desired variable amount of anti-squat as the rear suspension cycles through its travel. Manipulating rear axle path in a tactical manner using a linkage system allows the designer to obtain a desired range of anti-squat curves. A preferred anti-squat curve is one that features a higher amount of anti-squat in the beginning of the suspension travel, and a lesser amount as the suspension cycles compressively through its travel. This anti-squat amount lessens with regard to the amount of spring force provided by a spring damper unit. In addition to this lessening anti-squat amount as the suspension compresses, the linkage arrangement is also designed to impart a minimal amount of feedback to the pedals as the suspension cycles. The preferred linkage arrangement also can be optimized so that a spring damper unit can be driven at a strategic leverage rate, furthermore reducing inefficient rear wheel movement. Also the linkage arrangement can be strategically placed so that the effect of braking force on rear wheel movement is minimized.

The description of the invention which follows, together with the accompanying drawings should not be construed as limiting the invention to the example shown and described, because those skilled in the art to which this invention appertains will be able to devise other forms thereof within the ambit of the appended claims.

FIG. **1** is a diagrammatic view of a mode of adjusting bicycle rear wheel suspension according to the present invention;

FIG. **2** is a partial perspective view of a bicycle component providing the rear wheel suspension; and

FIG. **3** is a view similar to FIG. **2** on an enlarged scale.

A link suspension system according to the present invention is embodied in a bicycle **10** having a body frame member 12 which extends from a handlebar **14** rearwardly downward at an angular orientation to a pedal mechanism **16** and is integral at juncture **18** to a vertically oriented frame member **20** which supports a bicycle seat **22**. At the junction **18**, a cylindrical configuration **24** is provided for journaling in rotation the rotor **26** of the pedal mechanism **16**.

Mounted to extend rearwardly of the frame members **12**, **20** are an upper angularly oriented pair of supports **28** and **30** and lower horizontally pair of supports **32** and **34** which at respective ends **36** and **38** are attached to a rear wheel **40** for rotatably mounting of the rear wheel **40** to the bicycle.

Just above the juncture **18** are spaced apart brackets **42** and **44** welded as at **46** to frame member 12 having aligned openings **48** for receiving therethrough bolt means **50** connecting thereto the bottom end **52** of a housing **54** of an internally mounted damper spring **56**, the upper housing end **58** being connected to a pair of triangular brackets **60** and **62**, in turn connected, as at **64**, to cooperating openings **66** provided in rear wheel supports **32** and **34**, the remaining bracket opening **72** being bolted to support bracket **74** and **76** welded, as at **78**, to the seat support frame **20**.

Completing the link suspension system are brackets **80** and **82** connected at opposite ends **84** to cooperating openings **86** provided in the rear wheel supports **32** and **34** and at ends **88** to the junction **18**.

Referring to the diagrammatic illustration of FIG. **1**, it will be understood that the interconnections at **48**, **72**, **66** enable the interconnected components to partake of a multitude of pivotal traverses, of which a significant pivotal traverse **90** implements axle path changes in a rear sprocket **92** of the rear wheel **40** contributing to a range **94** of rear wheel positions, all to the end of achieving a selected (1) extent of pressure feedback to the pedals, (2) an easing of suspension reaction to bumps, and (3) as known in the parlance of the art, multiple chainstay lengths. Thus, one well versed in the art is able by selection to tune the described combination of linkages to balance forces in the rear suspension of the bicycle as desired and as dictated by the end use of the bicycle and the terrain on which it is used.

While the apparatus herein shown and disclosed in detail is fully capable of attaining the objects and providing the advantages hereinbefore stated, it is to be understood that it is merely illustrative of the presently preferred embodiment of the invention and that no limitations are intended to the detail of construction or design herein shown other than as defined in the appended claims.

What is claimed is:

1. A compressible linkage suspension system for a bicycle rear wheel comprising a plurality of links and pivots isolating the rear wheel from a frame member which comprises a seat and a crank pedal unit, wherein said links partake in pivotal traverses to achieve an anti-squat response, where said anti-squat response is higher in the beginning of the suspension travel, and lesser thereafter.

2. The compressible linkage suspension system according to claim 1, wherein the center of said rear wheel is located below the upper pivot of a damper unit of said suspension system.

3. The compressible linkage suspension system according to claim 1, wherein said frame member is located behind a damper unit of said suspension system.

4. The compressible linkage suspension system according to claim 1, wherein said the tube of said seat is located behind a damper unit of said suspension system.

5. The compressible linkage suspension system according to claim 1, wherein the center of said crank pedal unit is located below the upper pivot of a damper unit of said suspension system.

6. The compressible linkage suspension system according to claim 1, wherein said pivotal traverses facilitate rear wheel suspension while maintaining a low squat response.

7. The compressible linkage suspension system according to claim 1, wherein said links are located on the side of said frame member.

8. The compressible linkage suspension system according to claim 1, wherein said upper link pivots are located above said crank pedal unit.

* * * * *

Exhibit B

US007128329B2

(12) **United States Patent**
Weagle

(10) **Patent No.: US 7,128,329 B2**
(45) **Date of Patent: Oct. 31, 2006**

(54) **VEHICLE SUSPENSION SYSTEMS**

(76) Inventor: **David Weagle**, 11 Boldt Farms Rd., Edgartown, MA (US) 02539

(*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 3 days.

(21) Appl. No.: **10/949,264**

(22) Filed: **Sep. 24, 2004**

(65) **Prior Publication Data**

US 2005/0067806 A1 Mar. 31, 2005

Related U.S. Application Data

(63) Continuation-in-part of application No. 10/669,412, filed on Sep. 25, 2003, now Pat. No. 7,048,292.

(51) **Int. Cl.**
B62K 25/00 (2006.01)

(52) **U.S. Cl.** **280/284**

(58) **Field of Classification Search** 280/283–286, 280/275
See application file for complete search history.

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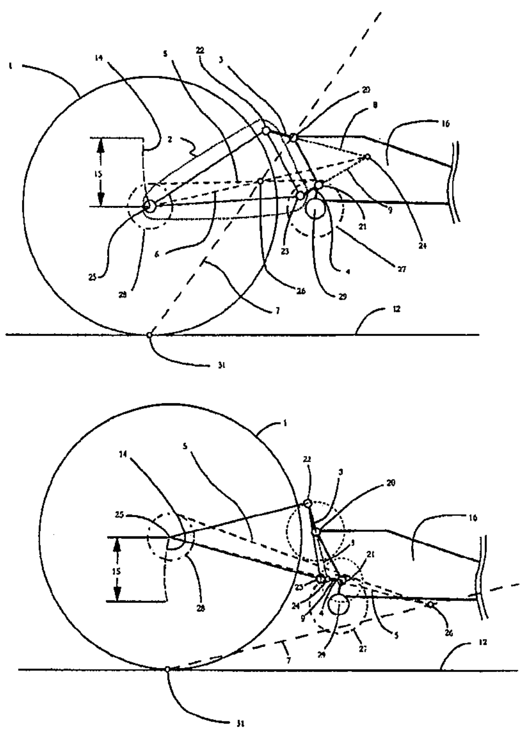
Assistant Examiner—L. Lum
(74) *Attorney, Agent, or Firm*—Stahl Law Firm

(57) **ABSTRACT**

A wheel suspension system having under powered acceleration a squat response that begins in the realm of anti squat and passes through a point of lessened anti squat at a further point in the travel.

9 Claims, 19 Drawing Sheets



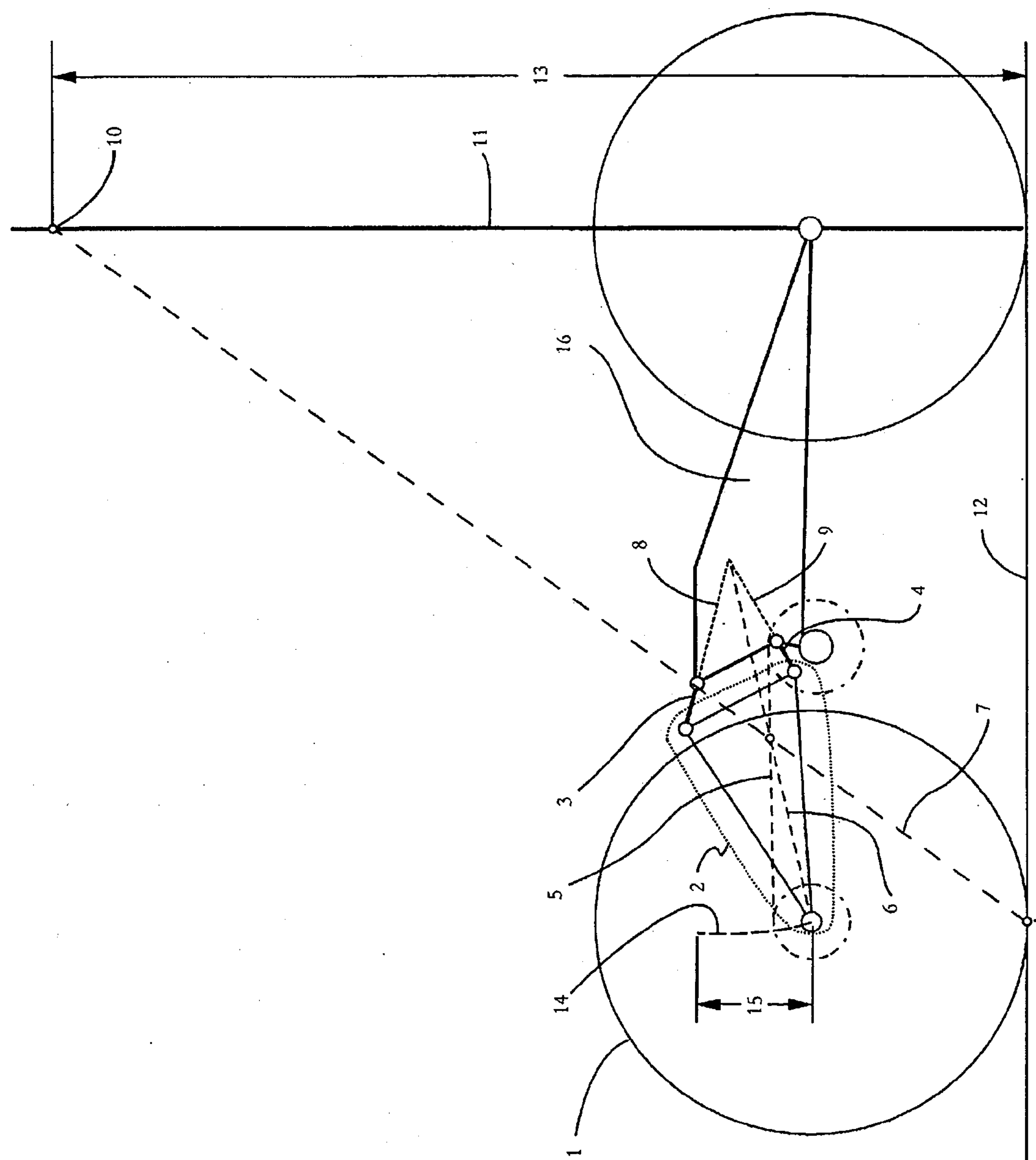


FIGURE 1a

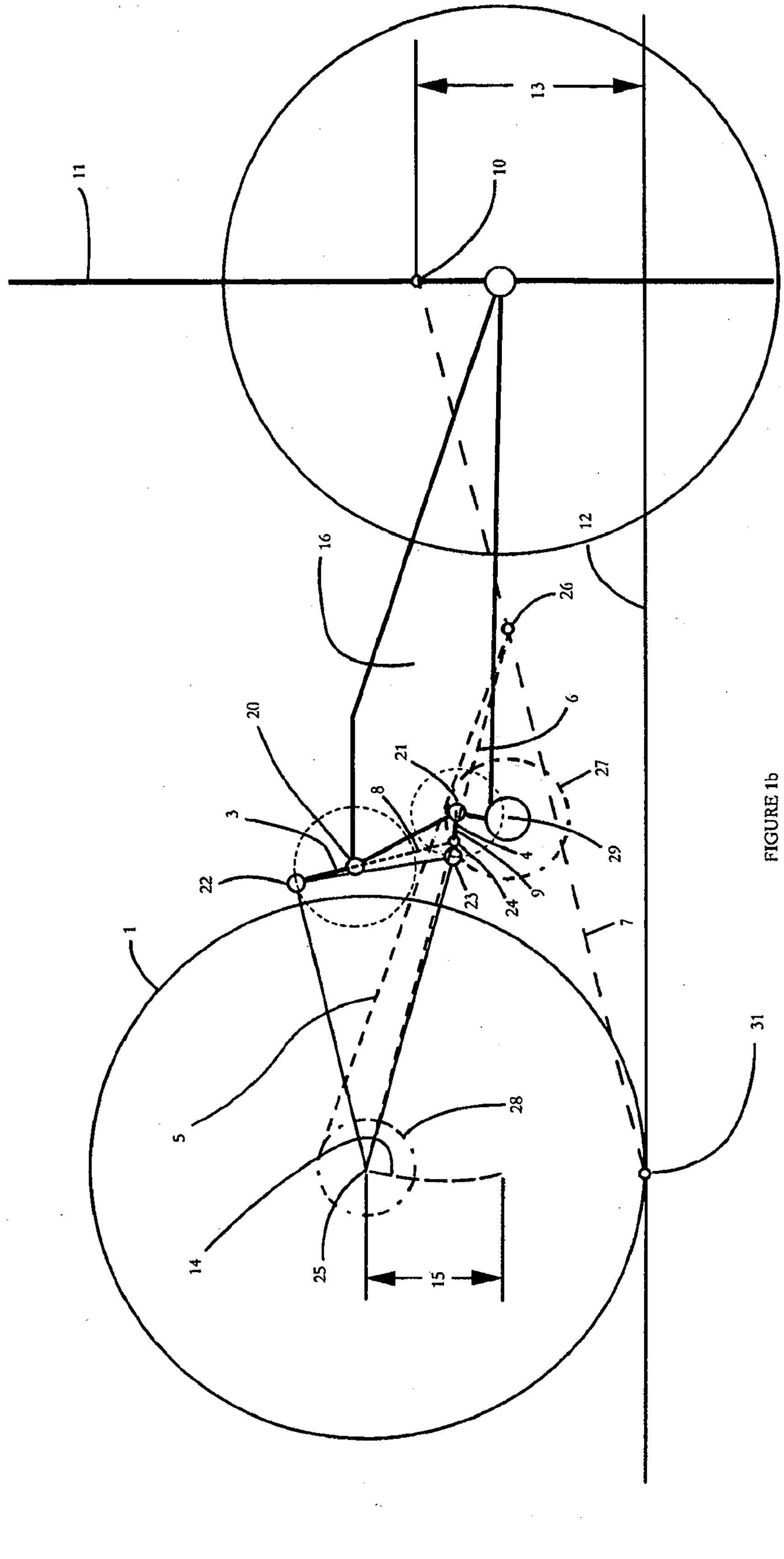


FIGURE 1b

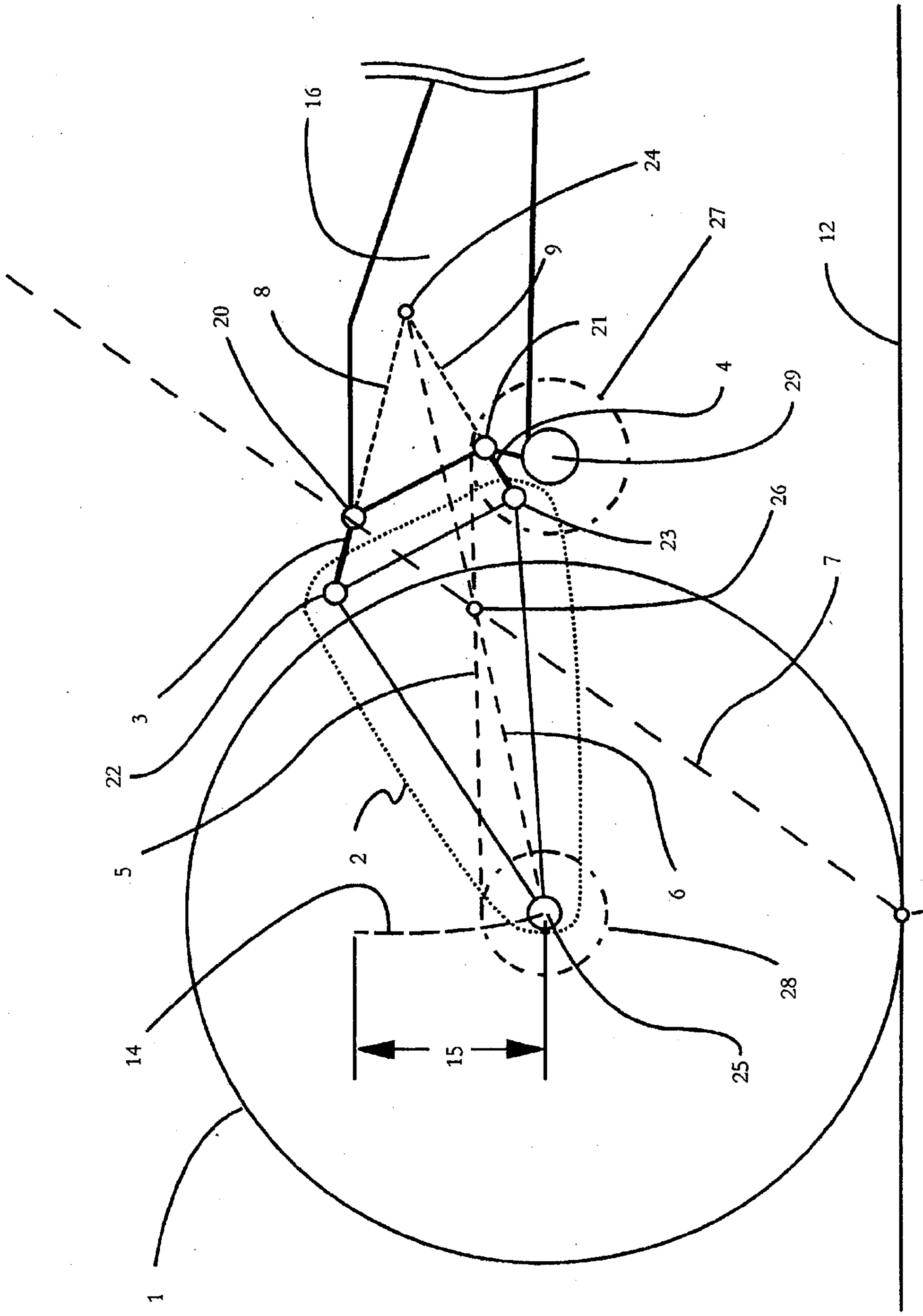


FIGURE 1c

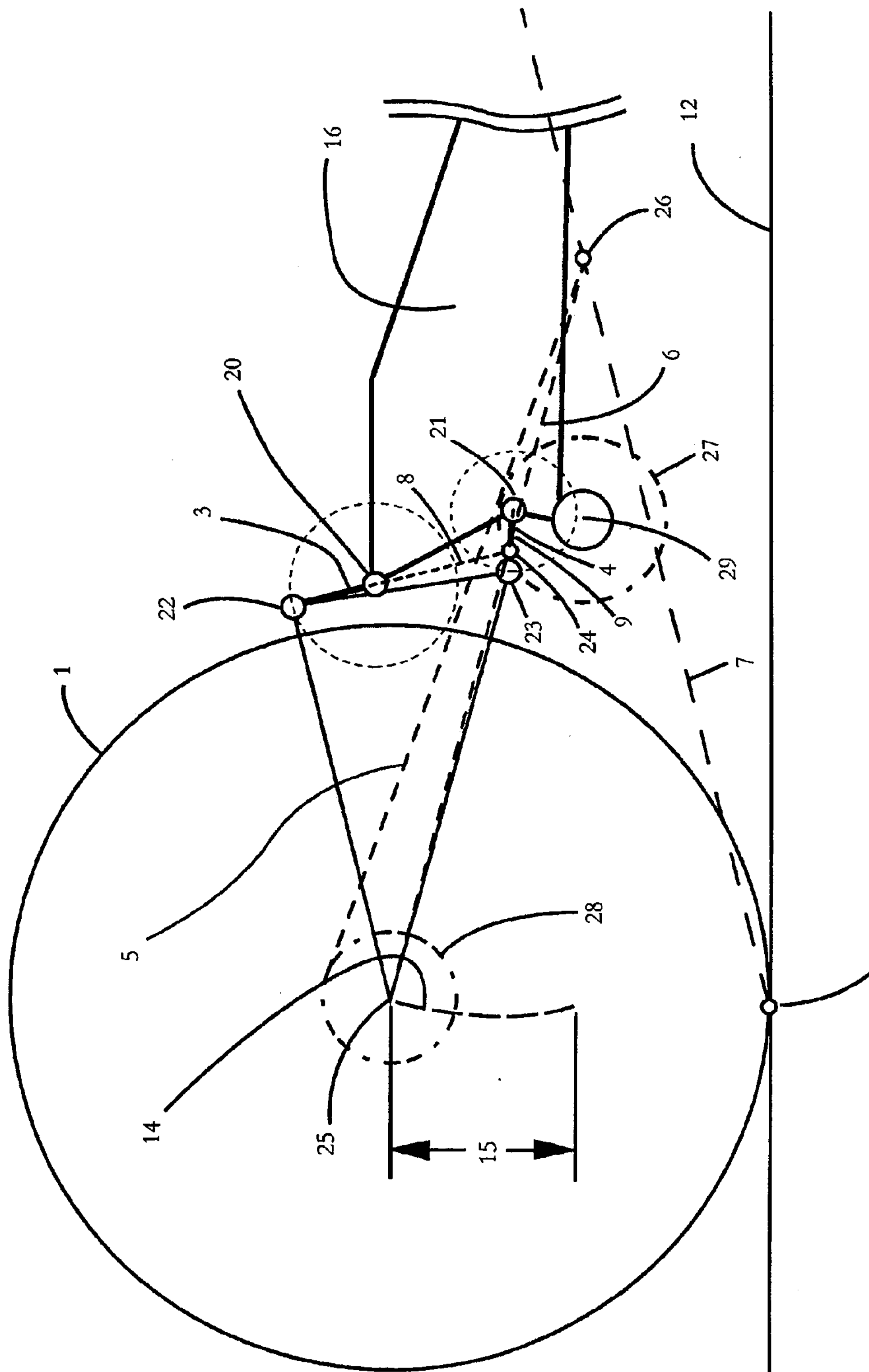


FIGURE 1d

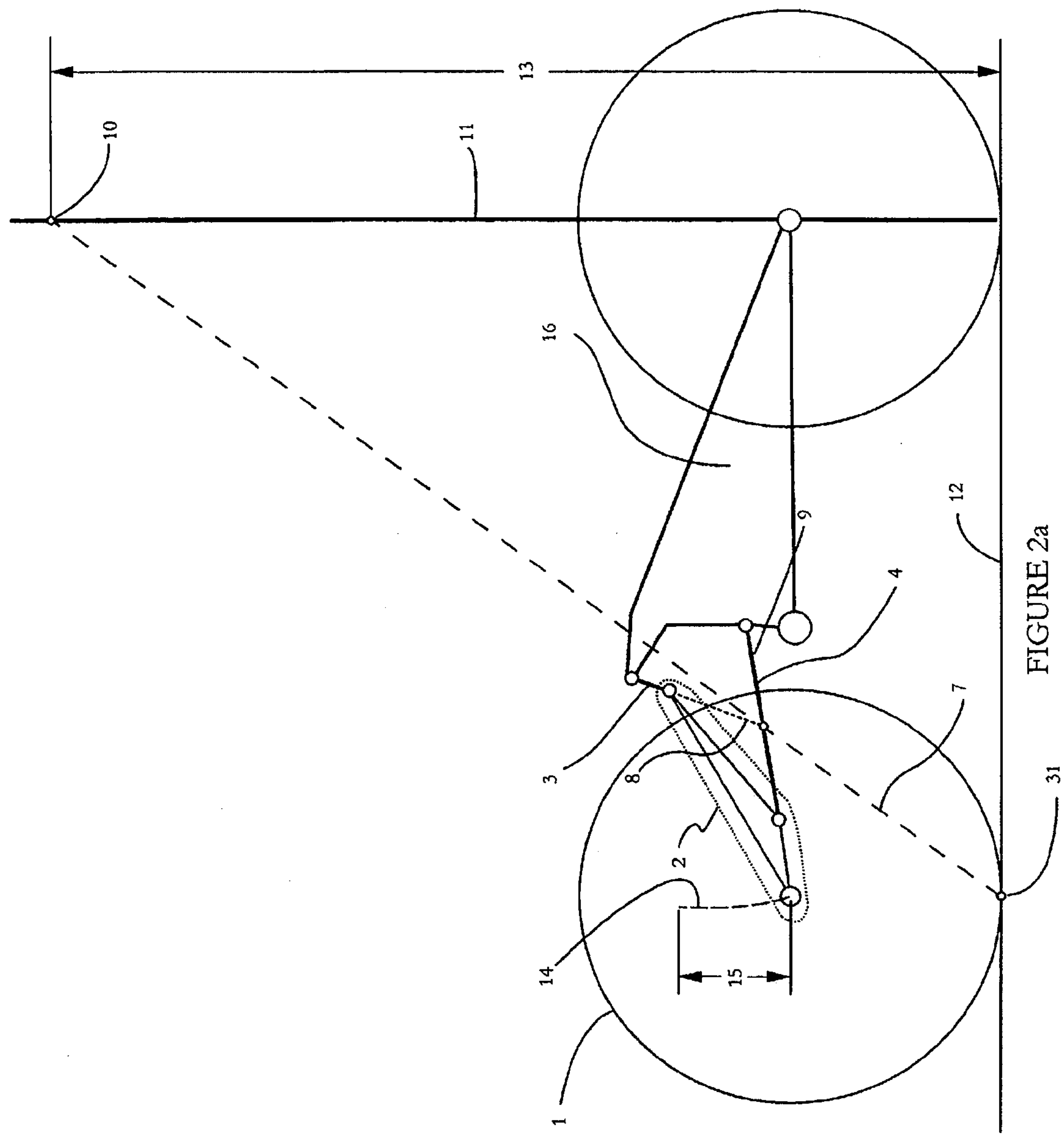


FIGURE 2a

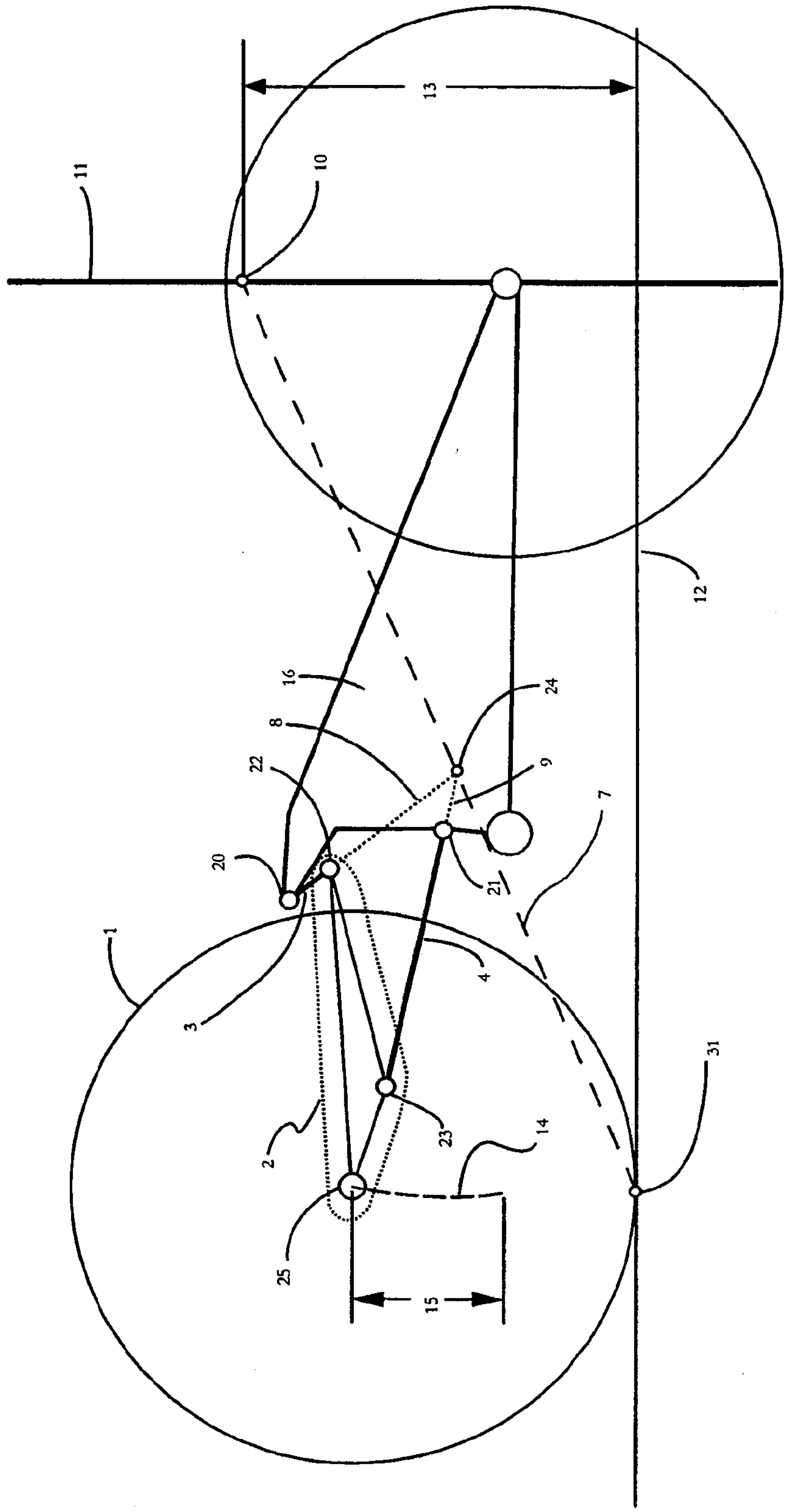


FIGURE 2b

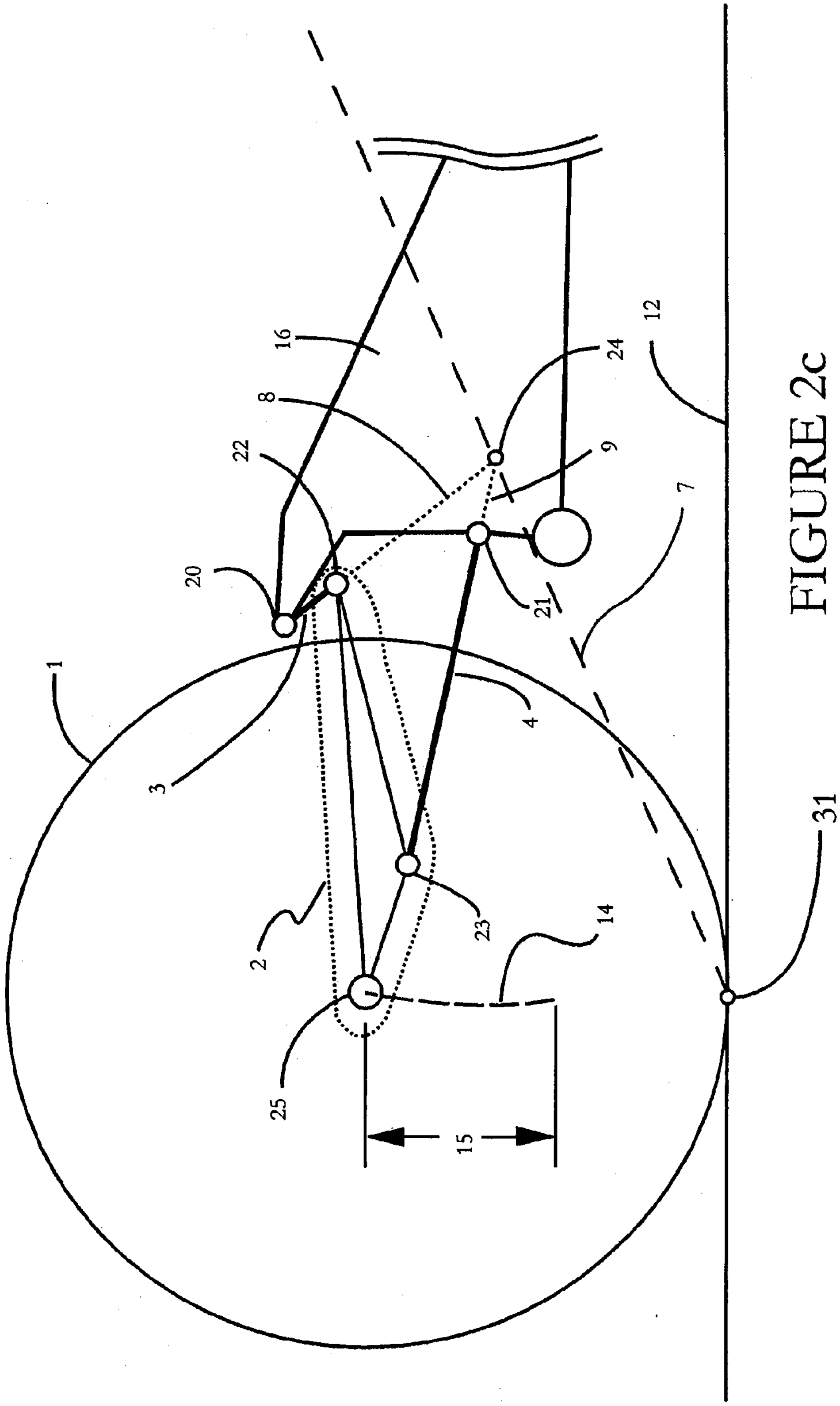


FIGURE 2c

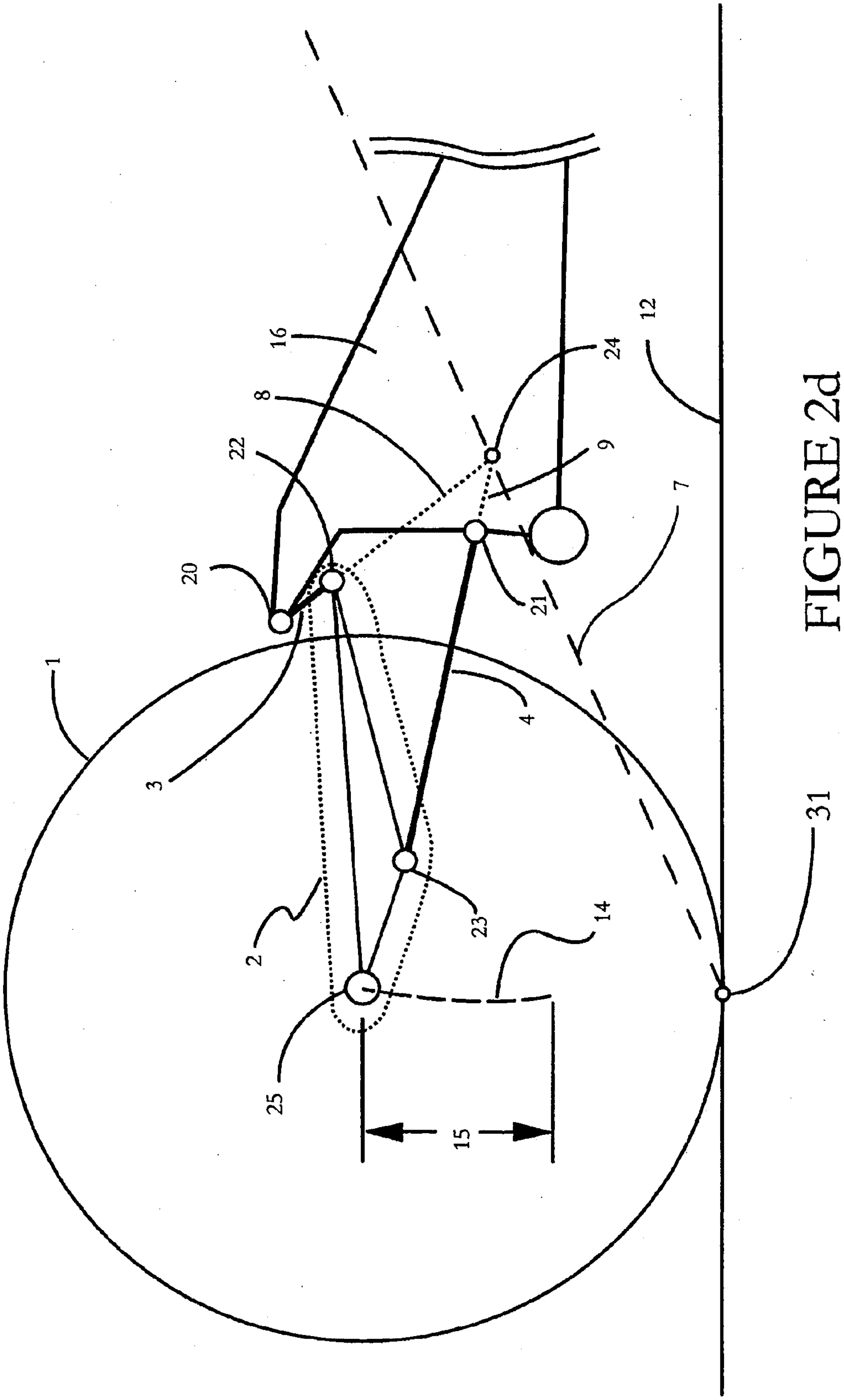
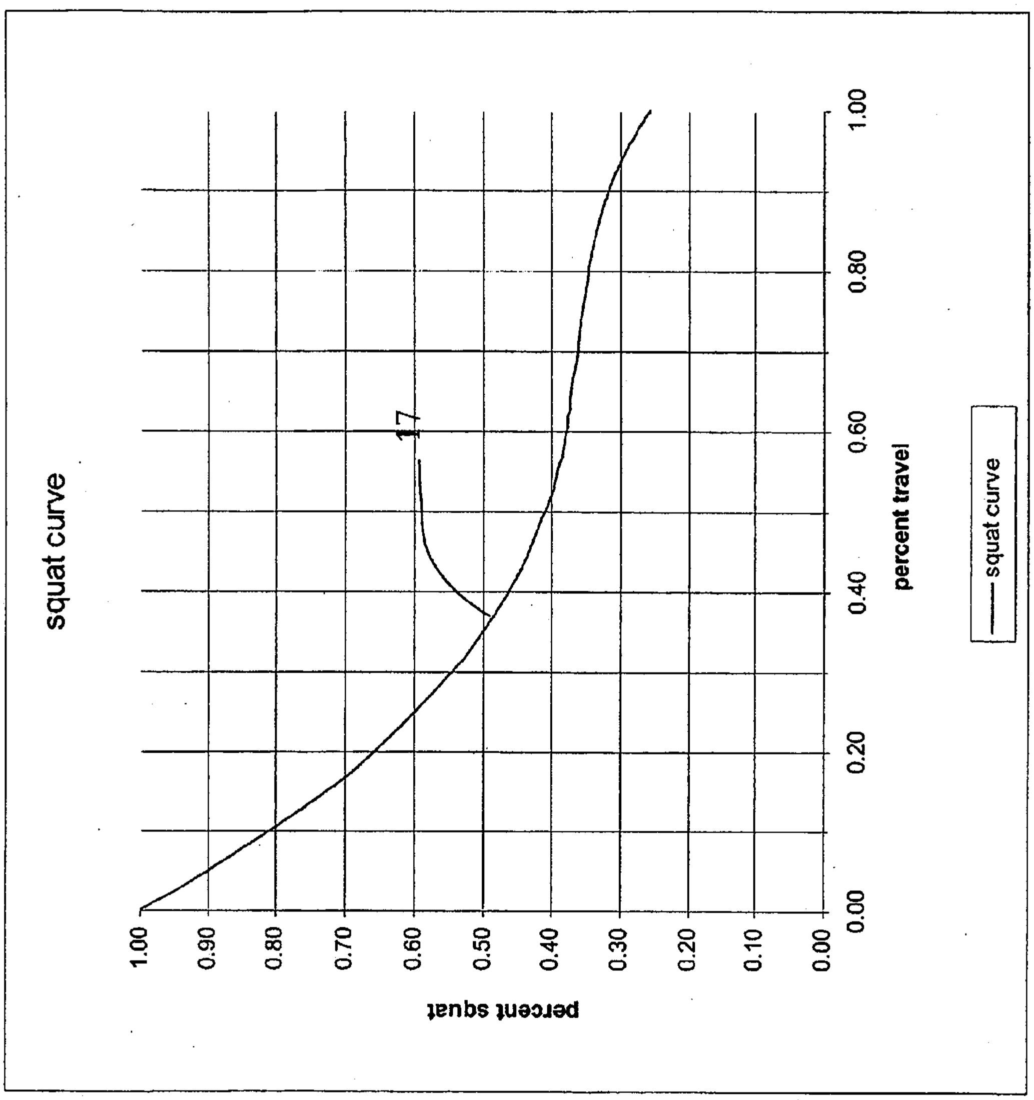


FIGURE 2d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0.90
0.80
0.70
0.60
0.50
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percent travel
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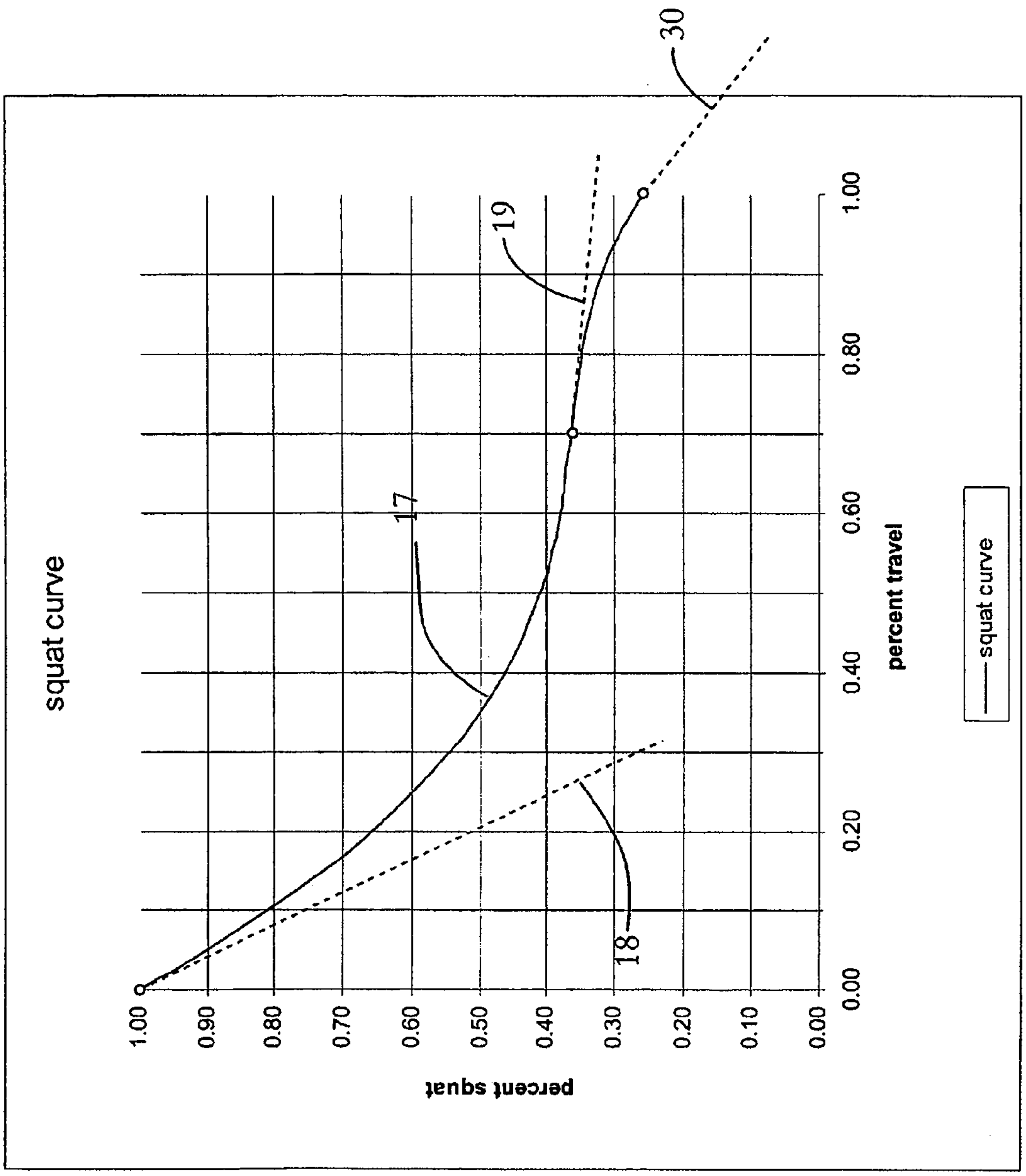
0.80
1.00
17
squat curve

FIGURE 3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percent travel
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squat curve
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FIGURE 4

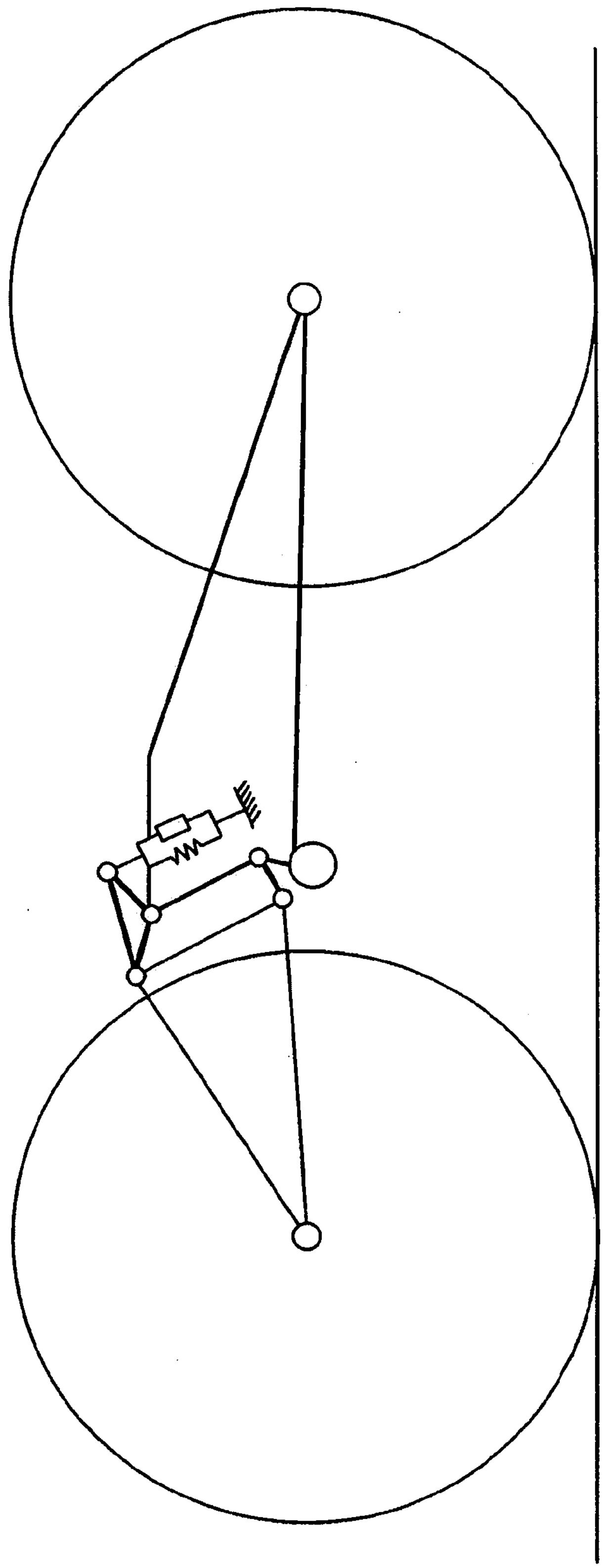

FIGURE 5

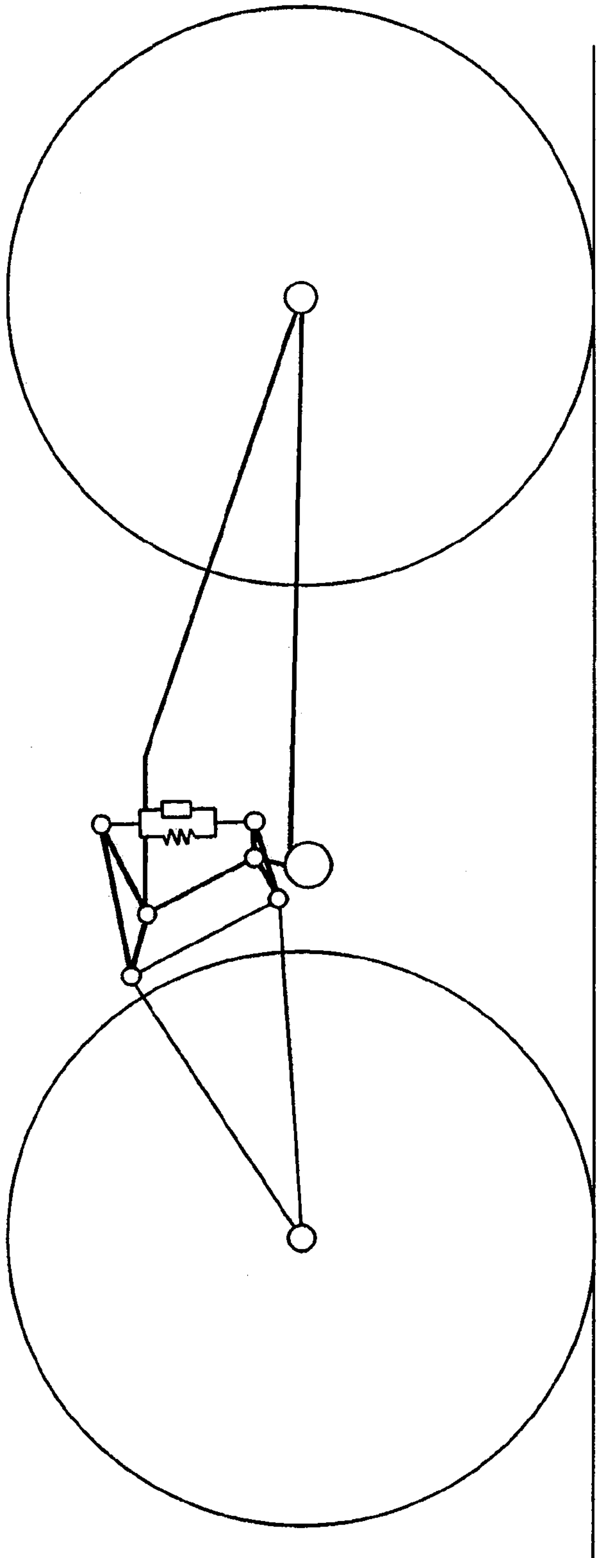

FIGURE 6

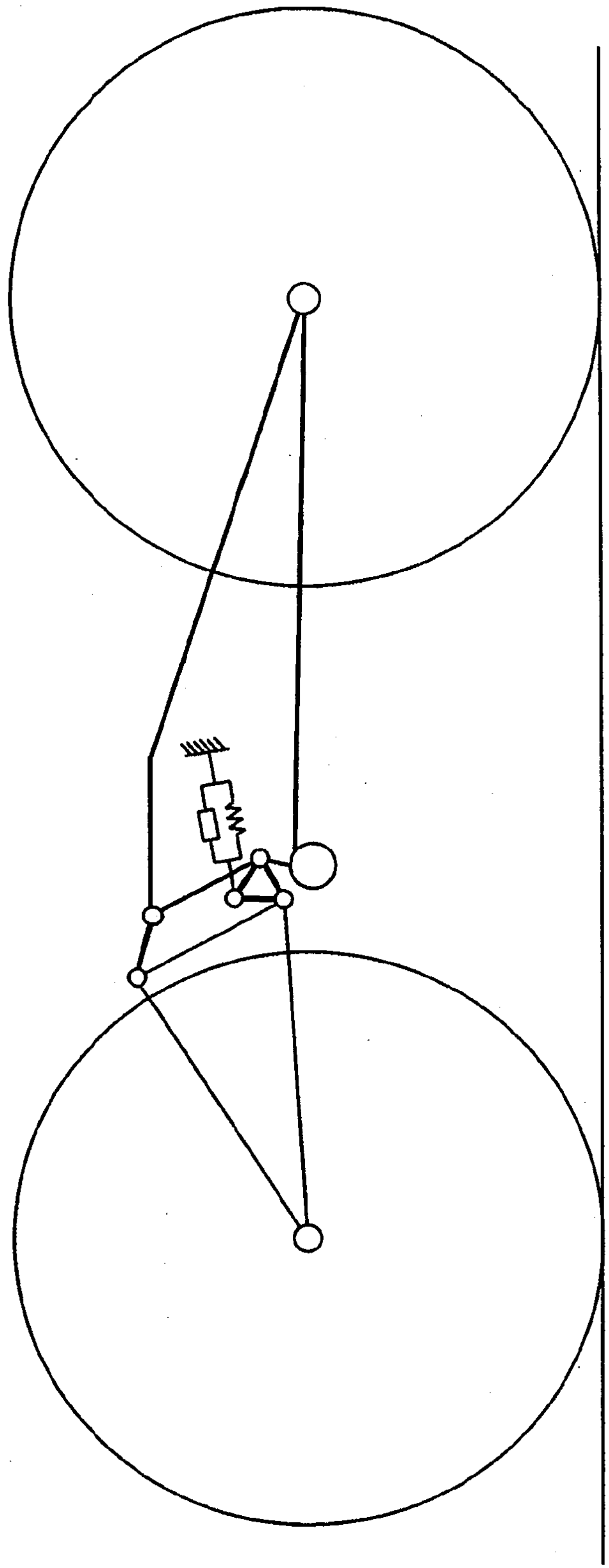

FIGURE 7

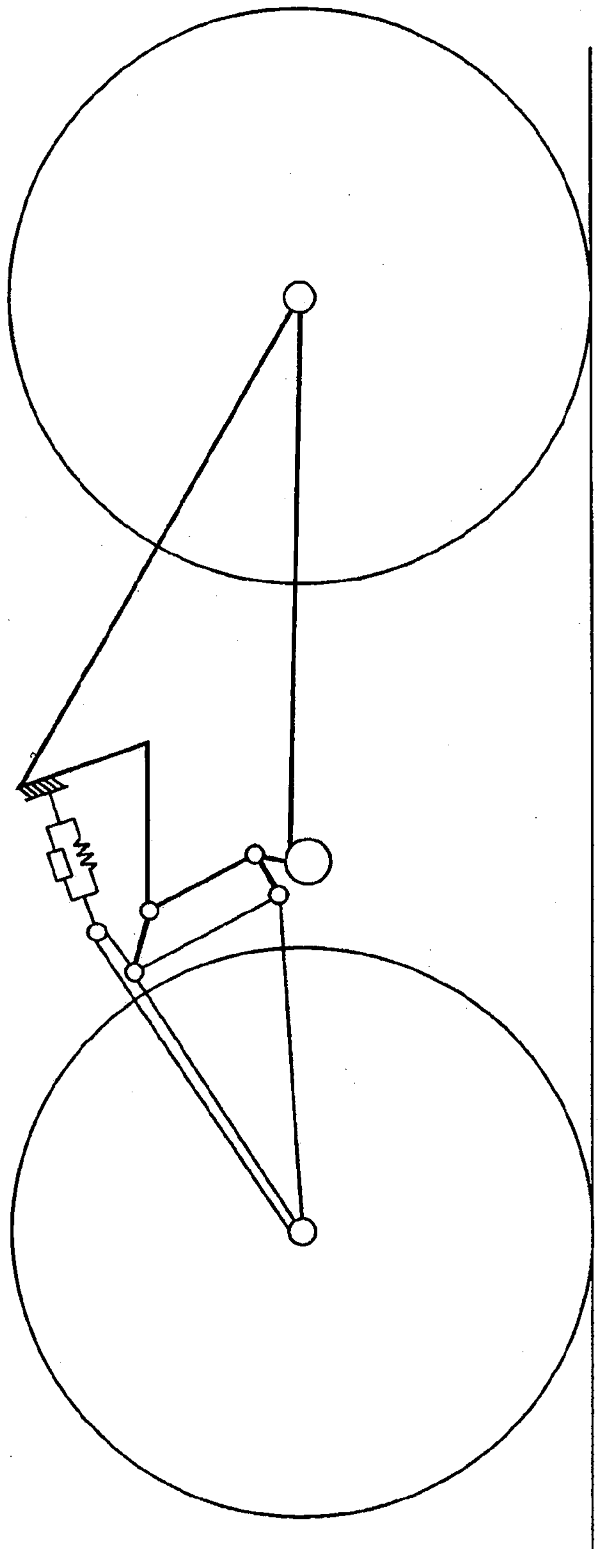

FIGURE 8

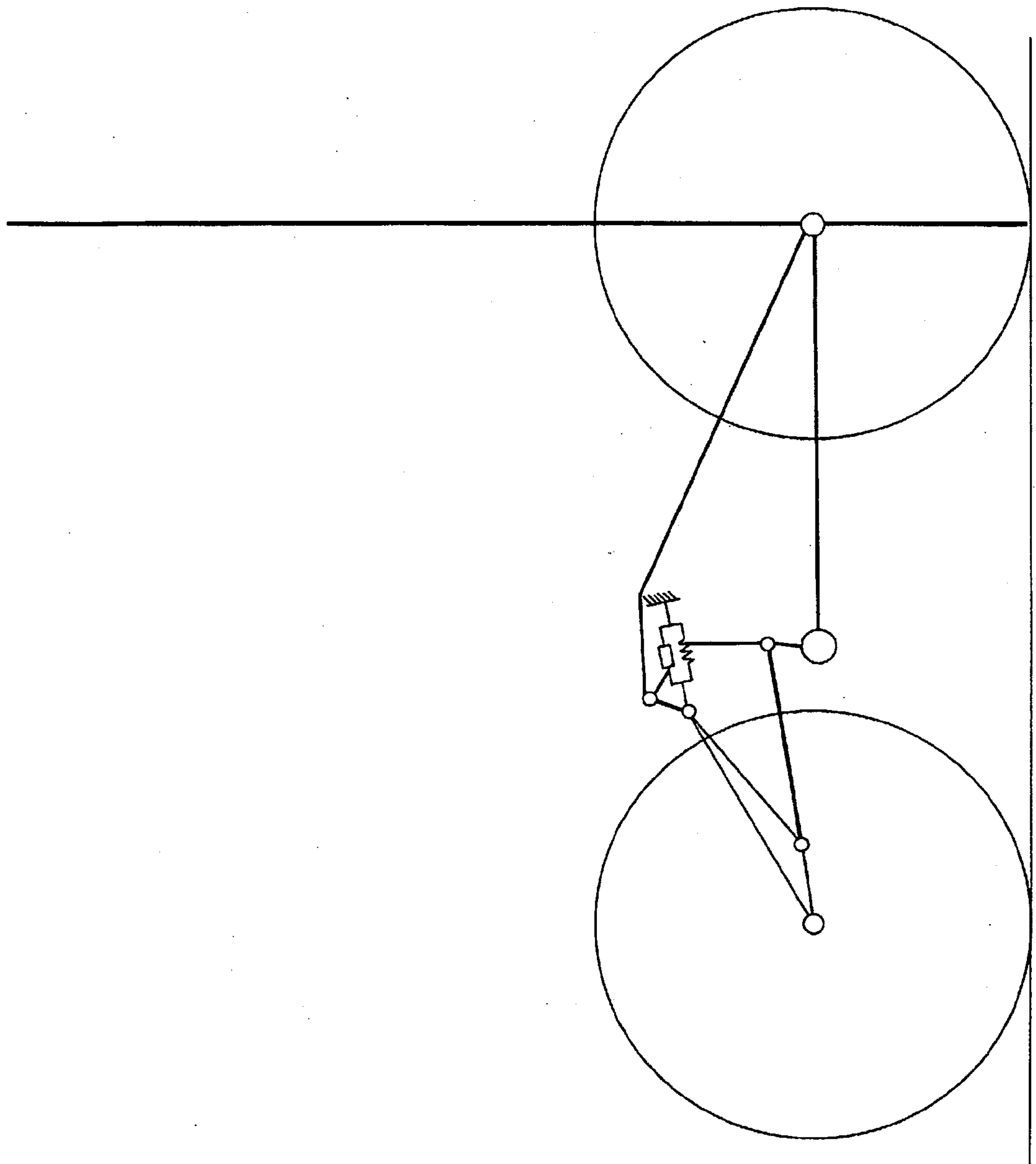

FIGURE 9

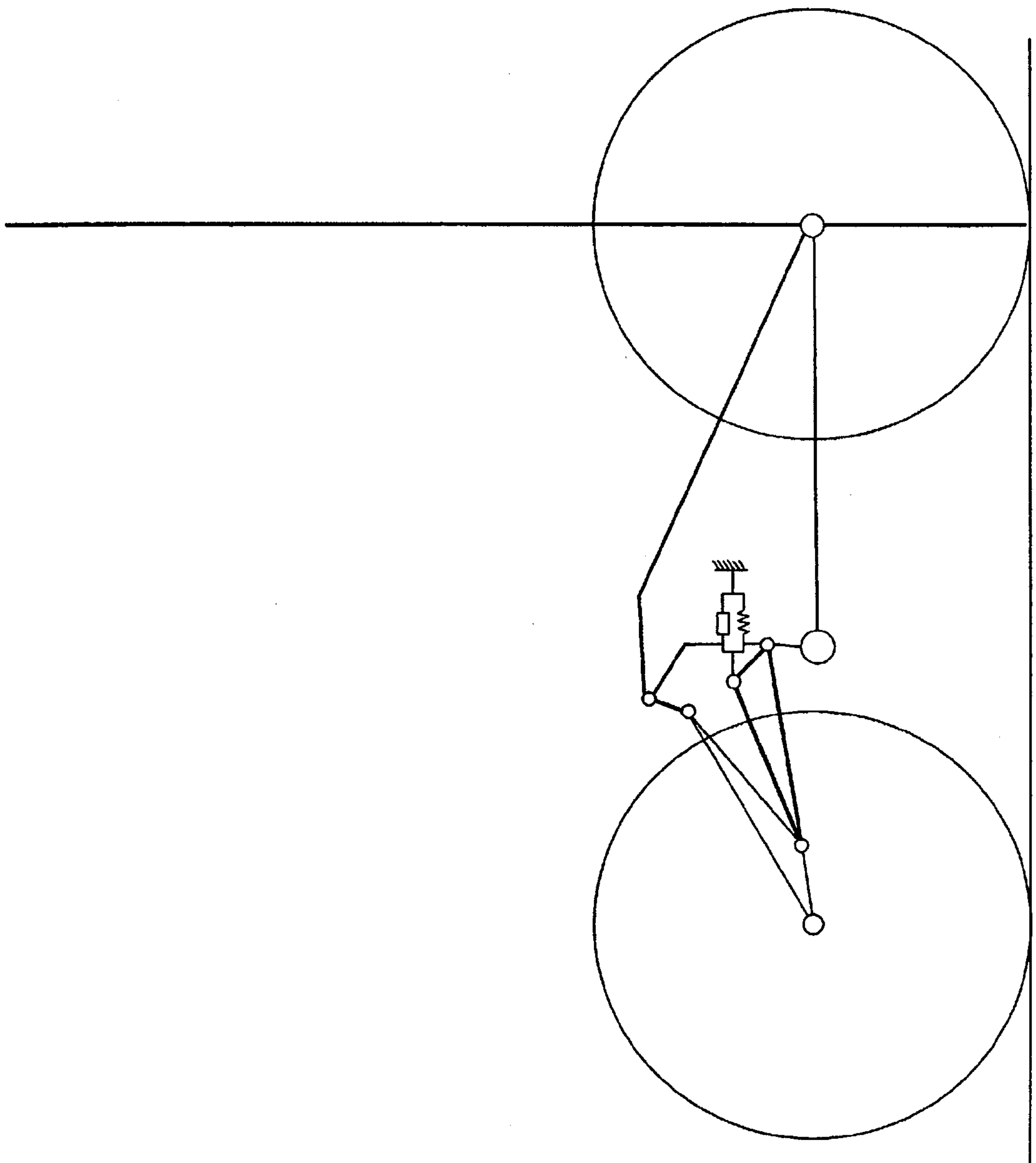

FIGURE 10

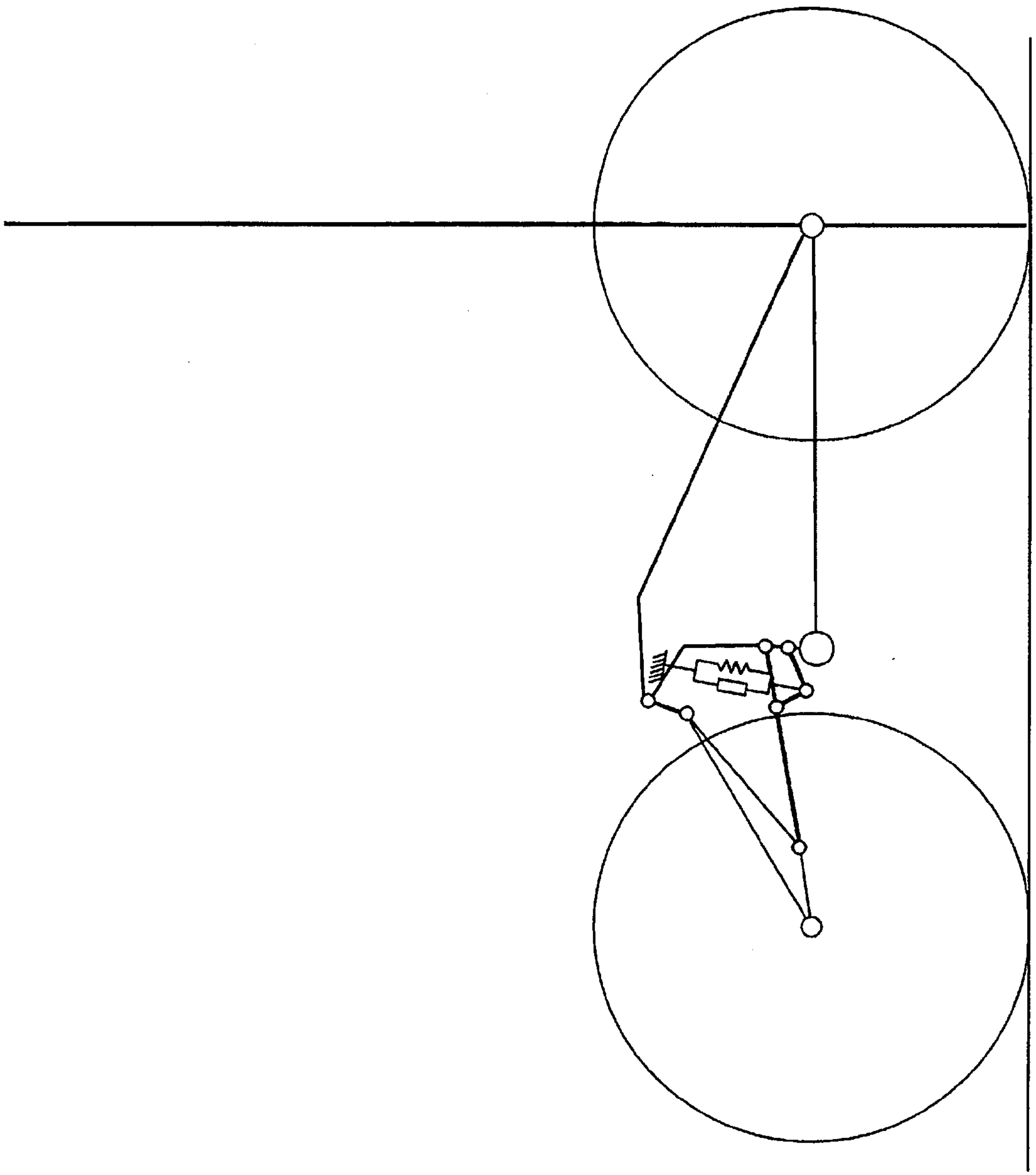

FIGURE 11

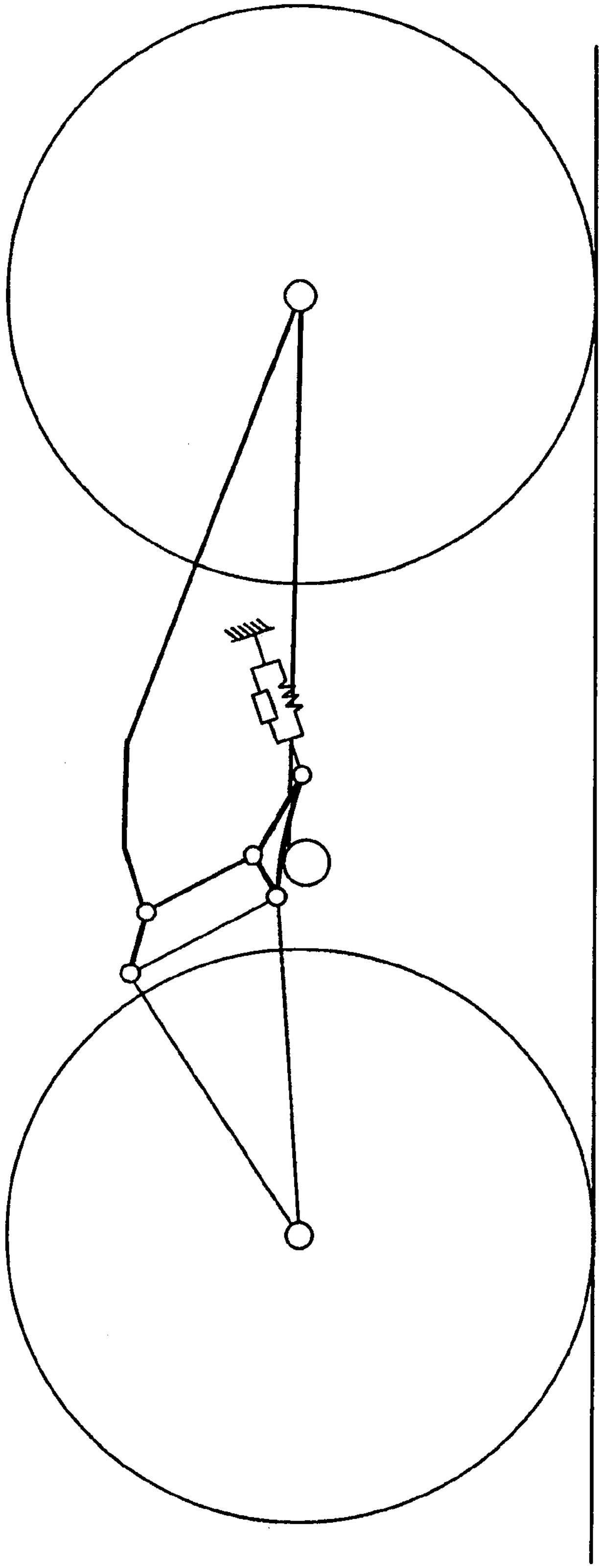

FIGURE 12

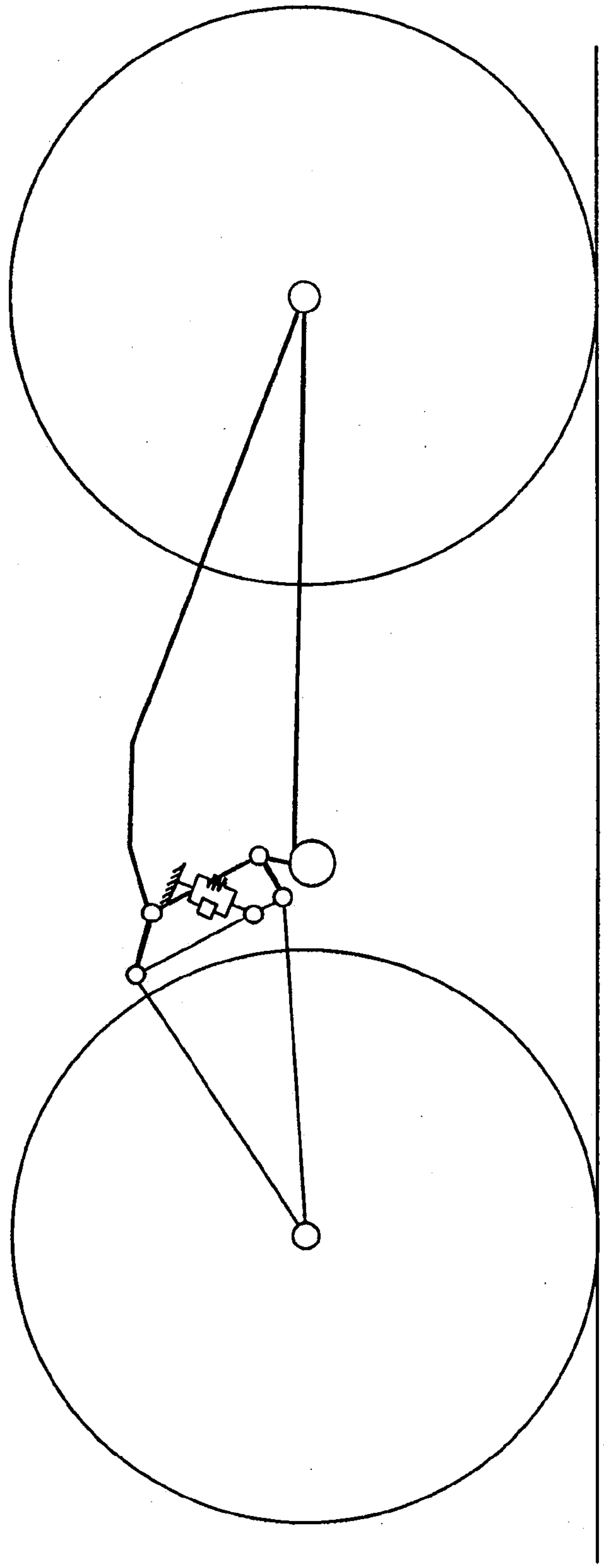

FIGURE 13

VEHICLE SUSPENSION SYSTEMS

This application is a continuation in part of U.S. application Ser. No. 10/669,412, filed Sep. 25, 2003 now U.S. Pat. No. 7,048,292, which is incorporated herein by reference in its entirety.

BACKGROUND

This invention relates to suspension systems capable of reducing or eliminating a squat response.

Automobiles, bicycles, motorcycles, all terrain vehicles, and other wheel driven vehicles are used for various purposes, including transportation and leisure. These vehicles are designed to use a power source to drive through a power transmission system to a wheel or wheels, which transfers rotary motion to the ground via tractive force between a wheel or wheels and the ground. Vehicles are also used to traverse even terrain like paved streets, and uneven terrain like off-road dirt trails. Off road trails are generally bumpier and allow for less wheel traction than paved roads. A bumpier terrain is best navigated with a vehicle that has a suspension system. A suspension system in a vehicle is aimed to provide a smoother ride for an operator or rider, and increase wheel traction over varied terrain. Vehicle suspension systems for the front wheel and for the back wheel are available.

One undesirable effect of suspension systems is the loss of energy in the way of suspension compression or extension during powered acceleration. Such energy loss is particularly notable in vehicles that are driven by low energy power sources, for example, bicycles and solar vehicles. For example, the average rider of a bicycle can exert only a limited amount of power or energy for a short period of time and an even lesser amount for an extended period of time. Therefore, even a very small power loss can have a significant effect on rider performance and comfort. Suspension travel is the distance a suspended wheel travels when the suspension is moved from a fully extended state to a fully compressed state. In bicycles, suspension travel has been increased for many designs and with these increases in suspension travel; the aforementioned energy loss has become even more apparent to riders. But even for a vehicle with a high power energy source, any loss in energy reduces the vehicle's efficiency, for example its fuel efficiency. Where vehicles are used in a manner that requires frequent accelerations, including positive and negative accelerations, the efficiency of the vehicle is particularly affected by any loss of energy resulting from the vehicles geometry, including the geometry and design of its suspension systems.

Thus, by minimizing energy loss resulting from the design of a vehicle's suspension system, the efficiency of the vehicle is improved and thereby its environmental impact. The need for a suspension system that can better preserve a vehicles efficiency and energy has therefore become more pressing. The present invention provides suspension system designs for vehicles that reduce these energy losses.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1***a* is a side view of a chain driven vehicle using a driven wheel suspension system that achieves a squat curve according to certain embodiments of the current invention. The vehicle is shown with the driven wheel suspension system in an uncompressed state.

FIG. **1***b* is a side view of a chain driven vehicle as shown in FIG. **1***a* with the driven wheel suspension system in a completely compressed state.

FIG. **1***c* is an enlarged section of the side view of the chain driven vehicle shown in FIGS. **1***a* and **1***b* with the driven wheel suspension system in a completely uncompressed state.

FIG. **1***d* is an enlarged section of the side view of the chain driven vehicle shown in FIGS. **1***a*, **1***b*, and **1***c* with the driven wheel suspension system in a completely compressed state.

FIG. **2***a* is a side view of a shaft driven vehicle using a driven wheel suspension system that achieves a squat curve according to certain embodiments of the current invention. The vehicle is shown with the driven wheel suspension system in an uncompressed state.

FIG. **2***b* is a side view of a shaft driven vehicle as shown in FIG. **2***a* with the driven wheel suspension system in a completely compressed state.

FIG. **2***c* is an enlarged section of the side view of the shaft driven vehicle shown in FIGS. **2***a* and **2***b* with the driven wheel suspension system in a completely uncompressed state.

FIG. **2***d* is an enlarged section of the side view of the shaft driven vehicle shown in FIGS. **2***a*, **2***b*, and **2***c* with the driven wheel suspension system in a completely compressed state.

FIGS. **3** and **4** show squat curves for suspension systems according to certain embodiments of the invention graphed on a squat curve graph as disclosed herein.

FIGS. **5**–**13** show alternative embodiments of suspension systems comprising a squat curve of the invention. Each embodiment shown includes a spring/damper unit (small irregular box) and different frame members (thicker lines) interconnected through pivots (small circles).

SUMMARY OF THE INVENTION

The current invention relates to new suspension systems for vehicles, for example, bicycles, motorcycles, cars, SUVs, trucks, two wheel vehicles, four wheel vehicles, front wheel suspension vehicles, driven wheel suspension vehicles, and any other kind of vehicle with a suspension system. In certain embodiments of the invention, a suspension system of the invention is capable of facilitating a squat response that lowers the energy loss resulting from squat. In certain preferred embodiments, a suspension system of the invention is capable of lowering energy loss resulting from squat by producing an anti-squat response. An anti-squat response of a suspension system of the invention, in certain embodiments, varies along suspension travel of the vehicle and preferably is higher at the beginning of suspension travel and less thereafter.

Certain embodiments of the invention comprise a wheel suspension design that uses a tuned squat response to reduce powered acceleration induced suspension movement at tactical points during the driven wheel suspension travel. A vehicle designed to use the preferred embodiment of the invention can accelerate under power with a lower amount of energy loss and a more stable vehicle chassis than known systems.

Suspension systems of the invention are useful for a variety of vehicles and preferably in human powered vehicles. The average rider of a bicycle or other human powered vehicle can exert only a limited amount of power or energy for a short period of time and an even lesser amount for an extended period of time. Therefore, even a very small power loss can have a significant detrimental effect on rider performance and comfort. The need for a suspension system that can better preserve the rider's energy has therefore become more pressing. The present invention provides suspension system designs for vehicles that reduce energy loss during powered acceleration.

In certain embodiments of the invention, a wheel suspension system comprises a wheel connected to a wheel carrier unit and said wheel carrier unit connected to spring damper means; and isolating said wheel from a frame structure with the wheel suspension system having an squat curve with said squat curve having a decreasing rate of squat as the suspension system moves from a beginning point in the wheel travel to an ending point in the wheel travel.

In certain embodiments of the invention, a compressible wheel suspension system comprises a wheel connected to a wheel carrier unit and said wheel carrier unit connected to spring damper means; and isolating said wheel from a frame structure with the wheel suspension system having a squat curve with said squat curve having a decreasing squat amount and without said squat amount increasing as the suspension system moves from a beginning point in the wheel travel towards an ending point in the wheel travel increase.

In certain embodiments of the invention, a compressible vehicle suspension system comprises a defined squat curve, with said squat curve having a maximum value at the lowest amount of suspension compression, and a minimum value at a further point in the travel, and a continuously decreasing amount of squat throughout the wheel travel.

In certain embodiments of the invention, a vehicle suspension system comprises a defined squat curve, with said squat curve having a slope that is generally negative at an earlier point in the suspension travel, and a slope that is less negative at a interim point in the suspension travel, and a slope that is then more negative at a latter point in the suspension travel.

In certain embodiments of the invention, a compressible wheel suspension system comprises a wheel connected to a wheel carrier unit and said wheel carrier unit connected to a top link and a bottom link, with a top link connected to spring damper means; With said top and bottom links rotating together in a clockwise direction, and said top and bottom links connecting said wheel carrier to a frame structure, isolating said wheel from the frame structure. Said top link and said bottom link having projected link force lines and said top link projected force line intersecting said lower link projected force line at a point in the beginning of the suspension travel and said top link projected force line intersecting said lower link at a point later in the travel.

In certain embodiments of the invention, a compressible wheel suspension system comprises a wheel connected to a wheel carrier unit and said wheel carrier unit connected to a top link and a bottom link, with said wheel carrier connected to spring damper means; with said top and bottom links rotating together in a clockwise direction, and said top and bottom links connecting said wheel carrier to a frame structure, isolating said wheel from the frame structure. Said top link and said bottom link having projected link force lines and said top link projected force line intersecting said lower link projected force line at a point in the beginning of the suspension travel and said top link projected force line intersecting said lower link at a point later in the travel.

In certain embodiments of the invention, a compressible wheel suspension system comprises a wheel connected to a wheel carrier unit and said wheel carrier unit connected to a top link and a bottom link, with said bottom link connected to spring damper means; with said top and bottom links rotating together in a clockwise direction, and said top and bottom links connecting said wheel carrier to a frame structure, isolating said wheel from the frame structure, said top link and said bottom link having projected link force lines and said top link projected force line intersecting said lower link projected force line at a point in the beginning of the suspension travel and said top link projected force line intersecting said lower link at a point later in the travel.

In certain embodiments of the invention, a compressible wheel suspension system comprises a wheel connected to a wheel carrier unit and said wheel carrier unit connected to a top link and a bottom link, with said top and bottom links connected to spring damper means; with said top and bottom links rotating together in a clockwise direction, and said top and bottom links connecting said wheel carrier to a frame structure, isolating said wheel from the frame structure. Said top link and said bottom link having projected link force lines and said top link projected force line intersecting said lower link projected force line at a point in the beginning of the suspension travel and said top link projected force line intersecting said lower link at a point later in the travel.

In practice, precisely controlling squat in a suspension system can allow for very little suspension movement during powered acceleration with favorable bump compliance. The further a vehicle suspension is compressed, the higher the spring force at the wheel rotational axis. Most powered acceleration happens within the first 40 percent of the suspension travel. Because spring force is lowest in the beginning of a suspension travel, a suspension is more susceptible to manipulation due to squat forces at that time. If enough anti squat force is not present to inhibit mass transfer in the beginning of the suspension travel, the suspension will compress, and when it rebounds, energy will be lost through the damper. The low spring force in the beginning of the suspension travel allows for higher levels of movement than at later points in the suspension travel. Minimizing suspension movement due to mass transfer during powered acceleration reduces the amount of damper movement that occurs at that time. With lower amounts of damper movement comes a lower amount of energy that the damper must dissipate, and therefore more of the acceleration force provided by a power source can be used to accelerate the vehicle. The amount of energy consumed to produce enough anti-squat force to reduce movement earlier in the suspension travel is less than the amount of energy that would be lost in the damper during suspension movement. As a driven wheel suspension system is compressed through its travel, spring force increases, and therefore driven wheel resistance to movement increases. At this later point in the suspension travel, because of the increased spring force, squat force has less of manipulating effect on a wheel suspension. A lower amount of anti squat can be used so that more energy can be transferred to forward movement.

DETAILED DESCRIPTION

Vehicles must be accelerated against their environment to propel an operator or rider across terrain. In order to accelerate these vehicles, a certain amount of energy must be exerted and transformed into rotary motion at a wheel or plurality of wheels. Suspended wheeled vehicle energy conversion types are widely varied. Some vehicles like bicycles, tricycles, and pedal cars use converted human energy as the drive unit. Other vehicles use electric motors or combustion engines, as their drive unit. These electric motors and combustion engines extract rotary motion through the controlled release of chemically stored energy.

Almost all vehicle types use some sort of rotary motion transmission system to transfer rotational force from a drive unit to a wheel or plurality of wheels. A simple bicycle or motorcycle or all terrain vehicle uses a chain or belt to transfer power from a drive unit to a wheel. These chain or belt drive transmissions typically use one sprocket in the front which is coupled to a drive system and one sprocket in the rear which is coupled to a wheel.

More complex bicycles, motorcycles, all terrain vehicles, and automobiles use a shaft drive system to transfer power from a drive system to a driven wheel or wheels. These shaft drive systems transfer power through a rotating shaft that is usually reasonably perpendicular to the driven wheel spinning axis, with power transferred to the driven wheel via a bevel, spiral bevel, hypoid, worm gear drivetrain, or some other means. These single sprocket chain and belt, and shaft driven vehicles can use a direct driven single speed arrangement, where drive unit output shaft speed and torque is transferred to the driven wheel at a constant unchanging ratio. These single sprocket chain and belt, and shaft driven vehicles can also use a commonly found multi speed arrangement, where drive unit output shaft speed and torque is transferred to the driven wheel at a variable ratio through operator selected or automatically selected ratio changing mechanisms.

A bicycle with a more advanced design includes gear changing systems that have clusters of selectable front chainrings and rear sprockets. These gear changing systems give the bicycle rider a selectable mechanical advantage for use during powered acceleration. The mechanical advantage selection, allows a rider spinning a front sprocket cluster via crank arms, to attain lower revolution speed and higher torque values, or conversely, higher revolution speed and lower torque values at a driven wheel.

The current invention, in certain embodiments, is directed at suspension systems that can maintain low energy loss under powered acceleration of the vehicle, for example, a bicycle, a motorcycle, a car, an SUV, a truck, or any other kind of vehicle. Suspension systems of the current invention are useful for a large variety of vehicles, including, but not limited to, human powered vehicles, off road use vehicles with long displacement suspension, high efficiency road going vehicles, and other vehicles.

A vehicle suspension system isolates a vehicle chassis from forces imparted on the vehicle when traversing terrain by allowing the vehicle's ground contact points to move away from impacts at the terrain level and in relation to the vehicle chassis by a compressible suspension movement. The compressible suspension movement that isolates a chassis from these impacts is called suspension displacement or suspension travel. Compressible suspension travel has a beginning point where the suspension is in a completely uncompressed state, and an ending point of displacement, where the suspension is in a completely compressed state. Suspension travel displacement is measured in a direction parallel to and against gravity. In certain preferred embodiments, a suspension system of the invention uses a tuned squat curve to provide an amount of squat closer to or higher in the range of the squat condition known as anti squat in the beginning of a suspension travel displacement, and an amount of squat closer to the range of the squat condition known as anti squat than the initial measurement at a later point in the suspension travel displacement. As a suspension system of the invention is compressed, a spring or damper unit is compressed. As this spring or damper unit is compressed, the force output from the unit rises. As the suspended wheel moves through its axle path, spring force at the wheel rises. A suspended wheel has a compressible wheel suspension travel distance that features a beginning travel point where the suspension is completely uncompressed to a point where no further suspension extension can take place, and an end travel point where a suspension is completely compressed to a point where no further suspension compression can take place. At the beginning of the wheel suspension travel distance, when the suspension is in a completely uncompressed state, the spring is in a state of least compression, and the suspension is easily compressed. In certain preferred embodiments, a higher amount of anti squat in the beginning of the suspension travel is needed to keep the suspension from compressing due to mass transfer under acceleration. As the suspension compresses, spring force at the wheel rises. When spring force rises to levels present in the middle of the suspension travel, mass transfer due to acceleration has a much smaller effect on vehicle traction or chassis attitude because the mass transfer is not capable of significantly compressing the suspension system. At this point, in certain preferred embodiments, the present invention decreases anti squat amount so that a greater amount of mass transfer towards the wheel can occur. This mass transfer allows increased wheel traction while transferring more energy towards forward propulsion.

FIG. **1***a* shows certain embodiments of the invention and it presents a graphical method useful to attain a squat point measurement, and a graphical method to attain suspension geometry kinematical layout from an existing desired measured squat point. Shown in FIG. **1***a* are the following: driven wheel (**1**); swinging wheel carrier link (**2**); upper carrier manipulation link (**3**); lower carrier manipulation link (**4**); chain force vector (**5**); driving force vector (**6**); squat force vector (**7**); upper carrier manipulation link force vector (**8**); lower carrier manipulation link force vector (**9**); squat definition point (**10**); squat layout line (**11**); lower squat measurement definition line (**12**); measured squat distance (**13**); driven wheel axle path (**14**); driven wheel suspension travel distance (**15**); vehicle chassis (**16**); center of the driven wheel tire to ground contact patch (**31**).

FIG. **1***a* exemplifies that as the driven wheel **1** suspension system is completely uncompressed in its driven wheel suspension travel distance **15**, its squat force vector **7** is shown in relation to the vehicle chassis **16**. The squat force vector's **7** measured squat distance **13** which is measured as the perpendicular distance between the lower squat measurement definition line **12** and the squat definition point **10**, is also shown in FIG. **1***a*. As the suspension system is compressed through its driven wheel suspension travel distance **15**, change in measured squat distance **13** over the driven wheel suspension travel distance **15** is used to create a squat curve **17**. FIG. **1***b* shows a side view of a chain driven vehicle as shown in FIG. **1***a* with the driven wheel suspension system in a completely compressed state. Certain embodiments are further exemplified, for example, vectors useful to a graphical method to attain a squat point measurement are shown. Also exemplified is a graphical method useful to attain suspension geometry kinematical layout from an existing desired measured squat point. Shown in FIG. **1***b* in addition to what is presented in FIG. **1***a*, are the following: upper link fixed pivot (**20**); lower link fixed pivot (**21**); upper link floating pivot (**22**); lower link floating pivot (**23**); instant force center (**24**); driven wheel rotation axis (**25**); chain force vector and driving force vector intersection point (**26**); driving cog (**27**); driven cog (**28**); driving cog rotation axis (**29**).

FIG. **1***b* exemplifies that as the driven wheel **1** suspension system is completely compressed through its driven wheel suspension travel distance **15**, its squat force vector **7** moves in relation to the vehicle chassis **16**. The squat force vector's

7 measured squat distance **13**, which is measured as the perpendicular distance between the lower squat measurement definition line **12** and the squat definition point **10**, decreases in relation to the measured squat distance **13** shown in FIG. **1***a*. This change in measured squat distance **13** over the driven wheel suspension travel distance **15**, in certain preferred embodiments, is used to create a squat curve **17**. FIG. **1***b* shows the graphical method used to obtain a squat curve **17** from chain driven vehicle geometry, or chain driven vehicle geometry from a squat curve **17**. In the vehicle shown in FIG. **1***b*, a driven wheel **1** is attached to a swinging wheel carrier link **2**, which pivots at one end of an upper carrier manipulation link **3**. The upper carrier manipulation link **3** is pivotally attached to the vehicle chassis **16** at the upper link fixed pivot **20**. A lower carrier manipulation link **4** is also attached to the swinging wheel carrier link **2**. This lower carrier manipulation link **4** is attached to the vehicle chassis **16** at a lower link fixed pivot **21**. An upper carrier manipulation link force vector **8** is graphed coincident to the swinging wheel carrier link **2** upper pivot and the upper link fixed pivot **20**. The upper carrier manipulation link force vector **8** is graphed so that it intersects a lower carrier manipulation link force vector **9**, which is graphed coincident to the swinging wheel carrier link **2** lower pivot and the lower link fixed pivot **21**. The intersection point of the upper carrier manipulation link force vector **8**, and the lower carrier manipulation link force vector **9** is called the instant force center **24**. A driving force vector **6** is graphed beginning at the driven wheel rotation axis **25**, and passes through the instant force center **24**. A chain force vector **5** is drawn tangent to the tops of the driving cog **27** and driven cog **28**, and intersects the driving force vector **6** at a chain force vector and driving force vector intersection point **26**. The squat force vector **7** is graphed from a beginning point at the center of the driven wheel tire to ground contact patch **31**, and passes through the chain force vector and driving force vector intersection point **26**, before it terminates on a squat layout line **11**. The intersection of the squat force vector **7** and the squat layout line is called the squat layout point **10**. The squat layout line **11** is graphed at a perpendicular angle to gravitational force. A lower squat measurement definition line **12** is graphed beginning at the center of the driven wheel tire to ground contact patch **31** and terminating perpendicular and coincident to the squat layout line **11**. The perpendicular measurement from the lower squat measurement definition line **12** to the squat layout point **10** is called the measured squat distance **13**. This measured squat distance **13** changes as driven wheel suspension travel distance **15** compresses, and is used to create a squat curve **17** in a squat curve graph as shown in FIGS. **3** and **4**.

FIG. **1***c* shows an enlarged section of the side view of the chain driven vehicle shown in FIGS. **1***a* and **1***b* with the driven wheel suspension system in a completely uncompressed state.

FIG. **1***d* shows an enlarged section of the side view of the chain driven vehicle shown in FIGS. **1***a*, **1***b*, and **1***c* with the driven wheel suspension system in a completely compressed state. FIGS. **1***c* and **1***d* further exemplify certain embodiments, for example, points and vectors useful for a graphical method used to attain a squat point measurement, and a graphical method to attain suspension geometry kinematical layout from an existing desired measured squat point.

FIG. **2***a* shows certain embodiments of the invention and it presents a graphical method useful to attain a squat point measurement, and a graphical method to attain suspension geometry kinematical layout from an existing desired measured squat point. Shown in FIG. **2***a* are the following: driven wheel (**1**); swinging wheel carrier link (**2**); upper carrier manipulation link (**3**); lower carrier manipulation link (**4**); squat force vector (**7**); upper carrier manipulation link force vector (**8**); lower carrier manipulation link force vector (**9**); squat definition point (**10**); squat layout line (**11**); lower squat measurement definition line (**12**); measured squat distance (**13**); driven wheel axle path (**14**); driven wheel suspension travel distance (**15**); vehicle chassis (**16**); center of the driven wheel tire to ground contact patch (**31**).

FIG. **2***a* exemplifies that as the driven wheel **1** suspension system is completely uncompressed in its driven wheel suspension travel distance **15**, its defined squat force vector **7** is shown in relation to the vehicle chassis **16**. The squat force vector's **7** measured squat distance **13**, which is measured as the perpendicular distance between the lower squat measurement definition line **12** and the squat definition point **10**, is shown in FIG. **2***a*. As the suspension system is compressed through its driven wheel suspension travel distance **15**, change in measured squat distance **13** over the driven wheel suspension travel distance **15** is used to create a squat curve **17**.

FIG. **2***b* shows a side view of a shaft driven vehicle as shown in FIG. **2***a* with the driven wheel suspension system in a completely compressed state. Certain embodiments are further exemplified, for example, vectors useful to a graphical method to attain a squat point measurement are shown. Also exemplified is a graphical method useful to attain suspension geometry kinematical layout from an existing desired measured squat point. Shown in FIG. **2***b* in addition to what is presented in FIG. **2***a*, are the following: upper link fixed pivot (**20**); lower link fixed pivot (**21**); upper link floating pivot (**22**); lower link floating pivot (**23**); instant force center (**24**); driven wheel rotation axis (**25**); chain force vector and driving force vector intersection point (**26**); driving cog (**27**); driven cog (**28**); driving cog rotation axis (**29**).

FIG. **2***b* exemplifies that as the driven wheel **1** suspension system is completely compressed through its driven wheel suspension travel distance **15**, its defined squat force vector **7** moves in relation to the vehicle chassis **16**. The squat force vector's **7** measured squat distance **13** which is measured as the perpendicular distance between the lower squat measurement definition line **12** and the squat definition point **10**, decreases in relation to the measured squat distance **13** shown in FIG. **2***a*. This change in measured squat distance **13** over the driven wheel suspension travel distance **15** is used to create a squat curve **17**. FIG. **2***b* shows the graphical method used to obtain a squat curve **17** from shaft driven vehicle geometry, or shaft driven vehicle geometry from a squat curve **17**. In the vehicle shown in FIG. **2***b*, a driven wheel **1** is attached to a swinging wheel carrier link **2**, which pivots at one end of an upper carrier manipulation link **3**. The upper carrier manipulation link **3** is pivotally attached to the vehicle chassis **16** at the upper link fixed pivot **20**. A lower carrier manipulation link **4** is also attached to the swinging wheel carrier link **2**. This lower carrier manipulation link **4** is attached to the vehicle chassis **16** at a lower link fixed pivot **21**. An upper carrier manipulation link force vector **8** is graphed coincident to the swinging wheel carrier link **2** upper pivot and the upper link fixed pivot **20**. The upper carrier manipulation link force vector **8** is graphed so that it intersects a lower carrier manipulation link force vector **9**, which is graphed coincident to the swinging wheel carrier link **2** lower pivot and the lower link fixed pivot **21**. The intersection point of the upper carrier manipulation link force vector **8**, and the lower carrier manipulation link force vector **9** is called the instant force center **24**. The squat force vector 7 is graphed from a beginning point at the center of the driven wheel tire to ground contact patch 31, and passes through the instant force center 24, before it terminates on a squat layout line 11. The intersection of the squat force vector 7 and the squat layout line is called the squat layout point 10. The squat layout line 11 is graphed at a perpendicular angle to gravitational force. A lower squat measurement definition line 12 is graphed beginning at the center of the driven wheel tire to ground contact patch 31 and terminating perpendicular and coincident to the squat layout line 11. The perpendicular measurement from the lower squat measurement definition line 12 to the squat layout point 10 is called the measured squat distance 13. This measured squat distance 13 changes as driven wheel suspension travel distance 15 compresses, and is used to create a squat curve 17 in a squat curve graph as shown in FIGS. 3 and 4.

FIG. 2*c* shows an enlarged section of the side view of the shaft driven vehicle shown in FIGS. 2*a* and 2*b* with the driven wheel suspension system in a completely uncompressed state.

FIG. 2*d* shows an enlarged section of the side view of the shaft driven vehicle shown in FIGS. 2*a*, 2*b*, and 2*c* with the driven wheel suspension system in a completely compressed state. FIGS. 2*c* and 2*d* further exemplify certain embodiments, for example, points and vectors useful for a graphical method used to attain a squat point measurement, and a graphical method to attain suspension geometry kinematical layout from an existing desired measured squat point.

FIG. 3 shows a squat curve for suspension systems according to certain embodiments of the invention graphed on a squat curve graph as disclosed herein. The percent of total suspension travel is shown on the x-axis, and the percent of total squat is shown on the y-axis. FIG. 3 exemplifies a squat curve (17). The slope and shape of the squat curve shown in FIG. 3 exemplifies a squat curve produced by suspension systems of the invention, for example, suspension systems including features as illustrated in FIGS. 1*a*–1*d* and FIGS. 2*a*–2*d*. FIG. 3 also exemplifies a graphical method useful to obtain a squat curve graph.

FIG. 4 shows a squat curve for suspension systems according to certain embodiments of the invention. The percent of total suspension travel is shown on the x-axis, and the percent of total squat is shown on the y-axis. FIG. 4 exemplifies a squat curve 17 with tangent lines depicting a slope of the curve at certain points along the squat curve. The slopes exemplified by the tangent lines are the first squat curve slope 18, the second squat curve slope 19, and the third squat curve slope 30. FIG. 4 exemplifies a slope of the squat curve 17 as produced by a suspension system of certain embodiments of the current invention, for example, a suspension system including features as illustrated in FIGS. 1*a*–1*d* and FIGS. 2*a*–2*d*, and that the slope varies as the vehicle suspension travel distance increases. The squat curve 17 produced has a first squat curve slope 18 that has a negative value at the beginning point in the suspension travel, and a second squat curve slope 19 at an interim point that is higher, or less negative, than the first squat curve slope 18 in the suspension travel, and a third squat curve slope 30 at the ending point in the suspension travel that has a lower, or more negative, value than the second squat curve slope 19.

FIGS. 5–13 show alternative embodiments of suspension systems comprising a squat curve of the invention. Each embodiment shown includes a spring/damper unit (small irregular box) and different frame members (thicker lines) interconnected through pivots (small circles).

Mass transfer is discussed. All vehicles have mass. The mass of a suspended static vehicle system can be modeled as shown in the FIG. 1. Mass in all vehicles with a suspension system can be divided into sprung and unsprung mass. Unsprung mass is comprised of the sum of all vehicle parts that move with a suspended wheel. Sprung mass is comprised of the sum of vehicle parts that can remain stationary as a suspended wheel is moved. The dynamic center of the sprung mass as shown in FIG. 2 is a combination of rider and/or passenger mass and the vehicle mass.

The combination of a rider's mass and the sprung mass of the bicycle are always supported fully by the combination of the vehicle's tires. Powered forward acceleration transfers mass from the vehicle's front wheel(s) to the vehicle's driven wheel(s), braking transfers mass from the vehicle's front wheel(s) to the vehicle's driven wheel(s). Riding on the driven wheel(s) only transfers all of the mass to the driven wheel(s), and riding on the front wheel(s) only transfers all of the mass to the front wheel(s).

Due to their combination of short wheelbase (WB) and high center of gravity (CG), motorcycles and bicycles experience the affects of load transfer to a much greater extent than other vehicles in existence. The ratio of the distance from the ground to the CG and the distance between the points where the wheels touch the ground (WB) illustrates this point. For example, a common bicycle will exhibit a center of gravity to wheelbase ratio of nearly 100%, motorcycles are typically near 50%, and passenger cars are typically near 25%. Mass transfer is sometimes also referred to as load transfer.

Energy loss through mass transfer is discussed. One undesirable effect of driven wheel suspension systems is the loss of energy in the way of extreme suspension compression or extension during powered acceleration. This suspension compression or extension is categorized as squat.

A suspension system's geometry and positional relationships between the vehicle drive system components can greatly affect the internal distribution of forces within the vehicle chassis. As a suspension system cycles through its suspension travel, the positional relationships between the suspension system and the vehicle drive system can change, and at the same time, the suspension geometry itself will change. These fluctuations of internal forces are what govern suspension response to powered acceleration and braking. Vehicle attitude in relation to gravity, and sprung weight center of mass change will also govern suspension response to powered acceleration and braking. These external forces are considered stationary and equal when comparing like vehicles in order to determine squat characteristics.

Squat is the result of internal chassis forces that can cause a rear suspension to extend or compress during powered acceleration. Squat is an instantaneous condition that can vary throughout the suspension travel. Instantaneous squat response is governed by sprung mass CG placement, suspension geometry, powertrain component location, and grade in relation to gravity that the vehicle is traveling on. Sprung mass CG placement only defines the amount of squat present in a suspension, and does not change the squat conditions. The squat conditions define the direction of squat force in relation to gravity.

There are three squat conditions that must be considered. The first condition is pro-squat, and describes the condition present when a rear suspension is forced to compress by internal suspension forces under powered acceleration. The second condition is anti-squat. Anti-squat describes the condition present when a rear suspension compression is counteracted by internal suspension forces under powered acceleration. The third condition is zero-squat. Zero-squat occurs only at the instant in between pro-squat and anti-squat, where no suspension manipulating forces are present under powered acceleration. A vehicle suspension operating at the point of zero-squat will not use acceleration forces to manipulate suspension reaction in any way.

Squat force works independent of the spring force that supports a suspended vehicle. Because the squat force is independent of the vehicle spring force, when under acceleration, a vehicle suspension is acted upon by its spring and the squat force together. Suspended vehicles use springs to support the vehicle chassis and dampers to dissipate impact energy when the suspension system is compressed and extended while the vehicle travels over rough terrain. Springs can be in the form of compressive gas springs, leaf springs, or coil springs, and dampers can use fluid or friction to dissipate energy. When a vehicle is at rest, suspended wheels are compressed a certain amount so that the suspended wheel can follow irregular road surfaces with both bumps and dips. The spring that supports a wheel suspension acts as an energy storage device. Vehicle suspensions use the damper units to dissipate energy stored in a spring after the spring is compressed. The further a spring is compressed, the more energy is stored, and the more energy will be dissipated by the damper when the spring rebounds. Because spring force increases as a wheel is compressed into its suspension travel, force at the suspended wheel also increases.

Squat curve graphing is discussed. A squat curve graph is a representation of the squat produced by a compressible suspension system under powered acceleration. The squat curve graph is laid out so that the percentage of suspension travel is graphed on the X axis, and escalating in a positive direction. The minimum suspension travel, which is zero percent suspension compression, is shown at the far left of the x-axis, and the maximum suspension travel, which is represented by 100 percent suspension compression, is shown at the far right of the x-axis. Percent suspension compression is measured and graphed in minimum increments of 5 percent total suspension compression; measured Percent total squat is graphed on the y-axis in an escalating amount. The highest amount of squat is defined as 100 percent, and is represented at the top of the y-axis. These values are taken directly from the squat points which are measured from graphed squat points on the squat layout line. Measurement is taken at a perpendicular distance from the lower squat measurement definition line. Zero percent squat is always measured at the point of zero squat condition. This zero squat condition is measured when the squat point lies directly on the lower squat measurement definition line. At this point, the squat measurement has no value. Any measurement of a squat point that lies below the lower squat definition line is equal to a pro squat amount, and must be graphed as a negative percentage of the 100 percent squat value. The amount of squat closer to or highest in the range of the squat condition known as anti squat is listed as the highest positive squat value, and lower amounts of anti squat, zero squat, and pro-squat are listed as lower percentages of the highest anti squat value. Zero squat is shown when the squat curve crosses or terminates at zero value on the y-axis, and pro squat is graphed as a negative y-axis percentage below the x-axis. For example, if a squat curve begins with a measurement that is measured 100 millimeters above the lower squat measurement definition line, at a point of zero suspension compression, this point will be graphed at a value of 1 on the y-axis, and 0 on the x-axis. If a later point is measured 100 millimeters below the lower squat measurement definition line, at a point of 100 percent suspension compression, this point will be graphed at a value of −1 on the y-axis, and 1 on the x-axis. In the squat curve graph, the distance set to equal 100 percent suspension travel and the distance set to equal 100 percent squat should be set as equal distances. Therefore, the distance between zero value for squat to maximum value for squat will be equal to the graphed distance between zero value for suspension compression to maximum value for suspension compression. When desired squat point values are known and graphed versus their corresponding percent measured suspension compression values, the points can be connected from point to point using typical graphing method A curve can then be fit to the point to point graph so that the curve represents a smoothed best fit version of the point to point graph. The most efficient method to obtain such a curve is to use a computer program such as Microsoft Excel, available from Microsoft Corporation, One Microsoft Way, Redmond, Wash. 98052-6399, USA. Using Microsoft Excel, a user can input the escalating suspension travel measurements beginning with the zero percent measurement and ending with the 100 percent measurement, and can input the measured or preferred squat point measurements that coincide with their percent suspension travel measurements. Microsoft Excel then can be used to create a graph of the points with a curve fit to the graphed points. This graphed curve is the discussed squat curve.

Slope of a squat curve between two points on a curve is defined by the standard coordinate geometry equation: slope=rise/run. A squat curve that has a squat amount at zero suspension travel, with 20 percent less squat at a point 10 percent into the wheel suspension travel compression will have a slope of −2, because per the equation slope=rise/run, −0.2/0.1=−2. A squat curve that has a pro squat amount at zero suspension travel, with 20 percent more pro squat at a point 10 percent into the wheel suspension travel compression will have a slope of −2, because per the equation slope=rise/run, −0.2/0.1=−2. A squat curve can be produced for any wheel suspension system by graphing the percent of squat throughout the suspension travel.

In certain embodiments, a suspension system according to the invention has a squat curve with a negative, or decreasing, slope. In certain preferred embodiments, the slope of the squat curve is more negative at the beginning of suspension travel than in the interim, or mid range, of suspension travel. In certain other preferred embodiments, the slope of the squat curve is more negative at the end of suspension travel than in the interim, or mid range, of suspension travel. In certain other preferred embodiments, the slope of the squat curve is more negative at the beginning of suspension travel than at the end of suspension travel.

In certain embodiments, the beginning of the suspension travel is 0 to 50 percent, or about 0 to about 50 percent, of suspension travel; or 0 to 40 percent, or about 0 to about 40 percent, of suspension travel; or 0 to 30 percent, or about 0 to about 30 percent, of suspension travel; or 0 to 20 percent, or about 0 to about 20 percent, of suspension travel; or 0 to 10 percent, or about 0 to about 10 percent, of suspension travel; or 0 to 5 percent, or about 0 to about 5 percent, of suspension travel; or 0 or about 0 percent of suspension travel. In certain embodiments, the interim, or mid range, of the suspension travel is 25 to 75 percent, or about 25 to about 75 percent, of suspension travel; or 30 to 70 percent, or about 30 to about 70 percent, of suspension travel; or 35 to 65 percent, or about 35 to about 65 percent, of suspension travel; or 40 to 60 percent, or about 40 to about 60 percent, of suspension travel; or 45 to 55 percent, or about 45 to about 55 percent, of suspension travel; or 50 percent or about 50 percent, of suspension travel; or 60 to 80 percent, or about 60 to about 80 percent, of suspension travel; or 65 to 75 percent, or about 65 to about 75 percent, of suspension travel; or 70 percent or about 70 of suspension travel; or 50 to 60 percent, or about 50 to about 60 percent, of suspension travel. In certain embodiments, the end of the suspension travel is 70 to 100 percent, or about 70 to about 100 percent, of suspension travel; or 75 to 100 percent, or about 75 to about 100 percent, of suspension travel; or 80 to 100 percent, or about 80 to about 100 percent, of suspension travel; or 85 to 100 percent, or about 85 to about 100 percent, of suspension travel; or 90 to 100 percent, or about 90 to about 100 percent, of suspension travel; or 95 to 100 percent, or about 95 to about 100 percent, of suspension travel; or 100 or about 100 percent of suspension travel.

In certain embodiments, a suspension system of the invention has a squat curve with a slope in the beginning of suspension travel of −0.2 to −5, or about −0.2 to about −5; of −0.5 to −4.5, or about −0.5 to about −4.5; of −0.75 to −4.0, or about −0.75 to about −4.0; of −1.0 to −3.5, or about −1.0 to about −3.5; of −1.5 to −3.0, or about −1.5 to about −3.0; of −2.0 to −2.5, or about −2.0 to about −2.5. In certain embodiments, a suspension system of the invention has a squat curve with a slope in the interim, or mid range, of suspension travel of −0.0001 to −5, or about −0.0001 to about −5; of −0.01 to −4.0, or about −0.01 to about −4.0; of −0.1 to −3.0, or about −0.1 to about −3.0; of −0.2 to −2.0, or about −0.2 to about −2.0; of −0.3 to −1.2, or about −0.3 to about −1.2; of −0.4 to −0.8, or about −0.4 to about −0.8. In certain embodiments, a suspension system of the invention has a squat curve with a slope in the end of suspension travel of −0.0002 to −1000, or about −0.0002 to about −1000; of −0.1 to −500, or about −0.1 to about −500; of −0.2 to −50, or about −0.2 to about −50; of −0.3 to −10, or about −0.3 to about −10; of −0.4 to −5.0, or about −0.4 to about −5.0; of −0.6 to −2.0, or about −0.6 to about −2.0.

Graphical kinematical squat curves are discussed. Graphical methods can be used to determine suspension kinematical layout used to attain a desired squat curve for a suspension. For shaft drive and chain drive vehicles, graphical layout is identical until factoring in the unique features of each powertrain. Any suspended wheel in a vehicle has an axle path that a wheel follows when a suspension is moved through suspension travel. The curvature of this axle path and its layout in relation to specific powertrain components define a squat curve. A squat curve is a measurement of the changing magnitude and direction of squat developed under powered acceleration as suspension system is cycled through suspension travel from its beginning uncompressed point to its ending fully compressed point. Every instantaneous point in a suspension travel has a corresponding instantaneous amount of squat present. These instantaneous squat points can be measured or graphed as a point on the squat layout line at a perpendicular distance from the lower squat layout line. When the desired instantaneous amounts of squat at instantaneous points in the suspension travel are known, squat definition points can be graphed in conjunction with each other, beginning when a suspension is in its uncompressed state and ending in its fully compressed state, and in relation to the vehicle geometry to obtain a suspension kinematical layout which will attain the desired squat curve. The squat curve beginning value is measured at the point where the suspension system is in its completely uncompressed state. As the suspension is cycled further through suspension travel towards complete compression pausing at a minimum of 5 percent total suspension travel increments, further squat points are measured and graphed versus their correlating escalating percent total suspension travel increments. Suspension travel displacement is measured in a direction parallel to and against gravity, and parallel to the instantaneous squat point measurements. Critical and known preexisting defining points such as vehicle wheelbase, powertrain location, and center of mass are graphed alongside the squat definition points to obtain a clear picture of vehicle squat performance. Vehicle graphs for obtaining and defining squat performance are always laid out with the vehicle viewed in the side elevational view.

A squat layout line is drawn parallel to and against gravitational force through center of the front wheel contact patch between the tire and the ground and terminating at further points. A squat definition point, which is taken directly from the aforementioned squat curve will be graphed on this squat layout line. A squat lower measurement definition line is drawn from the center of the driven wheel tire to ground contact patch perpendicular to and terminating on the squat layout line. Squat definition points are drawn on the squat definition line in relation to one another, and in relation to the squat lower measurement definition line. A squat definition point drawn above the squat lower measurement definition line will correlate with a squat amount. A squat definition point drawn coincident with the squat lower measurement definition line will correlate with a zero squat amount. A squat definition point drawn below the squat lower measurement definition line will correlate with a pro squat amount. A squat force vector is drawn from the center of the driven wheel tire to ground contact patch to the squat point on the squat layout line. As the suspension is moved through instantaneous measured points through suspension travel, the squat force vector is drawn with a beginning point at the center of the rear tire to ground contact patch, and an ending point at its corresponding measured instantaneous squat point graphed on the squat layout line.

Diversion in graphical method to obtain specific suspension system kinematical layouts from a desired squat curve must occur when factoring in specifics for different types of power transfer systems such as shaft drive or chain drive.

A shaft drive system generally uses a power transmission system that can transmit power via rotary motion from a power unit output shaft to a wheel shaft. The two shafts are generally fixed at close to a perpendicular angle in one plane. Power transmission systems can vary from gears to cogs to friction wheels and other types of systems, all herein referred to universally as cogs. These shaft drive systems feature a driving cog which is rotatably attached to the power unit output, a first intermediate cog, which transfers rotational motion from the driving cog to a relatively perpendicular shaft, a second intermediate cog, which transfers rotational motion from the shaft to a driven cog which is rotatably attached to the rotation axis of a wheel.

Shaft drive vehicle powertrains and suspensions typically take one of two forms. These are, a single pivot system, or a multi link system. A simple single pivot system features a driven cog that is fixed to and housed within a swinging wheel carrier link which pivots around a single pivot. In this arrangement, there is only one pivot connecting the swinging wheel carrier link to the vehicle frame structure. The rotating drive torque is acted against by the driven cog housing, which is part of the swinging wheel carrier link. Action against the drive torque in the swinging wheel carrier link causes a torque about the ling single frame pivot. The addition of this torque plus the driving force imparted through the wheel tire combination to the ground through a tire to ground contact patch totals a squat response. An instantaneous pivot location for a single pivot shaft drive system can be found at any point on a drawn squat force vector that correlates with the desired instantaneous squat response. These single pivot systems produce a linear squat curve.

A multi pivot linkage can be used to alter squat characteristics and obtain a variable squat curve in a shaft driven wheel suspension system. A multi link shaft drive suspension system isolates the torque passed through the driven cog in the system from the swinging link system. In a 4-bar variation, the driven cog is attached to a swinging wheel carrier link, which pivots at one end of a first swinging link. The first carrier manipulation link is pivotally attached to the vehicle chassis at the end opposite of the swinging wheel carrier link pivot. A torque reaction, like the one discussed in the single pivot shaft drive system works to rotate the swinging wheel carrier link against the first carrier manipulation link. A second carrier manipulation link is also attached to the swinging wheel carrier link. This second carrier manipulation link is attached to the vehicle chassis at a different location from the first swinging carrier manipulation link. The second carrier manipulation link works to inhibit free rotation of the swinging wheel carrier link against the first carrier manipulation link. To find instantaneous carrier manipulation link pivot points which will give a desired instantaneous squat amount, its correlating desired squat force vector must be graphed. The two swinging wheel carrier link pivots are next defined. Carrier manipulation link force lines are drawn so that a force line passes directly through the center of the rearward pivots which are coincident with the pivots on the swinging wheel carrier link. The carrier manipulation link force lines are drawn so that they intersect on the desired squat force vector. The first and second vehicle chassis pivots can be positioned upon the corresponding first and second carrier manipulation link force lines to attain the desired instantaneous squat response. Graphing the carrier manipulation link force lines and desired squat force vectors together overlaid at multiple points in the suspension travel will allow the designer to choose pivot point locations and kinematical suspension layout that can attain a desired variable squat curve.

A chain drive powertrain system uses a chain or belt to transmit power between two reasonably parallel shafts. Chain drive systems are very common in motorcycle, ATV, and bicycle applications because of their light weight, robustness, and simplicity in both manufacturing and use. The chain drive systems feature a driving cog and a driven cog, with the driving cog attached to a power source, and a driven cog rotatably attached to the rotation axis of a wheel. The driven wheel or wheels is/are attached to a swinging link or linkage system via a bearing or bushing system, which allows rotational motion of the driven wheel or wheels in relation to the swinging link or linkage system.

Chain drive suspensions typically take one of several forms. These include single pivot systems, multi link systems, cam/track type systems, and flexure type systems. The suspensions can also feature variable chainline type designs, which manipulate a chain force vector line through the use of a pulley system that moves with the suspension. A single pivot system uses a single pivoting suspension link to transmit force between a suspended wheel and a chassis. A multi link system uses an arrangement of pivoting suspension links to transmit force between a suspended wheel and a chassis. A cam/track type system that uses sliding elements but does not use links to attain force transfer from a wheel axle to a chassis is also possible but uncommon in practice. Flexure type systems use flexing elements to transmit power from a suspended wheel to a chassis structure. In all types of the chain driven wheel suspension system mentioned above, the driving force can be represented as a vector drawn perpendicular to the driven wheel axle path. In a chain driven suspension, driving force is always the major force component when compared to chain pull.

There are two internal forces present within a chain driven vehicle chassis that together create a squat response. These two forces are driving force, and chain pull force.

When a chain driven vehicle is accelerated, force is transferred from a power source to a driving cog. This driving cog transmits its force through a chain to a driven cog. The force direction and magnitude present in the tensioned chain are referred to as chain pull force. Fixed chainline type designs are present where at any instantaneous point, a single driving cog is fixed rotationally on a chassis structure, and a driven cog is attached to a suspension member, and force is transmitted from the driving cog to the driven cog through a chain. In this fixed chainline type design, the chainline force vector is always located at one end by the tensioned chainline tangent point where the chain is fixed in relation to the vehicle chassis structure, and by the tensioned chainline tangent point of the moving pulley at the opposite end.

In variable chainline type designs, which manipulate a chain force vector line through the use of a pulley system that moves with the suspension, the chainline force vector is always located at one end by the tensioned chainline tangent point where the chain is fixed in relation to the vehicle chassis structure, and by the tensioned chainline tangent point of the moving pulley at the opposite end. Sliding elements can also be substituted for pulleys in this application.

In the chain drive powertrain, the driven cog is rotatably attached to a wheel/tire combination. The wheel pushes against the ground resulting in friction. As the wheel rotates a driving force transmitted from the contact patch through the wheel structure and a force is imparted at the rear hub axle. This pushing force can be transferred to the chassis via a wheel suspension system, ultimately pushes the vehicle forward. This pushing force is referred to as driving force. The driving force direction is measured and represented graphically as a driving force vector drawn from the driven wheel rotation axis, perpendicular to the driven axle path, where the axle path is defined as a line that a suspended wheel rotational axis travels as a suspension is moved through suspension travel. This axle path can be a constant curvature or changing curvature line depending on suspension layout.

A simple single pivot system features a driven cog that is rotatably attached to a wheel, which is rotatably attached to a swinging wheel carrier link which pivots around a singular pivot. In this arrangement, the suspended wheel travels in a constant radius arc. To find the instantaneous swinging link pivot point for a single pivot chain drive system, which will give a desired instantaneous squat amount, its correlating desired squat force vector must be graphed. Because there is only one link in the single pivot suspension, the swinging link pivot will lie coincident with the driving force line. Desired vehicle geometry is graphed in a side view. This vehicle geometry will include the size, location, and center points of vehicle tires, powertrain component layout, and the direction of gravitational force. A squat layout line is graphed first. A desired squat force vector is drawn from the center of a rear wheel contact patch to the desired squat layout point on a squat layout line as described previously.

Next, the chain force vector is graphed in relation to the powertrain components as described previously. The chain force vector is drawn so that it intersects the squat force vector. Finally, the driving force vector is drawn from the center of the rear wheel axis to the intersection point of the squat force vector and chain pull force vector. The pivot point for the single pivot swinging link suspension arm will lie at any point along the driving force vector to achieve the desired instantaneous squat amount. Graphing the chain pull force vector, and squat force vectors together overlaid at multiple points in the suspension travel will allow the designer to find driving force vectors at multiple points through the suspension travel. The crossing point of the overlaid driving force vectors for different points in the suspension travel define the single pivot point location and kinematical suspension layout that can attain the desired squat curve.

Multi link systems, cam/track (sliding link) type systems, and flexure type systems feature a driven cog that is rotatably attached to a wheel, which is rotatably attached to a swinging wheel carrier link which moves the wheel along an axle path that is defined by a multi element system. To aid the analysis of multi-element systems, it is simplest to define or measure an axle path which will guide a wheel, and then define the elements that will give the desired axle path later, as opposed to trying to define elements first and measure axle path as a byproduct later to attain a desired result. Multi element systems do not have a single hardware defined pivot point like a single fixed pivot system does. The multi element systems use combinations of links or cams to project a virtual or instantaneous pivot point. This pivot point can always be found at a point along a driving force vector, which is drawn perpendicular to a driven wheel axle path as previously described.

To find the axle path which will give a desired instantaneous squat amount, its correlating desired squat force vectors must be graphed. Desired vehicle geometry is graphed in a side view. This vehicle geometry will include the size, location, and center points of vehicle tires, vehicle ground plane, powertrain component layout, and the direction of gravitational force. A vehicle wheel suspension system always has a minimum suspension travel point, where the suspended wheel is at its zero compressed suspension travel point, and a maximum suspension travel point, where the suspended wheel is at its 100 percent compressed suspension travel point. Several overlaid graphs must be made to obtain a squat curve. The minimum increment in suspension compression displacement that can be used to graph an accurate squat curve from the graphical method using squat force vectors as presented has been found to be 5 percent of total suspension compression displacement between graphed squat force vectors. A squat layout line is graphed first. A desired squat force vector is drawn from the center of a driven wheel contact patch to the desired squat layout point on a squat layout line as described previously. Next, the chain force vector is graphed in relation to the powertrain components as described previously. The chain force vector is drawn so that it intersects the squat force vector. Finally, the driving force vector is drawn from the center of the driven wheel axis to the intersection point of the squat force vector and chain pull force vector. The instantaneous pivot point for the single pivot swinging link suspension arm will lie at any point along the driving force vector to achieve the desired instantaneous squat amount. Graphing the chain pull force vector, and squat force vectors together overlaid at multiple points in the suspension travel will allow the designer to find driving force vectors at multiple points through the suspension travel. The crossing point of the overlaid driving force vectors for different points in the suspension travel define the instantaneous pivot point movement through the suspension travel, and kinematical suspension layout that can attain the desired squat curve. For multi element systems, there are several methods that can define element layout based on a desired axle path, for example, by using kinematical analysis computer software. Software that can perform this specific function is marketed under the names SyMech, which is available from SyMech Inc, 600 Townsend Street, San Francisco, Calif., 94107, USA, and SAM, which is available from ARTAS—Engineering Software, Het Puyven 162, NL-5672 RJ Nuenen, The Netherlands. This software allows a user to define an axle path, and set parameters such as mechanical element type, number of mechanical elements, and desired location of anchor components. The software will then suggest multiple link layout choices that will meet all of the set forth parameters. Graphical analysis can also be performed by hand. In a hand graphical analysis, the mechanical components of a multi element system are measured at multiple points through the suspension travel. At each point in the suspension travel, the instant center of the link system is graphed. A common 4-bar linkage suspension system features a driven cog that is rotatably attached to a driven wheel, which is rotatably attached to a swinging wheel carrier link which is pivotably attached to two separate carrier manipulation links. The swinging links are pivotably attached to a vehicle chassis at their other ends. The instant center in a 4 bar pivoting linkage system such as shown in FIG. **1***a*, is found by projecting individual link force lines through both pivots of each of the two carrier manipulation links that support the swinging wheel carrier. The two carrier manipulation link force lines are projected so that they intersect each other. This intersection point is commonly known at the instant force center. A driving force line can be drawn directly from the rotation axis of the driven wheel to this instant force center. As the carrier manipulation links rotate on their pivots, the instant center position changes in relation to the driven wheel rotation axis and the vehicle chassis. This causes the driving force line to move in relation to the chain force line. Because the squat force line is defined in part by the location of the driven wheel contact patch, and the intersection between the driving force vector and the chain force vector, a change in squat vector direction can occur. The perpendicular distance from the lower squat definition line to the point at which this squat direction vector intersects the drawn squat layout line to is measured and recorded.

Four bar sliding link suspension systems are analyzed identically to 4 bar pivoting systems, but the identification of the instant center is performed in a slightly different manner due to the constraints of the sliding link system. Four bar sliding link systems feature a driven cog that is rotatably attached to a driven wheel, which is rotatably attached to a swinging wheel carrier link which is pivotably attached to two separate sliding carrier manipulation sliding blocks. The individual carrier manipulation sliding blocks move on individual sliding rails. The instant center in a 4 bar sliding linkage system is found by projecting individual sliding link force lines centered at the pivots of each of the two carrier manipulation sliding block that support the swinging wheel carrier. The carrier manipulation sliding block force lines are projected perpendicular to the sliding rail so that the two carrier manipulation sliding black force lines intersect each other. This intersection can be referred to as the instant force center. A driving force line can be drawn directly from the rotation axis of the driven wheel to this instant force center. As the carrier manipulation sliding blocks slide on their respective sliding rails, the instant center position changes in relation to the driven wheel rotation axis and the vehicle chassis. This causes the driving force line to move in relation to the chain force line. Because the squat force line is defined in part by the location of the driven wheel contact patch, and the intersection between the driving force vector and the chain force vector, a change in squat vector direction can occur. The perpendicular distance from the the lower squat definition line to the point at which this squat direction vector intersects the drawn squat layout line to is measured and recorded.

Measurement of multi element systems to determine axle path can be done graphically, or by using measurement equipment. Using measurement equipment, the vehicle can be rigidly mounted and oriented so that the suspended wheel can be moved freely through measured points in its suspension travel while the chassis stays stationary. In a side view orientation, the horizontal and vertical distance from the suspended wheel rotation axis to a fixed point on the vehicle frame at multiple points in the suspension travel is taken. As the suspension is cycled through suspension travel, the corresponding measurements of horizontal and vertical distance form a wheel rotation axis travel path in relation to the vehicle chassis. This path is referred to as the axle path.

Analysis has shown that a vehicle with a compressible suspension system using a chain driven suspended wheel achieves the squat curve **17** of the current invention by having a layout that features a driven cog that is rotatably attached to a driven wheel, which is rotatably attached to a swinging wheel carrier link which is pivotably attached to separate upper and lower carrier manipulation links. The upper and lower carrier manipulation links are pivotably attached to a vehicle chassis at their other ends. The upper and lower carrier manipulation links rotate in the same rotational direction about their respective fixed axis at the vehicle chassis. The upper carrier manipulation link is arranged in relation to the lower carrier manipulation link so that the instant center projected by the two carrier manipulation links, when measured at zero percent suspension compression, is at a point that is beyond the outer limits of the two pivots of the lower carrier manipulation link. This condition is shown in FIGS. **1***a* and **1***c*. As the suspension is compressed towards a point of full compression, the rotation of the upper and lower carrier manipulation links in relation to each other causes the instant center of the linkage system to lie at points on the lower carrier manipulation link in between the lower carrier manipulation link fixed vehicle chassis pivot, and moving pivot attached to the swinging wheel carrier link. This condition is shown in FIGS. **1***b* and **1***d*.

Analysis has shown that a vehicle with a compressible suspension system using a shaft driven suspended wheel achieves the squat curve **17** of the current invention by having a layout that features a driven cog that is rotatably attached to a driven wheel, which is rotatably attached to a swinging wheel carrier link which is pivotably attached to separate upper and lower carrier manipulation links. The upper and lower carrier manipulation links are pivotably attached to a vehicle chassis at their other ends. The upper and lower carrier manipulation links rotate in a contra rotational direction about their fixed axes at the vehicle chassis. The upper carrier manipulation link is arranged in relation to the lower carrier manipulation link so that the instant center projected by the two carrier manipulation links, when measured at zero percent suspension compression, lies at a point on the lower carrier manipulation link in between the lower carrier manipulation link fixed vehicle chassis pivot, and moving pivot attached to the swinging wheel carrier link. This condition is shown in FIGS. **2***a* and **2***c*. As the suspension is compressed towards a point of full compression, the rotation of the upper and lower carrier manipulation links in relation to each other causes the instant center of the linkage system to lie at a point that is beyond the outer limits of the two pivots of the lower carrier manipulation link. This condition is shown in FIGS. **2***a* and **2***d*.

The present invention is not to be limited in scope by the specific embodiments described herein, which are intended as single illustrations of individual aspects of the invention, and functionally equivalent methods and components are within the scope of the invention. Indeed, various modifications of the invention, in addition to those shown and described herein, will become apparent to those skilled in the art from the foregoing description. Such modifications are intended to fall within the scope of the appended claims. All cited publications, patents, and patent applications are herein incorporated by reference in their entirety.

What is claimed is:

1. A driven wheel suspension comprising a driven wheel, a damper unit, an upper carrier manipulation link and a lower carrier manipulation link, wherein said upper carrier manipulation link and said lower carrier manipulation link are arranged so that force lines through pivots of each of said manipulation links intersect in an instant center, and wherein said instant center is positioned beyond outer limits of two pivots of the lower carrier manipulation link at zero percent suspension compression and in between said two pivots as the suspension is compressed towards a point of full compression.

2. The suspension system of claim **1**, wherein the suspension system is useful for a chain driven vehicle.

3. The suspension system of claim **1**, wherein the suspension system is useful for a belt driven vehicle.

4. The suspension system of claim **1**, wherein the suspension system is useful for a human powered vehicle.

5. The suspension system of claim **1** wherein a damper unit is connected to the upper carrier manipulation link.

6. The suspension system of claim **1** wherein a damper unit is connected to the lower carrier manipulation link.

7. The suspension system of claim **1** wherein a damper unit is connected to a wheel carrier link.

8. The suspension system of claim **1** wherein a damper unit is connected to the upper carrier link and lower carrier manipulation link.

9. The suspension system of claim **1**, wherein the damper unit is selected from the group consisting of a spring, a compression gas spring, a leaf spring, a coil spring, and a fluid.

* * * * *

US007128329C1

(12) EX PARTE REEXAMINATION CERTIFICATE (6481st)

United States Patent

Weagle

(10) **Number: US 7,128,329 C1**

(45) **Certificate Issued: Oct. 14, 2008**

(54) **VEHICLE SUSPENSION SYSTEMS**

(75) Inventor: **David Weagle**, Edgartown, MA (US)

(73) Assignee: **DW-Link Incorporated**, Edgartown, MA (US)

Reexamination Request:
No. 90/008,792, Jul. 27, 2007

Reexamination Certificate for:

Patent No.:	**7,128,329**
Issued:	**Oct. 31, 2006**
Appl. No.:	**10/949,264**
Filed:	**Sep. 24, 2004**

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(63) Continuation-in-part of application No. 10/669,412, filed on Sep. 25, 2003, now Pat. No. 7,048,292.

(51) **Int. Cl.**
B62K 25/00 (2006.01)

(52) **U.S. Cl.** **280/284**

(58) **Field of Classification Search** .... None
See application file for complete search history.

(56) **References Cited**

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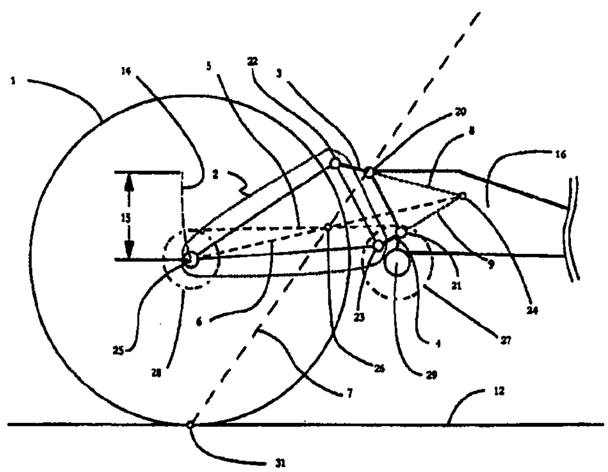
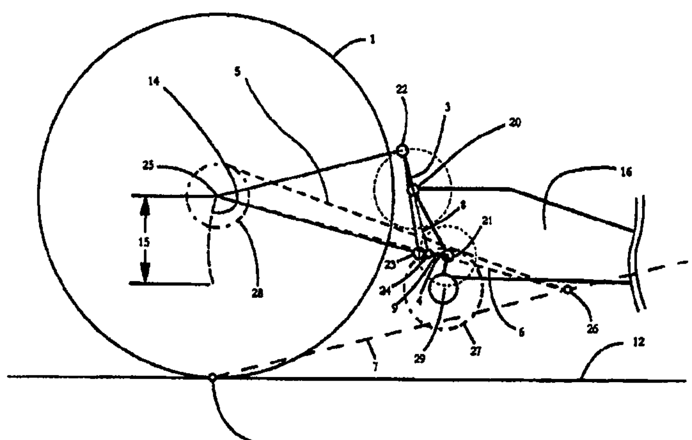
Primary Examiner—Joseph A. Kaufman

(57) **ABSTRACT**

A wheel suspension system having under powered acceleration a squat response that begins in the realm of anti squat and passes through a point of lessened anti squat at a further point in the travel.





EX PARTE REEXAMINATION CERTIFICATE ISSUED UNDER 35 U.S.C. 307

THE PATENT IS HEREBY AMENDED AS INDICATED BELOW.

Matter enclosed in heavy brackets [ ] appeared in the patent, but has been deleted and is no longer a part of the patent; matter printed in italics indicates additions made to the patent.

AS A RESULT OF REEXAMINATION, IT HAS BEEN DETERMINED THAT:

Claim **1** is determined to be patentable as amended.

Claims **2–9**, dependent on an amended claim, are determined to be patentable.

1. A driven wheel suspension comprising a driven wheel, a damper unit, an upper carrier manipulation link and a lower carrier manipulation link, wherein said upper carrier manipulation link and said lower carrier manipulation link are arranged so that force lines through pivots of each of said manipulation links intersect in an instant center, and wherein said instant center is positioned beyond outer limits of two pivots of the lower carrier manipulation link at zero percent suspension compression and in between said two pivots as the suspension is *fully* compressed **[**towards a point of full compression**]**.

* * * * *

Exhibit C

US007828314B2

(12) **United States Patent**
Weagle

(10) **Patent No.: US 7,828,314 B2**
(45) **Date of Patent: Nov. 9, 2010**

(54) **VEHICLE SUSPENSION SYSTEMS**

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(73) Assignee: **DW-Link Incorporated**, Edgartown, MA (US)

(*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 0 days.

(21) Appl. No.: **11/525,661**

(22) Filed: **Sep. 22, 2006**

(65) **Prior Publication Data**

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Related U.S. Application Data

(60) Division of application No. 10/949,264, filed on Sep. 24, 2004, now Pat. No. 7,128,329, which is a continuation-in-part of application No. 10/669,412, filed on Sep. 25, 2003, now Pat. No. 7,048,292.

(51) **Int. Cl.**
B62K 25/04 (2006.01)

(52) **U.S. Cl.** **280/284**

(58) **Field of Classification Search** 280/283, 280/284; 180/227

See application file for complete search history.

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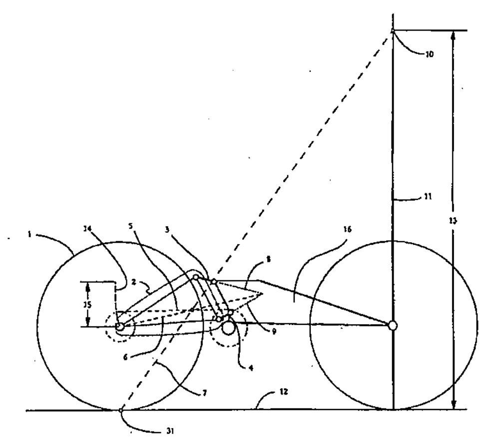
Primary Examiner—Kevin Hurley
(74) *Attorney, Agent, or Firm*—Stahl Law Firm

(57) **ABSTRACT**

A wheel suspension system having under powered acceleration a squat response that begins in the realm of anti squat and passes through a point of lessened anti squat at a further point in the travel.

8 Claims, 19 Drawing Sheets



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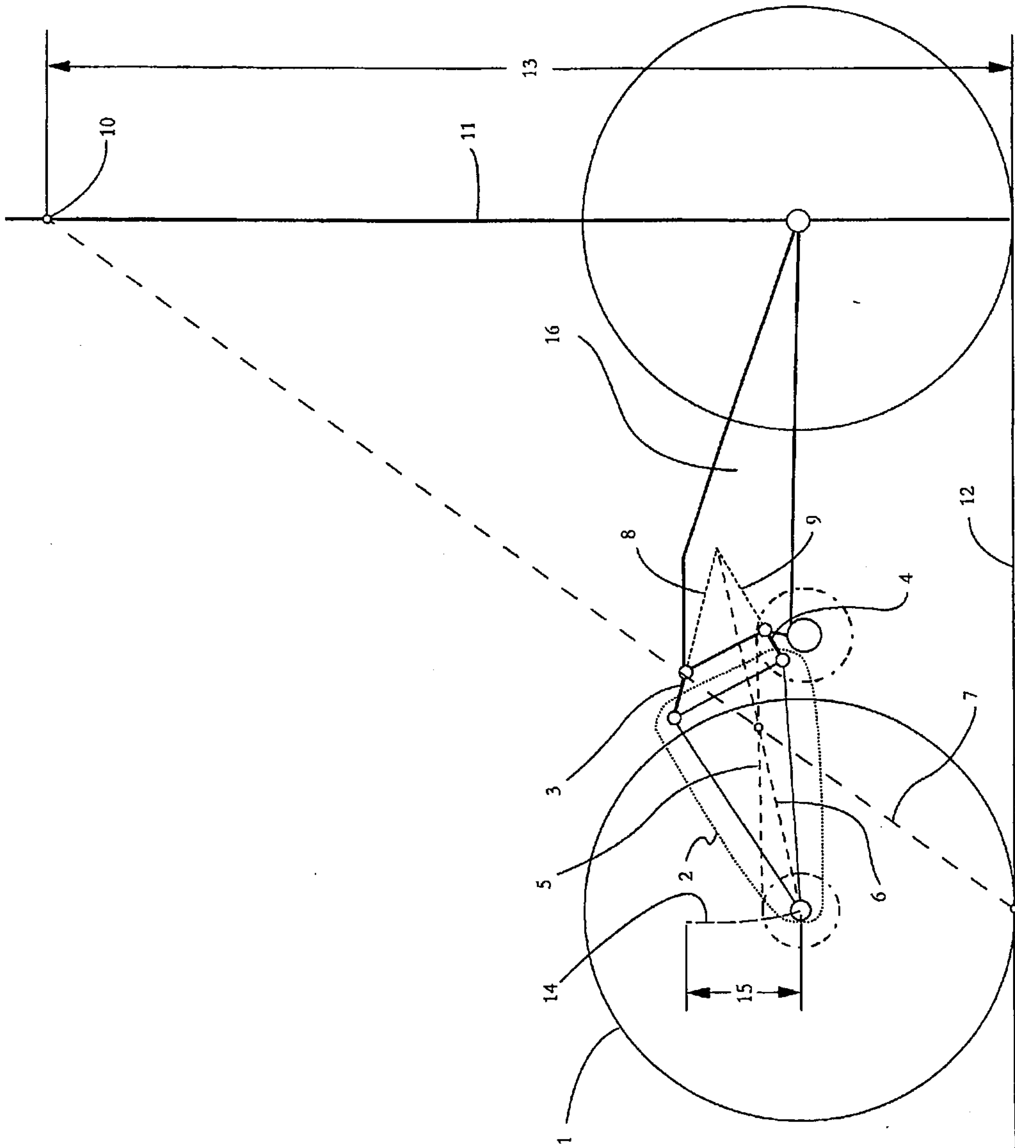


FIGURE 1a

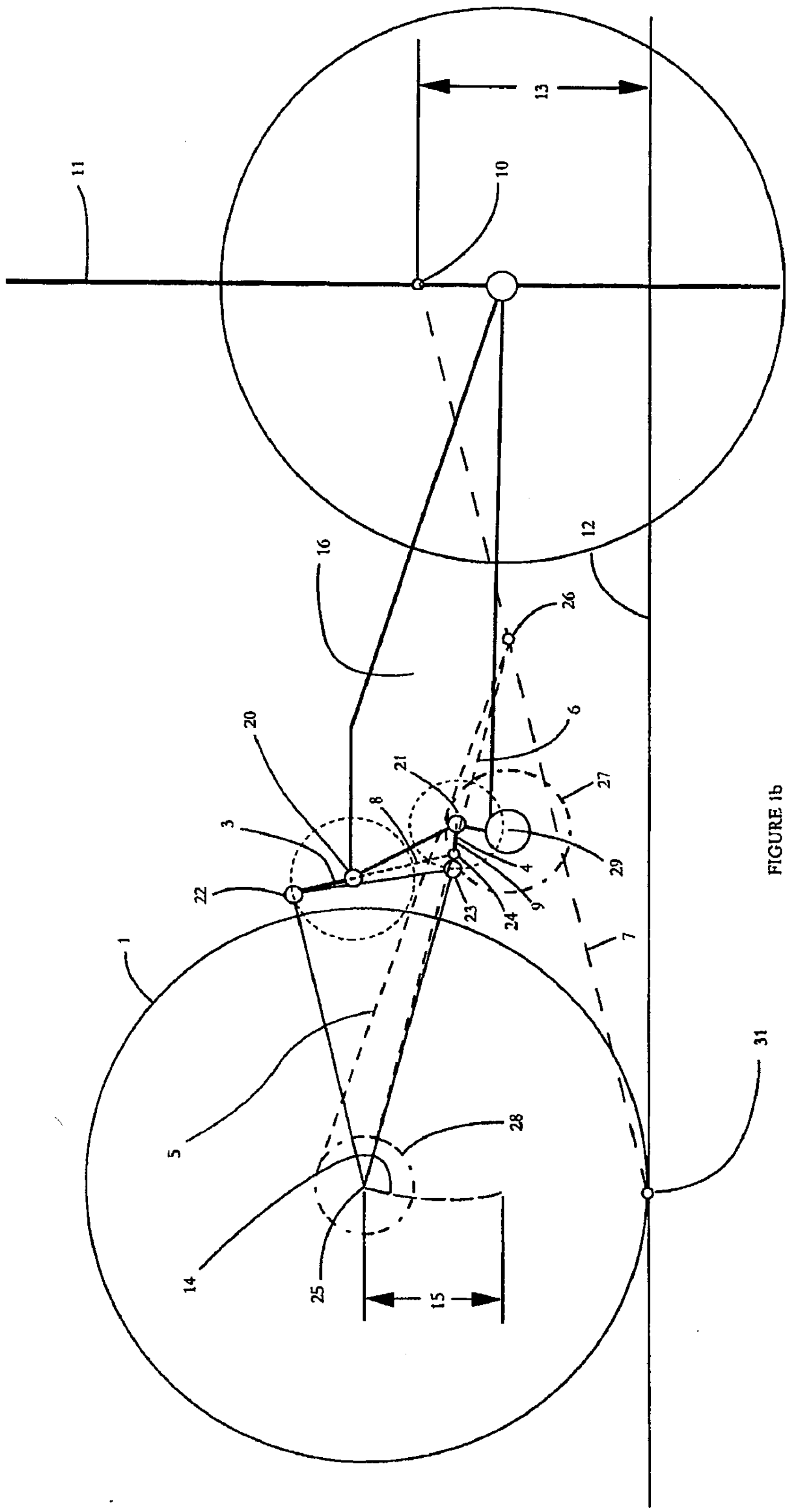


FIGURE 1b

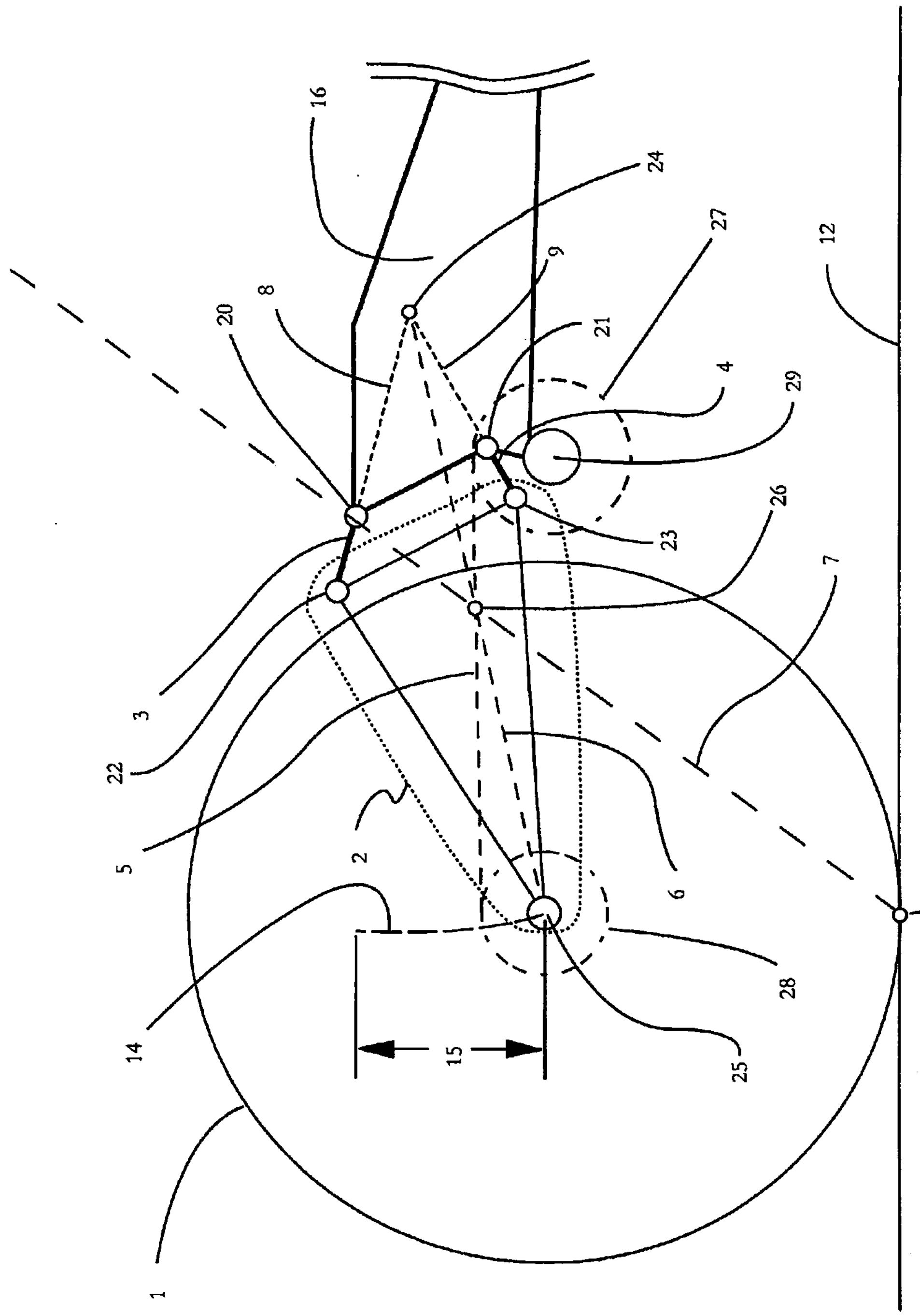


FIGURE 1c

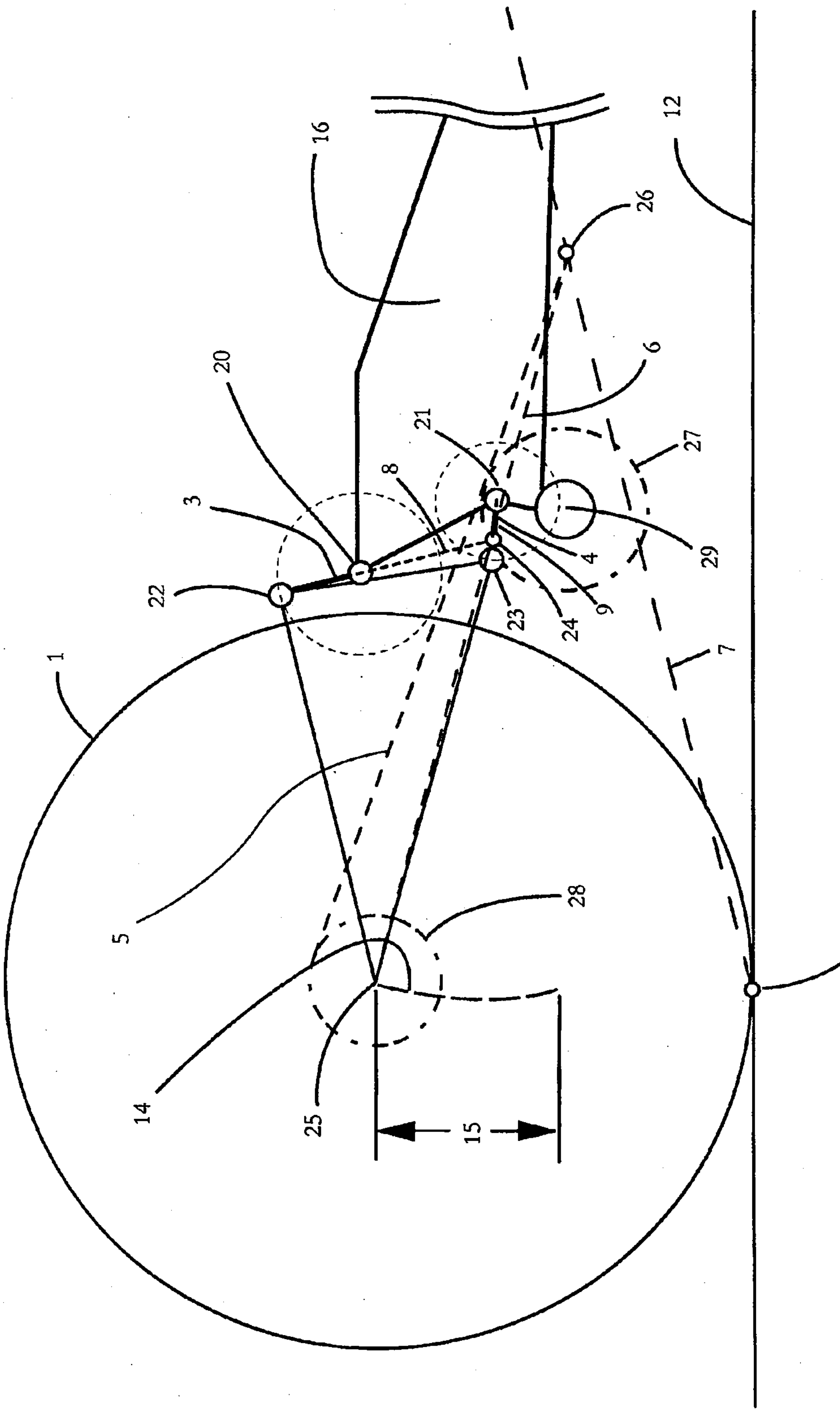


FIGURE 1d

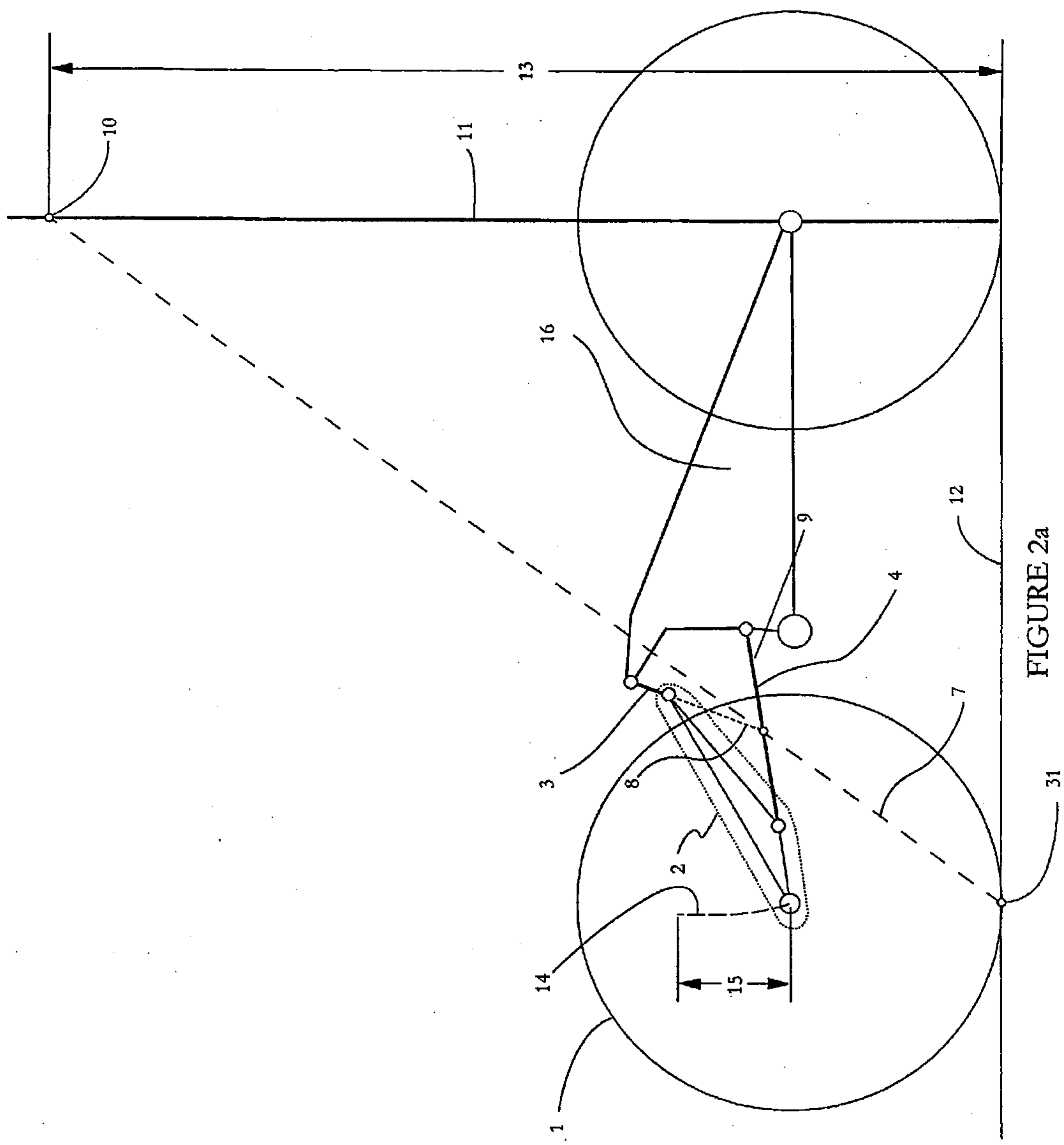


FIGURE 2a

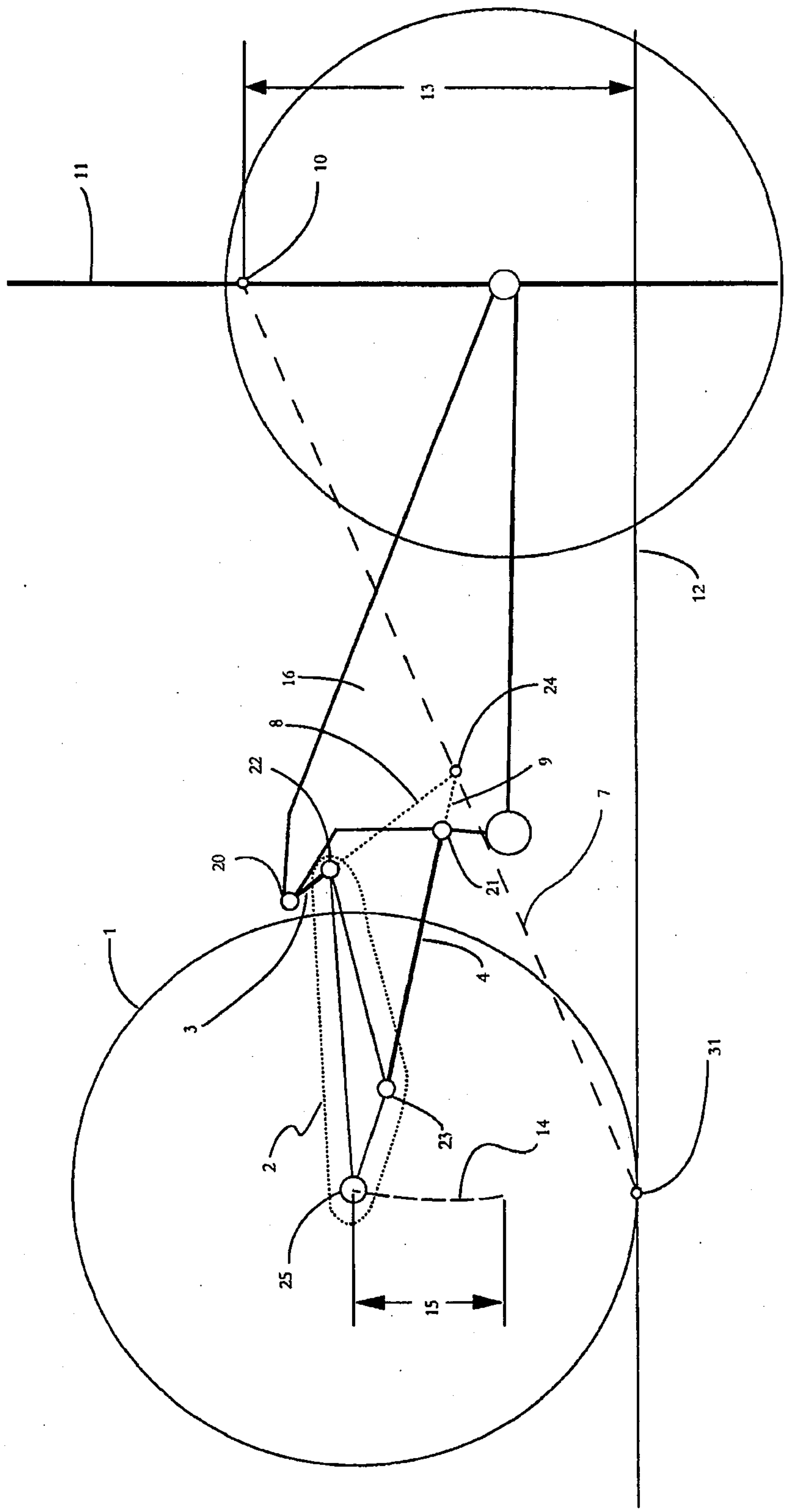


FIGURE 2b

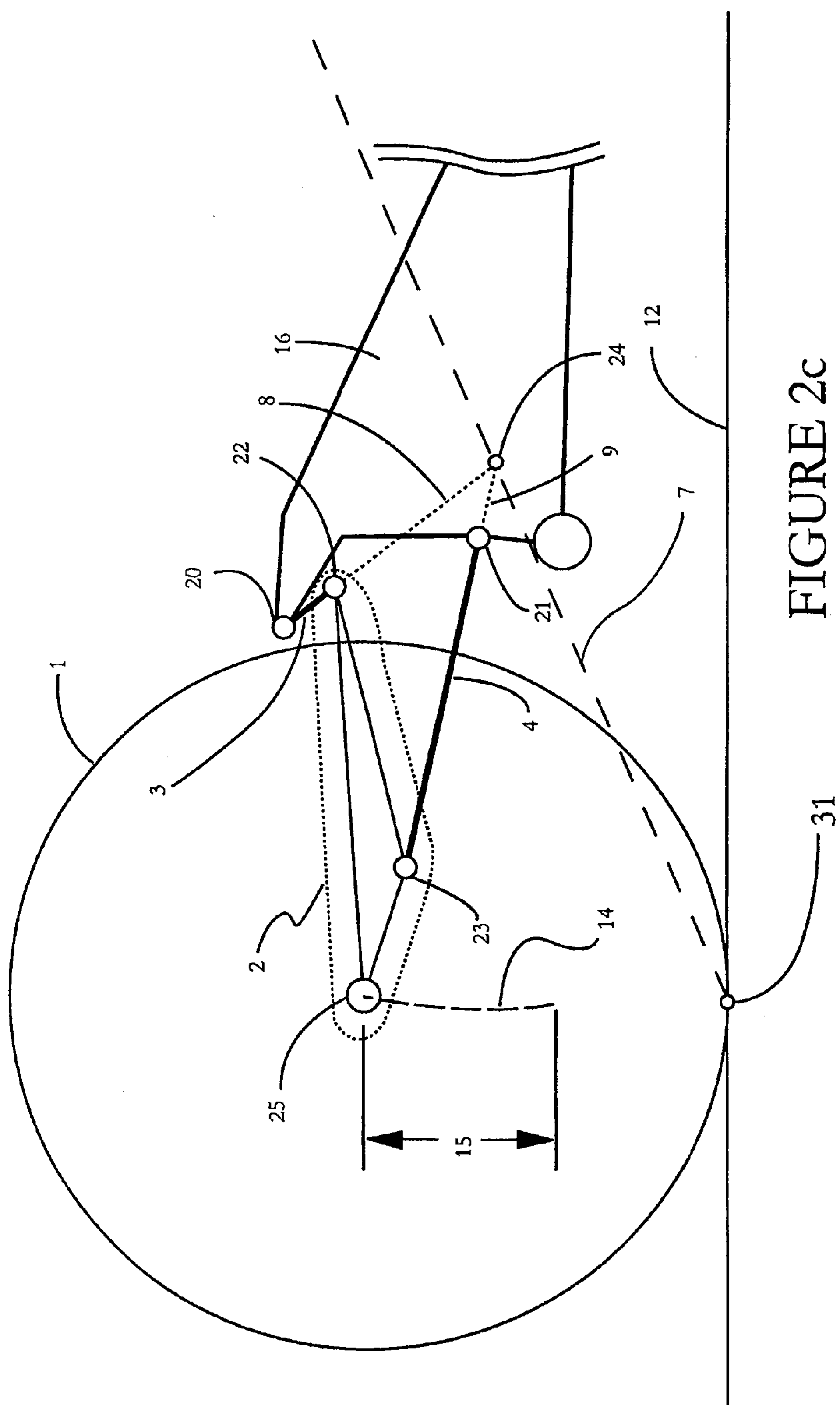


FIGURE 2c

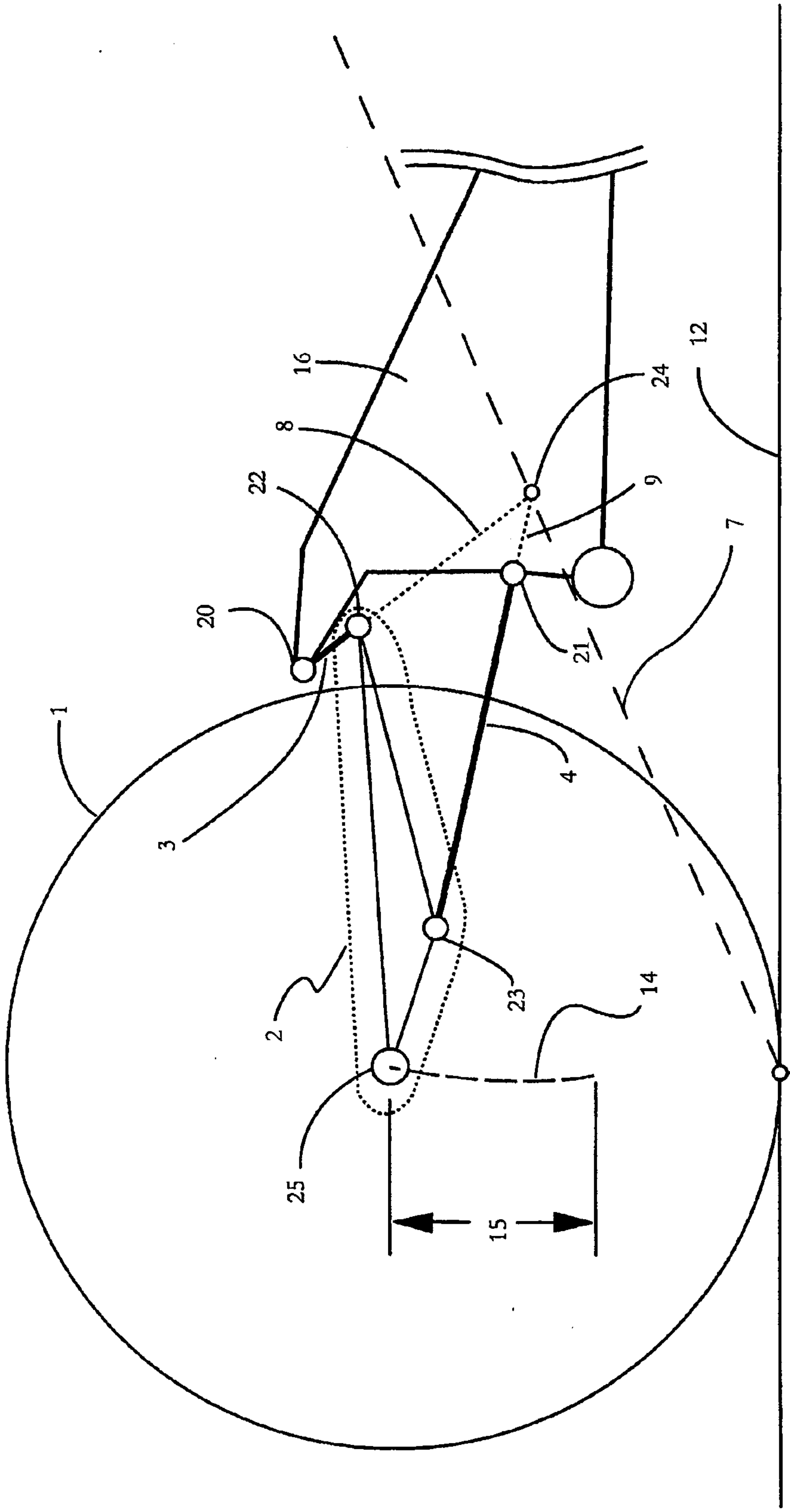
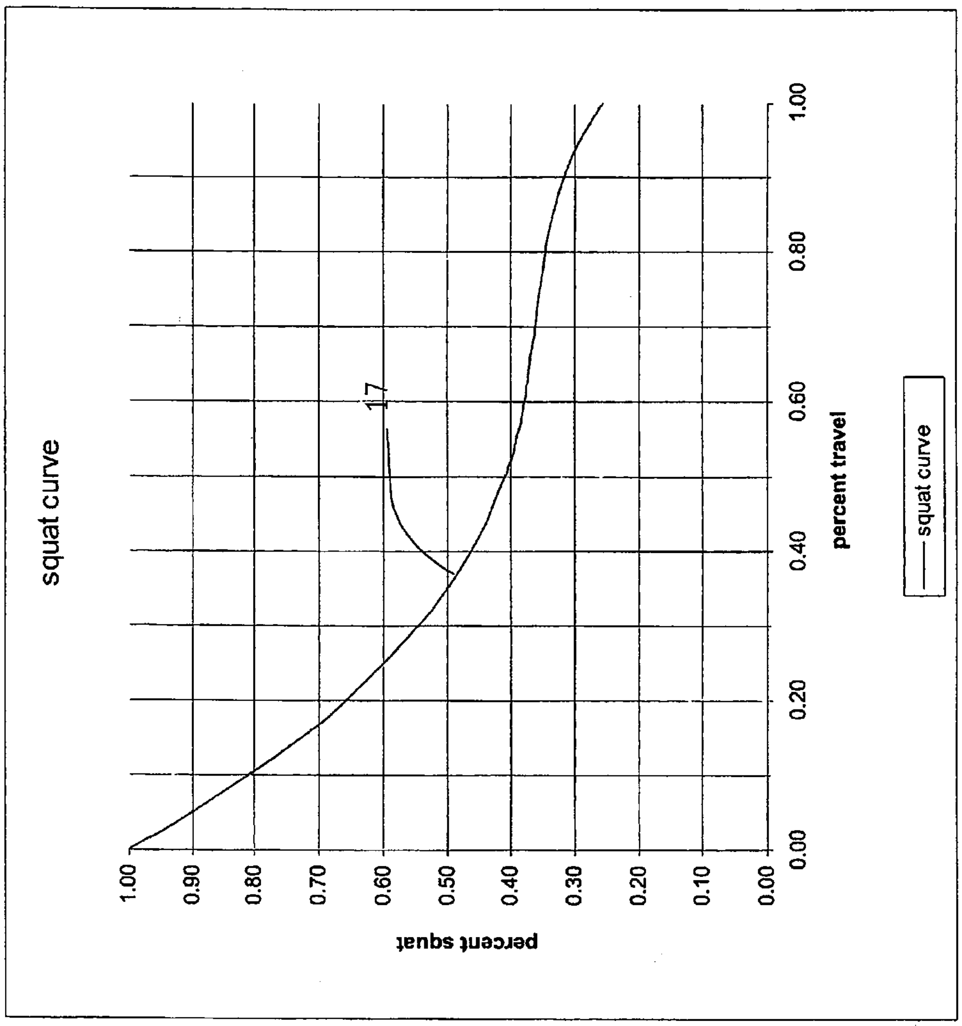


FIGURE 2d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percent squat
percent travel
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0.90
0.80
0.70
0.60
0.50
0.40
0.30
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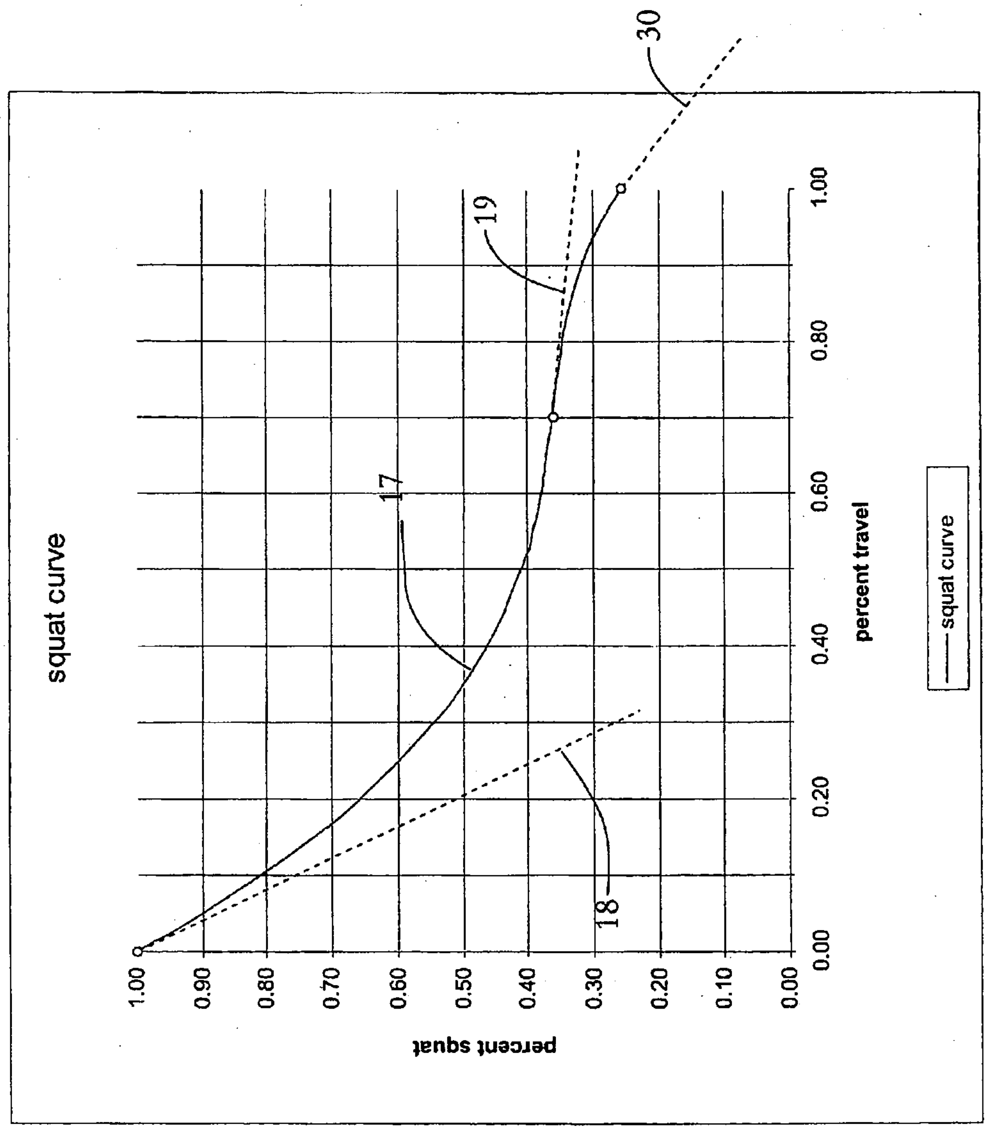
0.00
0.00
0.20
0.40
0.60
0.80
1.00
17
squat curve

FIGURE 3


squat curve
percent squat
percent travel
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0.90
0.80
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0.50
0.40
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0.20
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0.00
0.00
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0.80
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17
18
19
30
squat curve

FIGURE 4

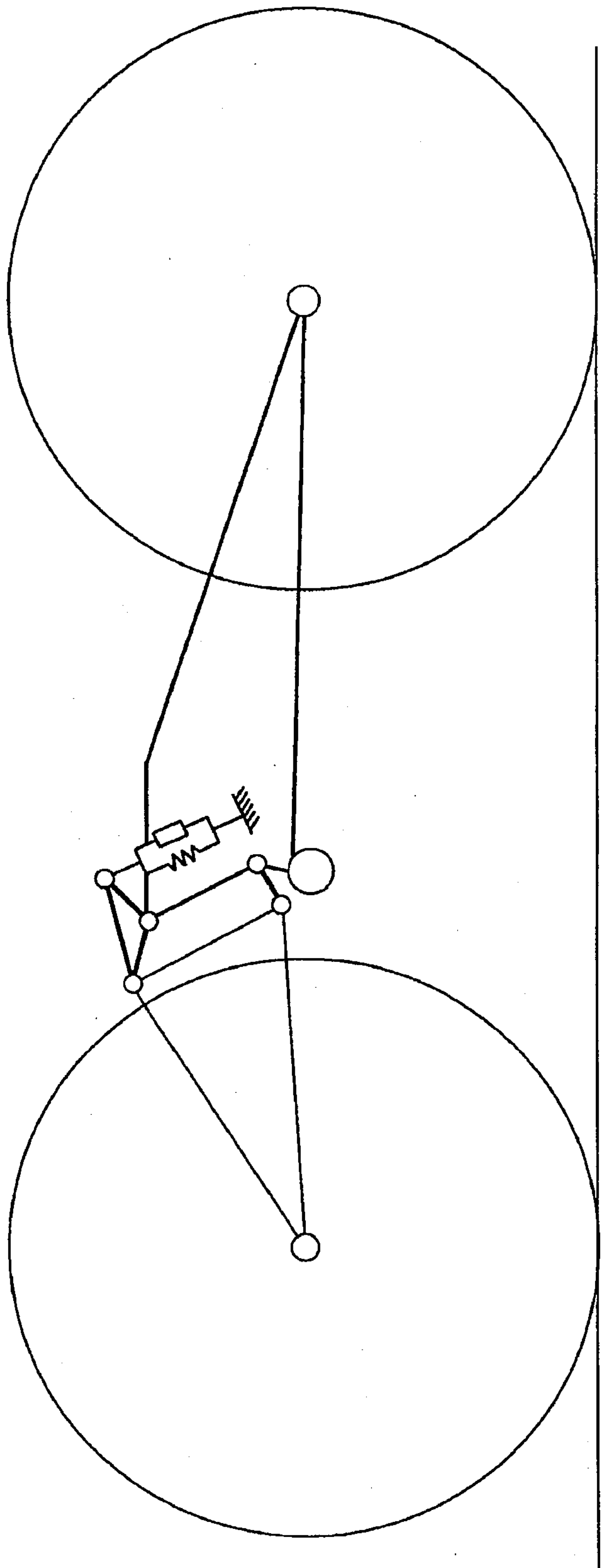

FIGURE 5

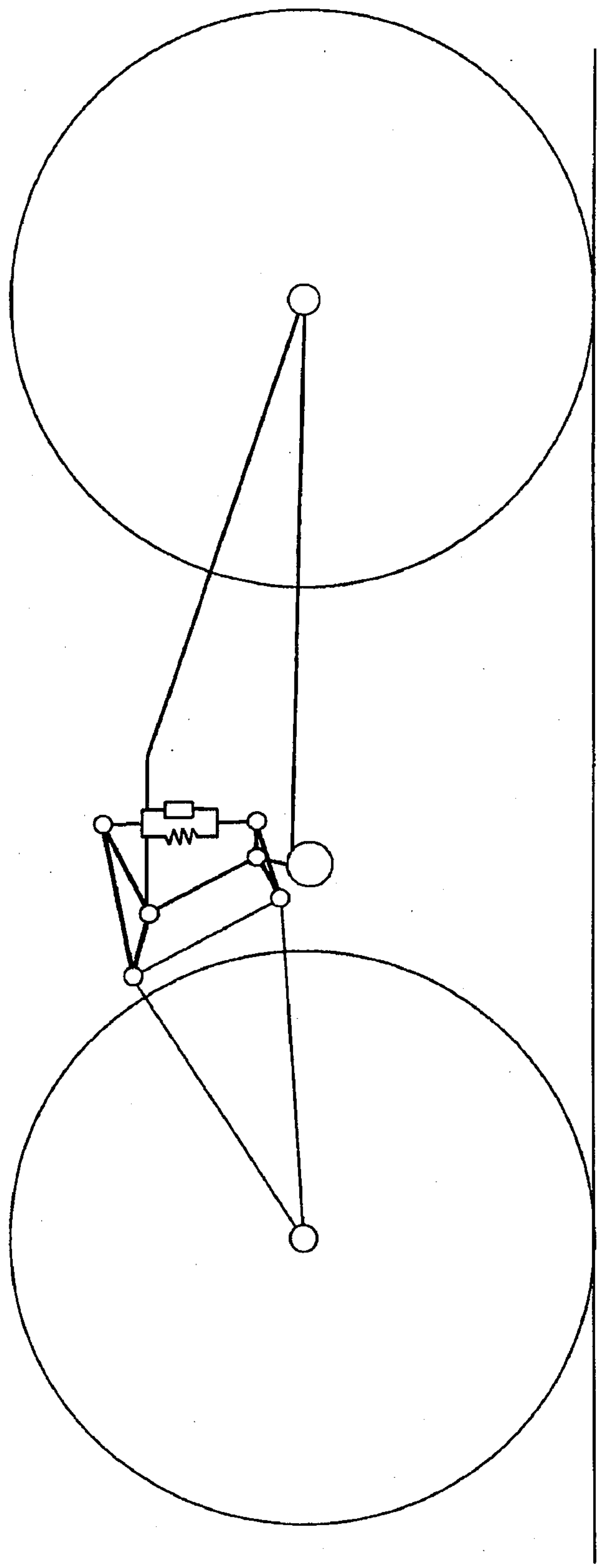

FIGURE 6

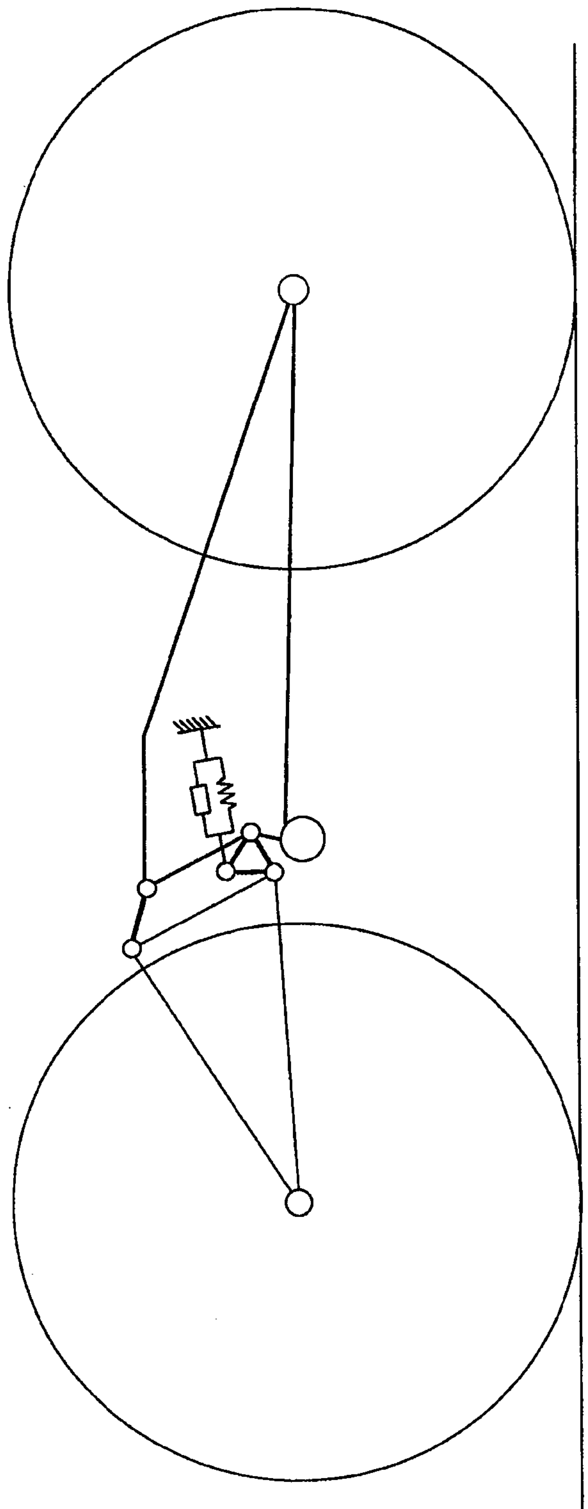

FIGURE 7

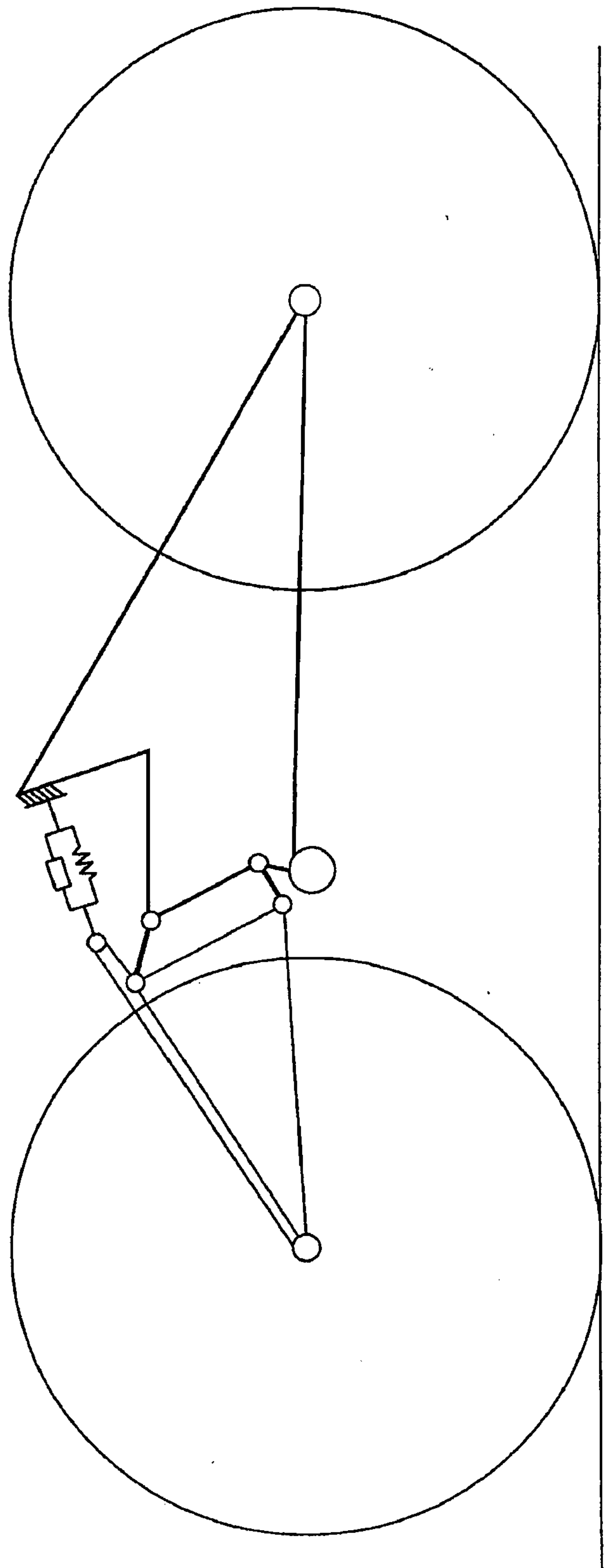

FIGURE 8

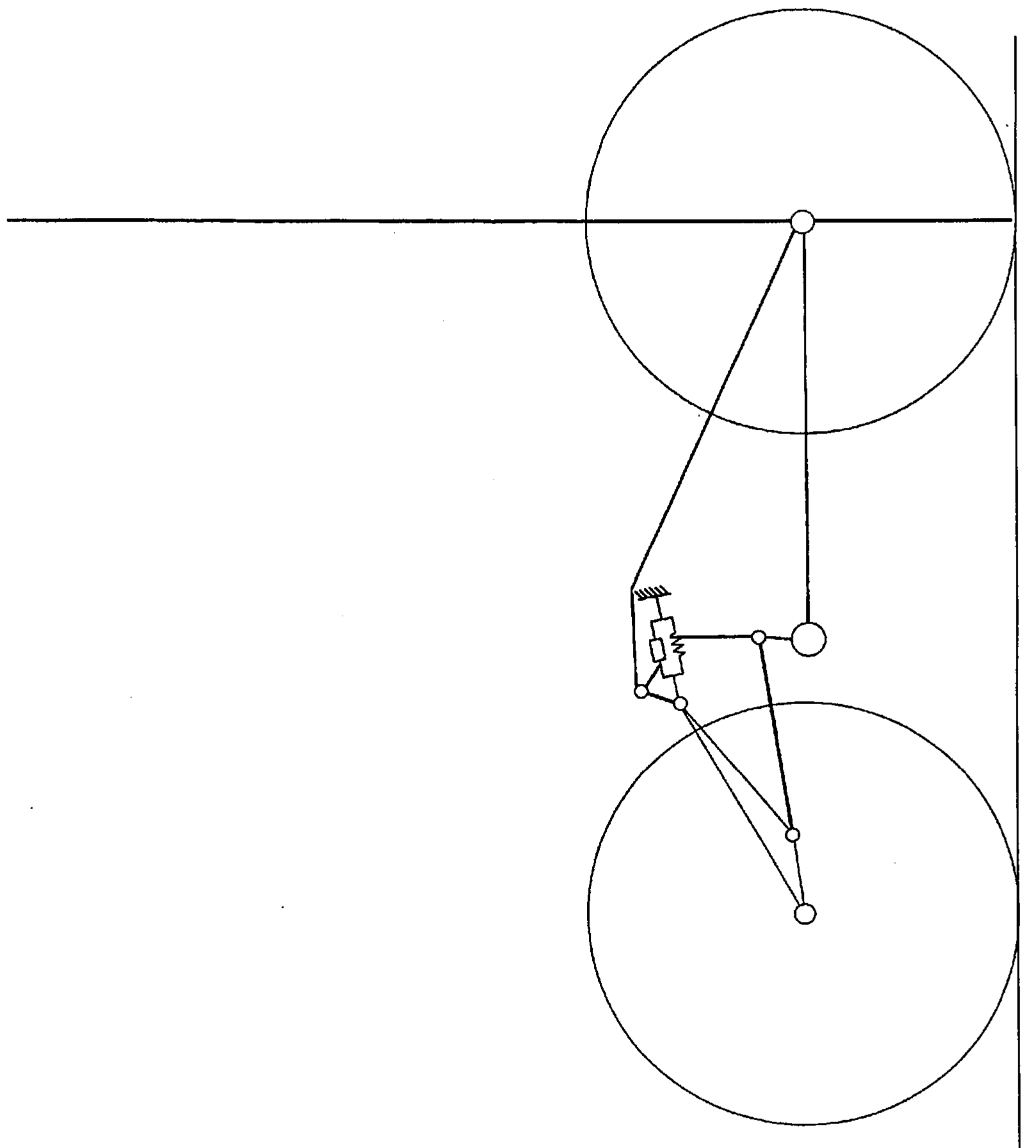

FIGURE 9

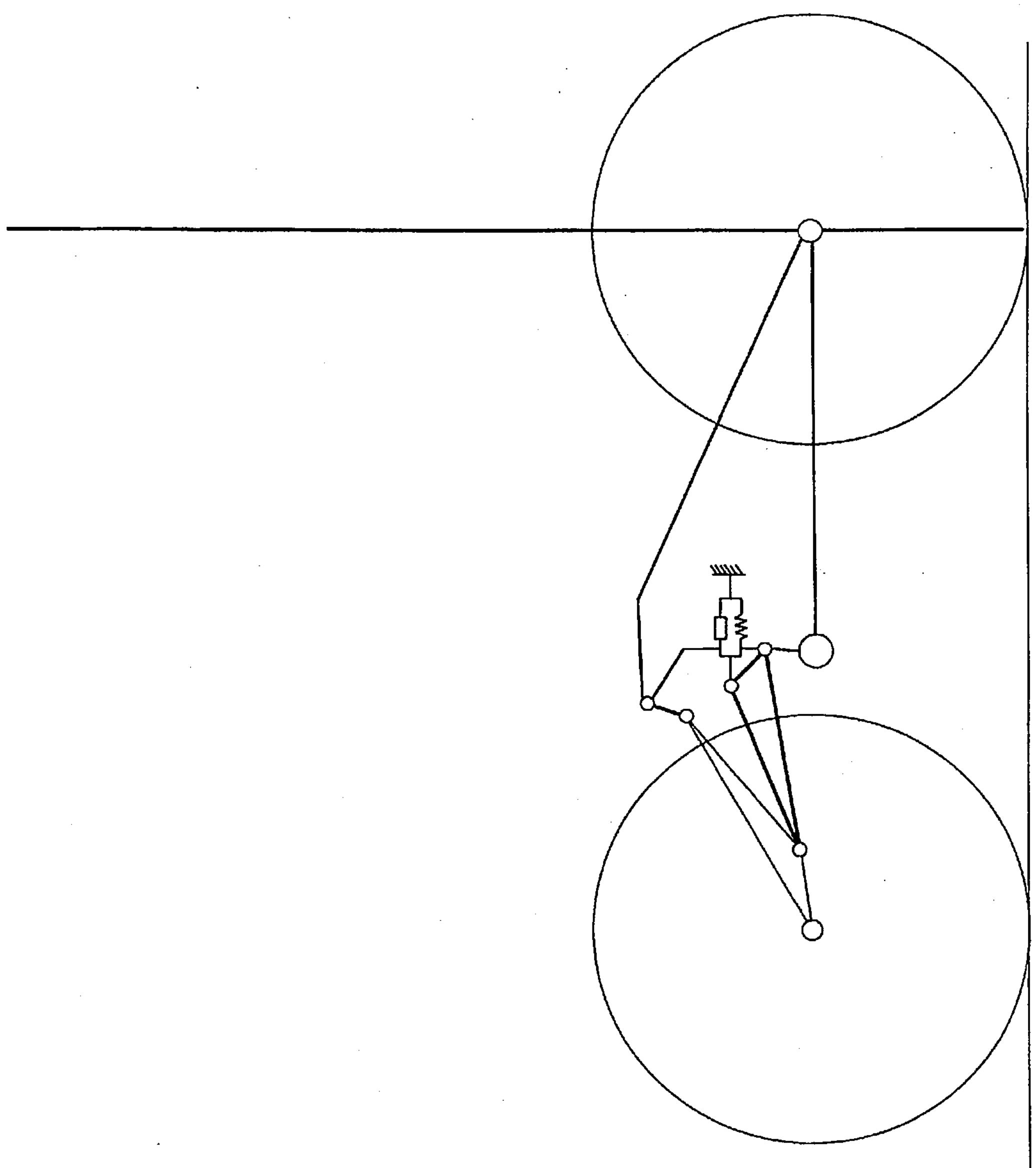

FIGURE 10

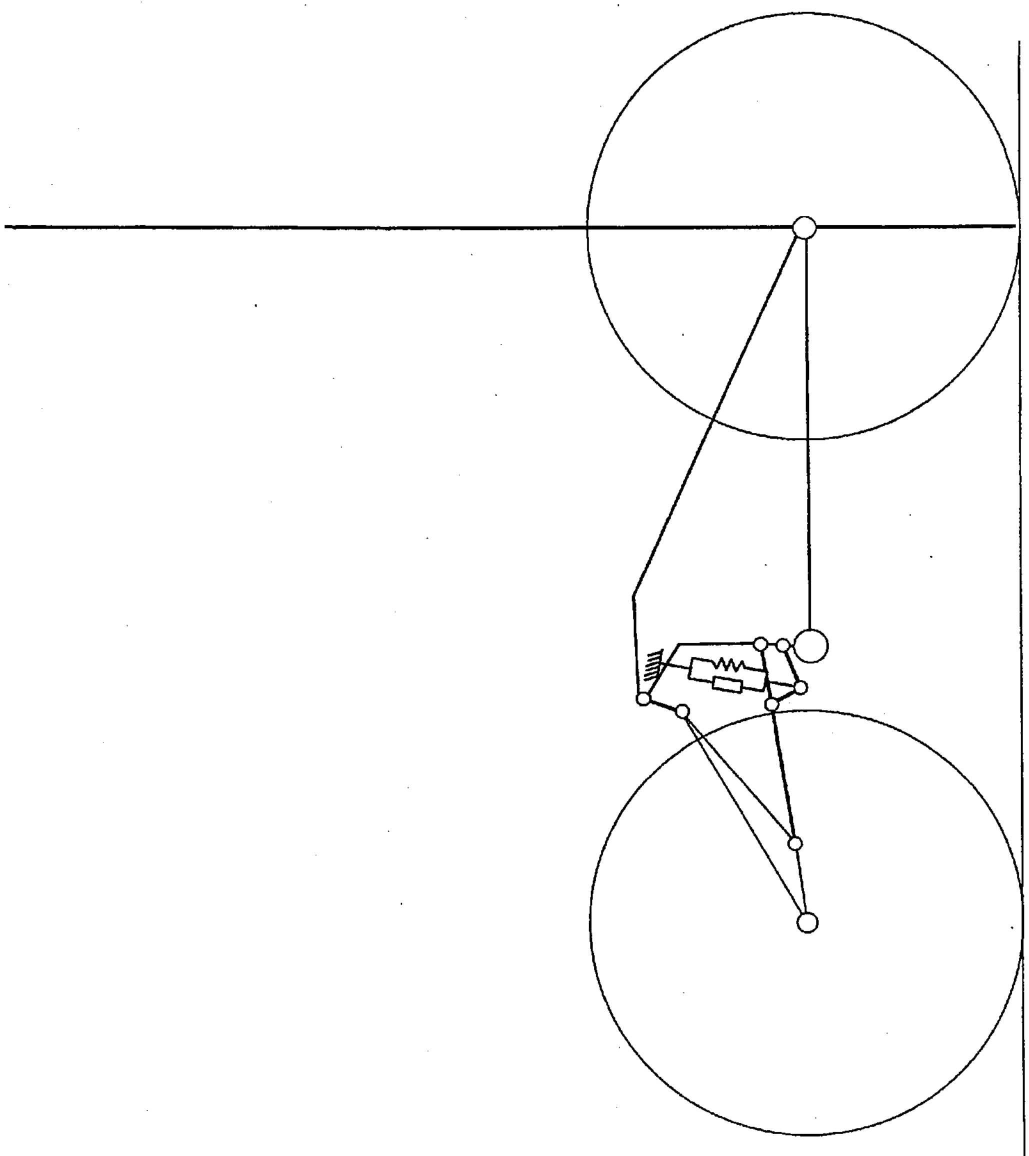

FIGURE 11

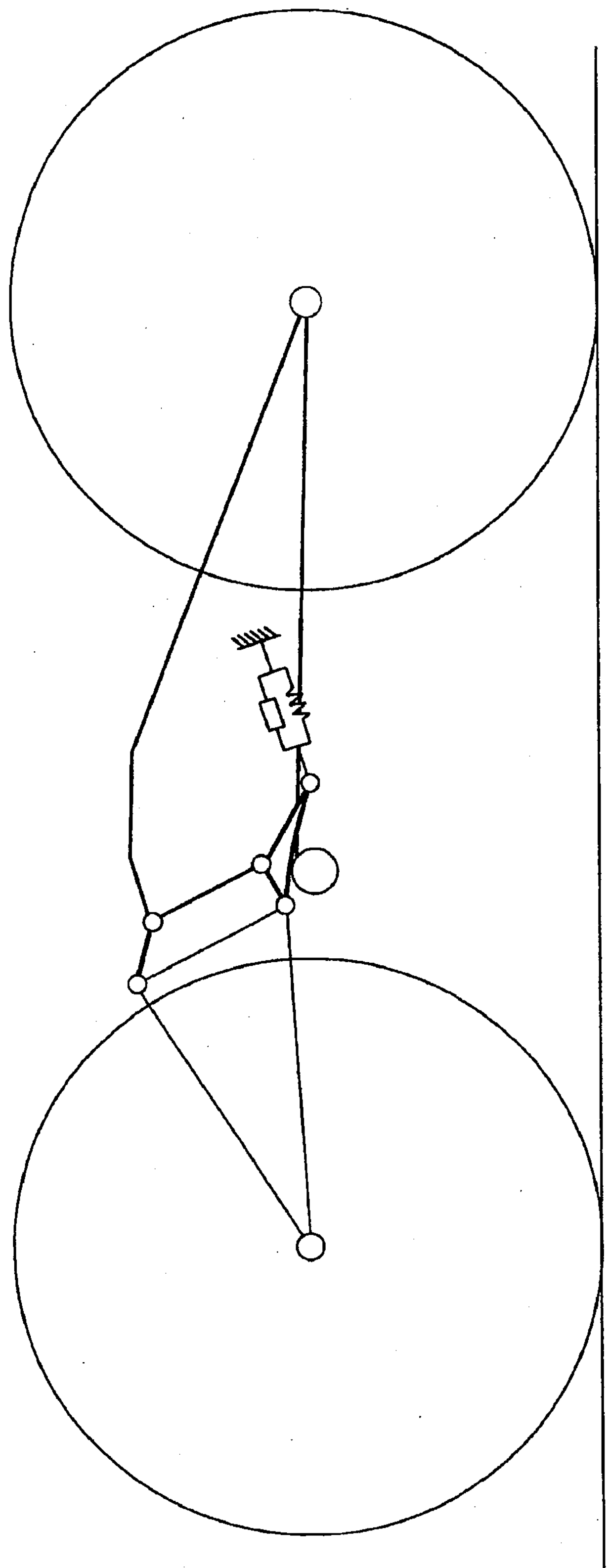

FIGURE 12

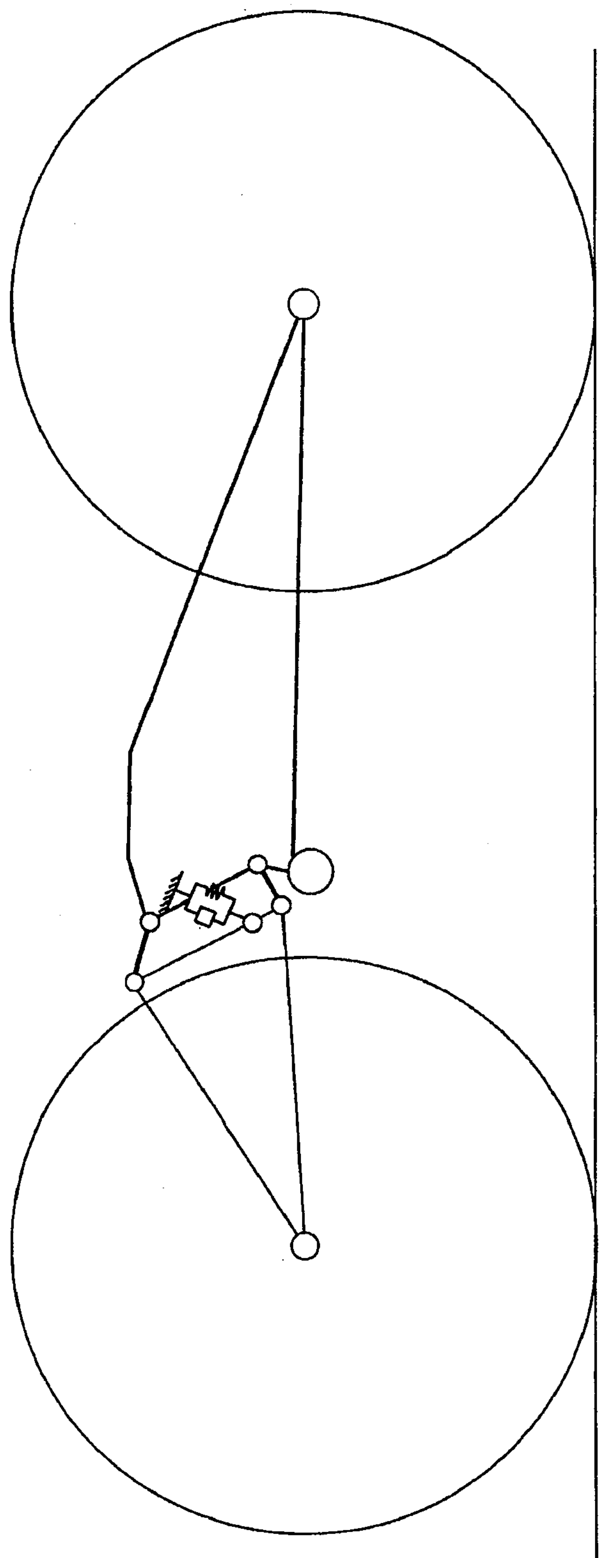

FIGURE 13

VEHICLE SUSPENSION SYSTEMS

This application is a divisional of U.S. application Ser. No. 10/949,264, filed Sep. 24, 2004, now U.S. Pat. No. 7,128,329, which is a continuation-in-part of U.S. application Ser. No. 10/669,412, filed Sep. 25, 2003, now U.S. Pat. No. 7,048,292 which are incorporated herein by reference in their entirety.

BACKGROUND

This invention relates to suspension systems capable of reducing or eliminating a squat response.

Automobiles, bicycles, motorcycles, all terrain vehicles, and other wheel driven vehicles are used for various purposes, including transportation and leisure. These vehicles are designed to use a power source to drive through a power transmission system to a wheel or wheels, which transfers rotary motion to the ground via tractive force between a wheel or wheels and the ground. Vehicles are also used to traverse even terrain like paved streets, and uneven terrain like off-road dirt trails. Off road trails are generally bumpier and allow for less wheel traction than paved roads. A bumpier terrain is best navigated with a vehicle that has a suspension system. A suspension system in a vehicle is aimed to provide a smoother ride for an operator or rider, and increase wheel traction over varied terrain. Vehicle suspension systems for the front wheel and for the back wheel are available.

One undesirable effect of suspension systems is the loss of energy in the way of suspension compression or extension during powered acceleration. Such energy loss is particularly notable in vehicles that are driven by low energy power sources, for example, bicycles and solar vehicles. For example, the average rider of a bicycle can exert only a limited amount of power or energy for a short period of time and an even lesser amount for an extended period of time. Therefore, even a very small power loss can have a significant effect on rider performance and comfort. Suspension travel is the distance a suspended wheel travels when the suspension is moved from a fully extended state to a fully compressed state. In bicycles, suspension travel has been increased for many designs and with these increases in suspension travel; the aforementioned energy loss has become even more apparent to riders. But even for a vehicle with a high power energy source, any loss in energy reduces the vehicle's efficiency, for example its fuel efficiency. Where vehicles are used in a manner that requires frequent accelerations, including positive and negative accelerations, the efficiency of the vehicle is particularly affected by any loss of energy resulting from the vehicles geometry, including the geometry and design of its suspension systems.

Thus, by minimizing energy loss resulting from the design of a vehicle's suspension system, the efficiency of the vehicle is improved and thereby its environmental impact. The need for a suspension system that can better preserve a vehicles efficiency and energy has therefore become more pressing. The present invention provides suspension system designs for vehicles that reduce these energy losses.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1***a* is a side view of a chain driven vehicle using a driven wheel suspension system that achieves a squat curve according to certain embodiments of the current invention. The vehicle is shown with the driven wheel suspension system in an uncompressed state.

FIG. **1***b* is a side view of a chain driven vehicle as shown in FIG. **1***a* with the driven wheel suspension system in a completely compressed state.

FIG. **1***c* is an enlarged section of the side view of the chain driven vehicle shown in FIGS. **1***a* and **1***b* with the driven wheel suspension system in a completely uncompressed state.

FIG. **1***d* is an enlarged section of the side view of the chain driven vehicle shown in FIGS. **1***a*, **1***b*, and **1***c* with the driven wheel suspension system in a completely compressed state.

FIG. **2***a* is a side view of a shaft driven vehicle using a driven wheel suspension system that achieves a squat curve according to certain embodiments of the current invention. The vehicle is shown with the driven wheel suspension system in an uncompressed state.

FIG. **2***b* is a side view of a shaft driven vehicle as shown in FIG. **2***a* with the driven wheel suspension system in a completely compressed state.

FIG. **2***c* is an enlarged section of the side view of the shaft driven vehicle shown in FIGS. **2***a* and **2***b* with the driven wheel suspension system in a completely uncompressed state.

FIG. **2***d* is an enlarged section of the side view of the shaft driven vehicle shown in FIGS. **2***a*, **2***b*, and **2***c* with the driven wheel suspension system in a completely compressed state.

FIGS. **3** and **4** show squat curves for suspension systems according to certain embodiments of the invention graphed on a squat curve graph as disclosed herein.

FIGS. **5-13** show alternative embodiments of suspension systems comprising a squat curve of the invention. Each embodiment shown includes a spring/damper unit (small irregular box) and different frame members (thicker lines) interconnected through pivots (small circles).

SUMMARY OF THE INVENTION

The current invention relates to new suspension systems for vehicles, for example, bicycles, motorcycles, cars, SUVs, trucks, two wheel vehicles, four wheel vehicles, front wheel suspension vehicles, driven wheel suspension vehicles, and any other kind of vehicle with a suspension system. In certain embodiments of the invention, a suspension system of the invention is capable of facilitating a squat response that lowers the energy loss resulting from squat. In certain preferred embodiments, a suspension system of the invention is capable of lowering energy loss resulting from squat by producing an anti-squat response. An anti-squat response of a suspension system of the invention, in certain embodiments, varies along suspension travel of the vehicle and preferably is higher at the beginning of suspension travel and less thereafter.

Certain embodiments of the invention comprise a wheel suspension design that uses a tuned squat response to reduce powered acceleration induced suspension movement at tactical points during the driven wheel suspension travel. A vehicle designed to use the preferred embodiment of the invention can accelerate under power with a lower amount of energy loss and a more stable vehicle chassis than known systems.

Suspension systems of the invention are useful for a variety of vehicles and preferably in human powered vehicles. The average rider of a bicycle or other human powered vehicle can exert only a limited amount of power or energy for a short period of time and an even lesser amount for an extended period of time. Therefore, even a very small power loss can have a significant detrimental effect on rider performance and comfort. The need for a suspension system that can better preserve the rider's energy has therefore become more pressing. The present invention provides suspension system designs for vehicles that reduce energy loss during powered acceleration.

In certain embodiments of the invention, a wheel suspension system comprises a wheel connected to a wheel carrier unit and said wheel carrier unit connected to spring damper means; and isolating said wheel from a frame structure with the wheel suspension system having an squat curve with said squat curve having a decreasing rate of squat as the suspension system moves from a beginning point in the wheel travel to an ending point in the wheel travel.

In certain embodiments of the invention, a compressible wheel suspension system comprises a wheel connected to a wheel carrier unit and said wheel carrier unit connected to spring damper means; and isolating said wheel from a frame structure with the wheel suspension system having a squat curve with said squat curve having a decreasing squat amount and without said squat amount increasing as the suspension system moves from a beginning point in the wheel travel towards an ending point in the wheel travel increase.

In certain embodiments of the invention, a compressible vehicle suspension system comprises a defined squat curve, with said squat curve having a maximum value at the lowest amount of suspension compression, and a minimum value at a further point in the travel, and a continuously decreasing amount of squat throughout the wheel travel.

In certain embodiments of the invention, a vehicle suspension system comprises a defined squat curve, with said squat curve having a slope that is generally negative at an earlier point in the suspension travel, and a slope that is less negative at a interim point in the suspension travel, and a slope that is then more negative at a latter point in the suspension travel.

In certain embodiments of the invention, a compressible wheel suspension system comprises a wheel connected to a wheel carrier unit and said wheel carrier unit connected to a top link and a bottom link, with a top link connected to spring damper means; With said top and bottom links rotating together in a clockwise direction, and said top and bottom links connecting said wheel carrier to a frame structure, isolating said wheel from the frame structure. Said top link and said bottom link having projected link force lines and said top link projected force line intersecting said lower link projected force line at a point in the beginning of the suspension travel and said top link projected force line intersecting said lower link at a point later in the travel.

In certain embodiments of the invention, a compressible wheel suspension system comprises a wheel connected to a wheel carrier unit and said wheel carrier unit connected to a top link and a bottom link, with said wheel carrier connected to spring damper means; with said top and bottom links rotating together in a clockwise direction, and said top and bottom links connecting said wheel carrier to a frame structure, isolating said wheel from the frame structure. Said top link and said bottom link having projected link force lines and said top link projected force line intersecting said lower link projected force line at a point in the beginning of the suspension travel and said top link projected force line intersecting said lower link at a point later in the travel.

In certain embodiments of the invention, a compressible wheel suspension system comprises a wheel connected to a wheel carrier unit and said wheel carrier unit connected to a top link and a bottom link, with said bottom link connected to spring damper means; with said top and bottom links rotating together in a clockwise direction, and said top and bottom links connecting said wheel carrier to a frame structure, isolating said wheel from the frame structure, said top link and said bottom link having projected link force lines and said top link projected force line intersecting said lower link projected force line at a point in the beginning of the suspension travel and said top link projected force line intersecting said lower link at a point later in the travel.

In certain embodiments of the invention, a compressible wheel suspension system comprises a wheel connected to a wheel carrier unit and said wheel carrier unit connected to a top link and a bottom link, with said top and bottom links connected to spring damper means; with said top and bottom links rotating together in a clockwise direction, and said top and bottom links connecting said wheel carrier to a frame structure, isolating said wheel from the frame structure. Said top link and said bottom link having projected link force lines and said top link projected force line intersecting said lower link projected force line at a point in the beginning of the suspension travel and said top link projected force line intersecting said lower link at a point later in the travel.

In practice, precisely controlling squat in a suspension system can allow for very little suspension movement during powered acceleration with favorable bump compliance. The further a vehicle suspension is compressed, the higher the spring force at the wheel rotational axis. Most powered acceleration happens within the first 40 percent of the suspension travel. Because spring force is lowest in the beginning of a suspension travel, a suspension is more susceptible to manipulation due to squat forces at that time. If enough anti squat force is not present to inhibit mass transfer in the beginning of the suspension travel, the suspension will compress, and when it rebounds, energy will be lost through the damper. The low spring force in the beginning of the suspension travel allows for higher levels of movement than at later points in the suspension travel. Minimizing suspension movement due to mass transfer during powered acceleration reduces the amount of damper movement that occurs at that time. With lower amounts of damper movement comes a lower amount of energy that the damper must dissipate, and therefore more of the acceleration force provided by a power source can be used to accelerate the vehicle. The amount of energy consumed to produce enough anti-squat force to reduce movement earlier in the suspension travel is less than the amount of energy that would be lost in the damper during suspension movement. As a driven wheel suspension system is compressed through its travel, spring force increases, and therefore driven wheel resistance to movement increases. At this later point in the suspension travel, because of the increased spring force, squat force has less of manipulating effect on a wheel suspension. A lower amount of anti squat can be used so that more energy can be transferred to forward movement.

DETAILED DESCRIPTION

Vehicles must be accelerated against their environment to propel an operator or rider across terrain. In order to accelerate these vehicles, a certain amount of energy must be exerted and transformed into rotary motion at a wheel or plurality of wheels. Suspended wheeled vehicle energy conversion types are widely varied. Some vehicles like bicycles, tricycles, and pedal cars use converted human energy as the drive unit. Other vehicles use electric motors or combustion engines, as their drive unit. These electric motors and combustion engines extract rotary motion through the controlled release of chemically stored energy.

Almost all vehicle types use some sort of rotary motion transmission system to transfer rotational force from a drive unit to a wheel or plurality of wheels. A simple bicycle or motorcycle or all terrain vehicle uses a chain or belt to transfer power from a drive unit to a wheel. These chain or belt drive transmissions typically use one sprocket in the front which is coupled to a drive system and one sprocket in the rear which is coupled to a wheel.

More complex bicycles, motorcycles, all terrain vehicles, and automobiles use a shaft drive system to transfer power from a drive system to a driven wheel or wheels. These shaft drive systems transfer power through a rotating shaft that is usually reasonably perpendicular to the driven wheel spinning axis, with power transferred to the driven wheel via a bevel, spiral bevel, hypoid, worm gear drivetrain, or some other means. These single sprocket chain and belt, and shaft driven vehicles can use a direct driven single speed arrangement, where drive unit output shaft speed and torque is transferred to the driven wheel at a constant unchanging ratio. These single sprocket chain and belt, and shaft driven vehicles can also use a commonly found multi speed arrangement, where drive unit output shaft speed and torque is transferred to the driven wheel at a variable ratio through operator selected or automatically selected ratio changing mechanisms.

A bicycle with a more advanced design includes gear changing systems that have clusters of selectable front chainrings and rear sprockets. These gear changing systems give the bicycle rider a selectable mechanical advantage for use during powered acceleration. The mechanical advantage selection, allows a rider spinning a front sprocket cluster via crank arms, to attain lower revolution speed and higher torque values, or conversely, higher revolution speed and lower torque values at a driven wheel.

The current invention, in certain embodiments, is directed at suspension systems that can maintain low energy loss under powered acceleration of the vehicle, for example, a bicycle, a motorcycle, a car, an SUV, a truck, or any other kind of vehicle. Suspension systems of the current invention are useful for a large variety of vehicles, including, but not limited to, human powered vehicles, off road use vehicles with long displacement suspension, high efficiency road going vehicles, and other vehicles.

A vehicle suspension system isolates a vehicle chassis from forces imparted on the vehicle when traversing terrain by allowing the vehicle's ground contact points to move away from impacts at the terrain level and in relation to the vehicle chassis by a compressible suspension movement. The compressible suspension movement that isolates a chassis from these impacts is called suspension displacement or suspension travel. Compressible suspension travel has a beginning point where the suspension is in a completely uncompressed state, and an ending point of displacement, where the suspension is in a completely compressed state. Suspension travel displacement is measured in a direction parallel to and against gravity. In certain preferred embodiments, a suspension system of the invention uses a tuned squat curve to provide an amount of squat closer to or higher in the range of the squat condition known as anti squat in the beginning of a suspension travel displacement, and an amount of squat closer to the range of the squat condition known as anti squat than the initial measurement at a later point in the suspension travel displacement. As a suspension system of the invention is compressed, a spring or damper unit is compressed. As this spring or damper unit is compressed, the force output from the unit rises. As the suspended wheel moves through its axle path, spring force at the wheel rises. A suspended wheel has a compressible wheel suspension travel distance that features a beginning travel point where the suspension is completely uncompressed to a point where no further suspension extension can take place, and an end travel point where a suspension is completely compressed to a point where no further suspension compression can take place. At the beginning of the wheel suspension travel distance, when the suspension is in a completely uncompressed state, the spring is in a state of least compression, and the suspension is easily compressed. In certain preferred embodiments, a higher amount of anti squat in the beginning of the suspension travel is needed to keep the suspension from compressing due to mass transfer under acceleration. As the suspension compresses, spring force at the wheel rises. When spring force rises to levels present in the middle of the suspension travel, mass transfer due to acceleration has a much smaller effect on vehicle traction or chassis attitude because the mass transfer is not capable of significantly compressing the suspension system. At this point, in certain preferred embodiments, the present invention decreases anti squat amount so that a greater amount of mass transfer towards the wheel can occur. This mass transfer allows increased wheel traction while transferring more energy towards forward propulsion.

FIG. **1***a* shows certain embodiments of the invention and it presents a graphical method useful to attain a squat point measurement, and a graphical method to attain suspension geometry kinematical layout from an existing desired measured squat point. Shown in FIG. **1***a* are the following: driven wheel (**1**); swinging wheel carrier link (**2**); upper carrier manipulation link (**3**); lower carrier manipulation link (**4**); chain force vector (**5**); driving force vector (**6**); squat force vector (**7**); upper carrier manipulation link force vector (**8**); lower carrier manipulation link force vector (**9**); squat definition point (**10**); squat layout line (**11**); lower squat measurement definition line (**12**); measured squat distance (**13**); driven wheel axle path (**14**); driven wheel suspension travel distance (**15**); vehicle chassis (**16**); center of the driven wheel tire to ground contact patch (**31**).

FIG. **1***a* exemplifies that as the driven wheel **1** suspension system is completely uncompressed in its driven wheel suspension travel distance **15**, its squat force vector **7** is shown in relation to the vehicle chassis **16**. The squat force vector's **7** measured squat distance **13** which is measured as the perpendicular distance between the lower squat measurement definition line **12** and the squat definition point **10**, is also shown in FIG. **1***a*. As the suspension system is compressed through its driven wheel suspension travel distance **15**, change in measured squat distance **13** over the driven wheel suspension travel distancel5 is used to create a squat curve **17**. FIG. **1***b* shows a side view of a chain driven vehicle as shown in FIG. **1***a* with the driven wheel suspension system in a completely compressed state. Certain embodiments are further exemplified, for example, vectors (also known as force-lines) useful to a graphical method to attain a squat point measurement are shown. The term "vector", as used in this specification and the appended claims does not refer to the mathematical term, but to lines showing directions of forces. Also exemplified is a graphical method useful to attain suspension geometry kinematical layout from an existing desired measured squat point. Shown in FIG. **1***b* in addition to what is presented in FIG. **1***a*, are the following: upper link fixed pivot (**20**); lower link fixed pivot (**21**); upper link floating pivot (**22**); lower link floating pivot (**23**); instant force center (**24**); driven wheel rotation axis (**25**); chain force vector and driving force vector intersection point (**26**); driving cog (**27**); driven cog (**28**); driving cog rotation axis (**29**).

FIG. **1***b* exemplifies that as the driven wheel **1** suspension system is completely compressed through its driven wheel suspension travel distance **15**, its squat force vector **7** moves in relation to the vehicle chassis **16**. The squat force vector's **7** measured squat distance **13**, which is measured as the perpendicular distance between the lower squat measurement definition line **12** and the squat definition point **10**, decreases in relation to the measured squat distance **13** shown in FIG. **1***a*. This change in measured squat distance **13** over the driven wheel suspension travel distance **15**, in certain preferred embodiments, is used to create a squat curve **17**. FIG. **1***b* shows the graphical method used to obtain a squat curve **17** from chain driven vehicle geometry, or chain driven vehicle geometry from a squat curve **17**. In the vehicle shown in FIG. **1***b*, a driven wheel **1** is attached to a swinging wheel carrier link **2**, which pivots at one end of an upper carrier manipulation link **3**. The upper carrier manipulation link **3** is pivotally attached to the vehicle chassis **16** at the upper link fixed pivot **20**. A lower carrier manipulation link **4** is also attached to the swinging wheel carrier link **2**. This lower carrier manipulation link **4** is attached to the vehicle chassis **16** at a lower link fixed pivot **21**. An upper carrier manipulation link force vector **8** is graphed coincident to the swinging wheel carrier link **2** upper pivot and the upper link fixed pivot **20**. The upper carrier manipulation link force vector **8** is graphed so that it intersects a lower carrier manipulation link force vector **9**, which is graphed coincident to the swinging wheel carrier link **2** lower pivot and the lower link fixed pivot **21**. The intersection point of the upper carrier manipulation link force vector **8**, and the lower carrier manipulation link force vector **9** is called the instant force center **24**. A driving force vector **6** is graphed beginning at the driven wheel rotation axis **25**, and passes through the instant force center **24**. A chain force vector **5** is drawn tangent to the tops of the driving cog **27** and driven cog **28**, and intersects the driving force vector **6** at a chain force vector and driving force vector intersection point **26**. The squat force vector **7** is graphed from a beginning point at the center of the driven wheel tire to ground contact patch **31**, and passes through the chain force vector and driving force vector intersection point **26**, before it terminates on a squat layout line **11**. The intersection of the squat force vector **7** and the squat layout line is called the squat layout point **10**. The squat layout line **11** is graphed at a perpendicular angle to gravitational force. A lower squat measurement definition line **12** is graphed beginning at the center of the driven wheel tire to ground contact patch **31** and terminating perpendicular and coincident to the squat layout line **11**. The perpendicular measurement from the lower squat measurement definition line **12** to the squat layout point **10** is called the measured squat distance **13**. This measured squat distance **13** changes as driven wheel suspension travel distance **15** compresses, and is used to create a squat curve **17** in a squat curve graph as shown in FIGS. **3** and **4**.

FIG. **1***c* shows an enlarged section of the side view of the chain driven vehicle shown in FIGS. **1***a* and **1***b* with the driven wheel suspension system in a completely uncompressed state.

FIG. **1***d* shows an enlarged section of the side view of the chain driven vehicle shown in FIGS. **1***a*, **1***b*, and **1***c* with the driven wheel suspension system in a completely compressed state. FIGS. **1***c* and **1***d* further exemplify certain embodiments, for example, points and vectors useful for a graphical method used to attain a squat point measurement, and a graphical method to attain suspension geometry kinematical layout from an existing desired measured squat point.

FIG. **2***a* shows certain embodiments of the invention and it presents a graphical method useful to attain a squat point measurement, and a graphical method to attain suspension geometry kinematical layout from an existing desired measured squat point. Shown in FIG. **2***a* are the following: driven wheel (**1**); swinging wheel carrier link (**2**); upper carrier manipulation link (**3**); lower carrier manipulation link (**4**); squat force vector (**7**); upper carrier manipulation link force vector (**8**); lower carrier manipulation link force vector (**9**); squat definition point (**10**); squat layout line (**11**); lower squat measurement definition line (**12**); measured squat distance (**13**); driven wheel axle path (**14**); driven wheel suspension travel distance (**15**); vehicle chassis (**16**); center of the driven wheel tire to ground contact patch (**31**).

FIG. **2***a* exemplifies that as the driven wheel **1** suspension system is completely uncompressed in its driven wheel suspension travel distance **15**, its defined squat force vector **7** is shown in relation to the vehicle chassis **16**. The squat force vector's **7** measured squat distance **13**, which is measured as the perpendicular distance between the lower squat measurement definition line **12** and the squat definition point **10**, is shown in FIG. **2***a*. As the suspension system is compressed through its driven wheel suspension travel distance **15**, change in measured squat distance **13** over the driven wheel suspension travel distance **15** is used to create a squat curve **17**.

FIG. **2***b* shows a side view of a shaft driven vehicle as shown in FIG. **2***a* with the driven wheel suspension system in a completely compressed state. Certain embodiments are further exemplified, for example, vectors useful to a graphical method to attain a squat point measurement are shown. Also exemplified is a graphical method useful to attain suspension geometry kinematical layout from an existing desired measured squat point. Shown in FIG. **2***b* in addition to what is presented in FIG. **2***a*, are the following: upper link fixed pivot (**20**); lower link fixed pivot (**21**); upper link floating pivot (**22**); lower link floating pivot (**23**); instant force center (**24**); driven wheel rotation axis (**25**); chain force vector and driving force vector intersection point (**26**); driving cog (**27**); driven cog (**28**); driving cog rotation axis (**29**).

FIG. **2***b* exemplifies that as the driven wheel **1** suspension system is completely compressed through its driven wheel suspension travel distance **15**, its defined squat force vector **7** moves in relation to the vehicle chassis **16**. The squat force vector's **7** measured squat distance **13** which is measured as the perpendicular distance between the lower squat measurement definition line **12** and the squat definition point **10**, decreases in relation to the measured squat distance **13** shown in FIG. **2***a*. This change in measured squat distance **13** over the driven wheel suspension travel distance **15** is used to create a squat curve **17**. FIG. **2***b* shows the graphical method used to obtain a squat curve **17** from shaft driven vehicle geometry, or shaft driven vehicle geometry from a squat curve **17**. In the vehicle shown in FIG. **2***b*, a driven wheel **1** is attached to a swinging wheel carrier link **2**, which pivots at one end of an upper carrier manipulation link **3**. The upper carrier manipulation link **3** is pivotally attached to the vehicle chassis **16** at the upper link fixed pivot **20**. A lower carrier manipulation link **4** is also attached to the swinging wheel carrier link **2**. This lower carrier manipulation link **4** is attached to the vehicle chassis **16** at a lower link fixed pivot **21**. An upper carrier manipulation link force vector **8** is graphed coincident to the swinging wheel carrier link **2** upper pivot and the upper link fixed pivot **20**. The upper carrier manipulation link force vector **8** is graphed so that it intersects a lower carrier manipulation link force vector **9**, which is graphed coincident to the swinging wheel carrier link **2** lower pivot and the lower link fixed pivot **21**. The intersection point of the upper carrier manipulation link force vector **8**, and the lower carrier manipulation link force vector **9** is called the instant force center **24**. The squat force vector **7** is graphed from a beginning point at the center of the driven wheel tire to ground contact patch **31**, and passes through the instant force center **24**, before it terminates on a squat layout line **11**. The intersection of the squat force vector **7** and the squat layout line is called the squat layout point **10**. The squat layout line **11** is graphed at a perpendicular angle to gravitational force. A lower squat measurement definition line **12** is graphed beginning at the center of the driven wheel tire to ground contact patch **31** and terminating perpendicular and coincident to the squat layout line **11**. The perpendicular measurement from the lower squat measurement definition line **12** to the squat layout point **10** is called the measured squat distance **13**. This measured squat distance **13** changes as driven wheel suspension travel distance **15** compresses, and is used to create a squat curve **17** in a squat curve graph as shown in FIGS. **3** and **4**.

FIG. **2***c* shows an enlarged section of the side view of the shaft driven vehicle shown in FIGS. **2***a* and **2***b* with the driven wheel suspension system in a completely uncompressed state.

FIG. **2***d* shows an enlarged section of the side view of the shaft driven vehicle shown in FIGS. **2***a*, **2***b*, and **2***c* with the driven wheel suspension system in a completely compressed state. FIGS. **2***c* and **2***d* further exemplify certain embodiments, for example, points and vectors useful for a graphical method used to attain a squat point measurement, and a graphical method to attain suspension geometry kinematical layout from an existing desired measured squat point.

FIG. **3** shows a squat curve for suspension systems according to certain embodiments of the invention graphed on a squat curve graph as disclosed herein. The percent of total suspension travel is shown on the x-axis, and the percent of total squat is shown on the y-axis. FIG. **3** exemplifies a squat curve (**17**). The slope and shape of the squat curve shown in FIG. **3** exemplifies a squat curve produced by suspension systems of the invention, for example, suspension systems including features as illustrated in FIGS. **1***a*-**1***d* and FIGS. **2***a*-**2***d*. FIG. **3** also exemplifies a graphical method useful to obtain a squat curve graph.

FIG. **4** shows a squat curve for suspension systems according to certain embodiments of the invention. The percent of total suspension travel is shown on the x-axis, and the percent of total squat is shown on the y-axis. FIG. **4** exemplifies a squat curve **17** with tangent lines depicting a slope of the curve at certain points along the squat curve. The slopes exemplified by the tangent lines are the first squat curve slope **18**, the second squat curve slope **19**, and the third squat curve slope **30**. FIG. **4** exemplifies a slope of the squat curve **17** as produced by a suspension system of certain embodiments of the current invention, for example, a suspension system including features as illustrated in FIGS. **1***a*-1*d* and FIGS. **2***a*-**2***d*, and that the slope varies as the vehicle suspension travel distance increases. The squat curve **17** produced has a first squat curve slope **18** that has a negative value at the beginning point in the suspension travel, and a second squat curve slope **19** at an interim point that is higher, or less negative, than the first squat curve slope **18** in the suspension travel, and a third squat curve slope **30** at the ending point in the suspension travel that has a lower, or more negative, value than the second squat curve slope **19**.

FIGS. **5**-**13** show alternative embodiments of suspension systems comprising a squat curve of the invention. Each embodiment shown includes a spring/damper unit (small irregular box) and different frame members (thicker lines) interconnected through pivots (small circles).

Mass transfer is discussed. All vehicles have mass. The mass of a suspended static vehicle system can be modeled as shown in the FIG. **1**. Mass in all vehicles with a suspension system can be divided into sprung and unsprung mass. Unsprung mass is comprised of the sum of all vehicle parts that move with a suspended wheel. Sprung mass is comprised of the sum of vehicle parts that can remain stationary as a suspended wheel is moved. The dynamic center of the sprung mass as shown in FIG. **2** is a combination of rider and/or passenger mass and the vehicle mass.

The combination of a rider's mass and the sprung mass of the bicycle are always supported fully by the combination of the vehicle's tires. Powered forward acceleration transfers mass from the vehicle's front wheel(s) to the vehicle's driven wheel(s), braking transfers mass from the vehicle's front wheel(s) to the vehicle's driven wheel(s). Riding on the driven wheel(s) only transfers all of the mass to the driven wheel(s), and riding on the front wheel(s) only transfers all of the mass to the front wheel(s).

Due to their combination of short wheelbase (WB) and high center of gravity (CG), motorcycles and bicycles experience the affects of load transfer to a much greater extent than other vehicles in existence. The ratio of the distance from the ground to the CG and the distance between the points where the wheels touch the ground (WB) illustrates this point. For example, a common bicycle will exhibit a center of gravity to wheelbase ratio of nearly 100%, motorcycles are typically near 50%, and passenger cars are typically near 25%. Mass transfer is sometimes also referred to as load transfer.

Energy loss through mass transfer is discussed. One undesirable effect of driven wheel suspension systems is the loss of energy in the way of extreme suspension compression or extension during powered acceleration. This suspension compression or extension is categorized as squat.

A suspension system's geometry and positional relationships between the vehicle drive system components can greatly affect the internal distribution of forces within the vehicle chassis. As a suspension system cycles through its suspension travel, the positional relationships between the suspension system and the vehicle drive system can change, and at the same time, the suspension geometry itself will change. These fluctuations of internal forces are what govern suspension response to powered acceleration and braking. Vehicle attitude in relation to gravity, and sprung weight center of mass change will also govern suspension response to powered acceleration and braking. These external forces are considered stationary and equal when comparing like vehicles in order to determine squat characteristics.

Squat is the result of internal chassis forces that can cause a rear suspension to extend or compress during powered acceleration. Squat is an instantaneous condition that can vary throughout the suspension travel. Instantaneous squat response is governed by sprung mass CG placement, suspension geometry, powertrain component location, and grade in relation to gravity that the vehicle is traveling on. Sprung mass CG placement only defines the amount of squat present in a suspension, and does not change the squat conditions. The squat conditions define the direction of squat force in relation to gravity.

There are three squat conditions that must be considered. The first condition is pro-squat, and describes the condition present when a rear suspension is forced to compress by internal suspension forces under powered acceleration. The second condition is anti-squat. Anti-squat describes the condition present when a rear suspension compression is counteracted by internal suspension forces under powered acceleration. The third condition is zero-squat. Zero-squat occurs only at the instant in between pro-squat and anti-squat, where no suspension manipulating forces are present under powered acceleration. A vehicle suspension operating at the point of zero-squat will not use acceleration forces to manipulate suspension reaction in any way.

Squat force works independent of the spring force that supports a suspended vehicle. Because the squat force is independent of the vehicle spring force, when under acceleration, a vehicle suspension is acted upon by its spring and the squat force together. Suspended vehicles use springs to support the vehicle chassis and dampers to dissipate impact energy when the suspension system is compressed and extended while the vehicle travels over rough terrain. Springs can be in the form of compressive gas springs, leaf springs, or coil springs, and dampers can use fluid or friction to dissipate energy. When a vehicle is at rest, suspended wheels are compressed a certain amount so that the suspended wheel can follow irregular road surfaces with both bumps and dips. The spring that supports a wheel suspension acts as an energy storage device. Vehicle suspensions use the damper units to dissipate energy stored in a spring after the spring is compressed. The further a spring is compressed, the more energy is stored, and the more energy will be dissipated by the damper when the spring rebounds. Because spring force increases as a wheel is compressed into its suspension travel, force at the suspended wheel also increases.

Squat curve graphing is discussed. A squat curve graph is a representation of the squat produced by a compressible suspension system under powered acceleration. The squat curve graph is laid out so that the percentage of suspension travel is graphed on the X axis, and escalating in a positive direction. The minimum suspension travel, which is zero percent suspension compression, is shown at the far left of the x-axis, and the maximum suspension travel, which is represented by 100 percent suspension compression, is shown at the far right of the x-axis. Percent suspension compression is measured and graphed in minimum increments of 5 percent total suspension compression; measured Percent total squat is graphed on the y-axis in an escalating amount. The highest amount of squat is defined as 100 percent, and is represented at the top of the y-axis. These values are taken directly from the squat points which are measured from graphed squat points on the squat layout line. Measurement is taken at a perpendicular distance from the lower squat measurement definition line. Zero percent squat is always measured at the point of zero squat condition. This zero squat condition is measured when the squat point lies directly on the lower squat measurement definition line. At this point, the squat measurement has no value. Any measurement of a squat point that lies below the lower squat definition line is equal to a pro squat amount, and must be graphed as a negative percentage of the 100 percent squat value. The amount of squat closer to or highest in the range of the squat condition known as anti squat is listed as the highest positive squat value, and lower amounts of anti squat, zero squat, and pro-squat are listed as lower percentages of the highest anti squat value. Zero squat is shown when the squat curve crosses or terminates at zero value on the y-axis, and pro squat is graphed as a negative y-axis percentage below the x-axis. For example, if a squat curve begins with a measurement that is measured 100 millimeters above the lower squat measurement definition line, at a point of zero suspension compression, this point will be graphed at a value of 1 on the y-axis, and 0 on the x-axis. If a later point is measured 100 millimeters below the lower squat measurement definition line, at a point of 100 percent suspension compression, this point will be graphed at a value of −1 on the y-axis, and 1 on the x-axis. In the squat curve graph, the distance set to equal 100 percent suspension travel and the distance set to equal 100 percent squat should be set as equal distances. Therefore, the distance between zero value for squat to maximum value for squat will be equal to the graphed distance between zero value for suspension compression to maximum value for suspension compression. When desired squat point values are known and graphed versus their corresponding percent measured suspension compression values, the points can be connected from point to point using typical graphing method A curve can then be fit to the point to point graph so that the curve represents a smoothed best fit version of the point to point graph. The most efficient method to obtain such a curve is to use a computer program such as Microsoft Excel, available from Microsoft Corporation, One Microsoft Way, Redmond, Wash. 98052-6399, USA. Using Microsoft Excel, a user can input the escalating suspension travel measurements beginning with the zero percent measurement and ending with the 100 percent measurement, and can input the measured or preferred squat point measurements that coincide with their percent suspension travel measurements. Microsoft Excel then can be used to create a graph of the points with a curve fit to the graphed points. This graphed curve is the discussed squat curve.

Slope of a squat curve between two points on a curve is defined by the standard coordinate geometry equation: slope=rise/run. A squat curve that has a squat amount at zero suspension travel, with 20 percent less squat at a point 10 percent into the wheel suspension travel compression will have a slope of −2, because per the equation slope=rise/run, −0.2/0.1=−2. A squat curve that has a pro squat amount at zero suspension travel, with 20 percent more pro squat at a point 10 percent into the wheel suspension travel compression will have a slope of −2, because per the equation slope=rise/run, −0.2/0.1=−2. A squat curve can be produced for any wheel suspension system by graphing the percent of squat throughout the suspension travel.

In certain embodiments, a suspension system according to the invention has a squat curve with a negative, or decreasing, slope. In certain preferred embodiments, the slope of the squat curve is more negative at the beginning of suspension travel than in the interim, or mid range, of suspension travel. In certain other preferred embodiments, the slope of the squat curve is more negative at the end of suspension travel than in the interim, or mid range, of suspension travel. In certain other preferred embodiments, the slope of the squat curve is more negative at the beginning of suspension travel than at the end of suspension travel.

In certain embodiments, the beginning of the suspension travel is 0 to 50 percent, or about 0 to about 50 percent, of suspension travel; or 0 to 40 percent, or about 0 to about 40 percent, of suspension travel; or 0 to 30 percent, or about 0 to about 30 percent, of suspension travel; or 0 to 20 percent, or about 0 to about 20 percent, of suspension travel; or 0 to 10 percent, or about 0 to about 10 percent, of suspension travel; or 0 to 5 percent, or about 0 to about 5 percent, of suspension travel; or 0 or about 0 percent of suspension travel. In certain embodiments, the interim, or mid range, of the suspension travel is 25 to 75 percent, or about 25 to about 75 percent, of suspension travel; or 30 to 70 percent, or about 30 to about 70 percent, of suspension travel; or 35 to 65 percent, or about 35 to about 65 percent, of suspension travel; or 40 to 60 percent, or about 40 to about 60 percent, of suspension travel; or 45 to 55 percent, or about 45 to about 55 percent, of suspension travel; or 50 percent or about 50 percent, of suspension travel; or 60 to 80 percent, or about 60 to about 80 percent, of suspension travel; or 65 to 75 percent, or about 65 to about 75 percent, of suspension travel; or 70 percent or about 70 of suspension travel; or 50 to 60 percent, or about 50 to about 60 percent, of suspension travel. In certain embodiments, the end of the suspension travel is 70 to 100 percent, or about 70 to about 100 percent, of suspension travel; or 75 to 100 percent, or about 75 to about 100 percent, of suspension travel; or 80 to 100 percent, or about 80 to about 100 percent, of suspension travel; or 85 to 100 percent, or about 85 to about 100 percent, of suspension travel; or 90 to 100 percent, or about 90 to about 100 percent, of suspension travel; or 95 to 100 percent, or about 95 to about 100 percent, of suspension travel; or 100 or about 100 percent of suspension travel.

In certain embodiments, a suspension system of the invention has a squat curve with a slope in the beginning of suspension travel of −0.2 to −5, or about −0.2 to about −5; of −0.5 to −4.5, or about −0.5 to about −4.5; of −0.75 to −4.0, or about −0.75 to about −4.0; of −1.0 to −3.5, or about −1.0 to about −3.5; of −1.5 to −3.0, or about −1.5 to about −3.0; of −2.0 to −2.5, or about −2.0 to about −2.5. In certain embodiments, a suspension system of the invention has a squat curve with a slope in the interim, or mid range, of suspension travel of −0.0001 to −5, or about −0.0001 to about −5; of −0.01 to −4.0, or about −0.01 to about −4.0; of −0.1 to −3.0, or about −0.1 to about −3.0; of −0.2 to −2.0, or about −0.2 to about −2.0; of −0.3 to −1.2, or about −0.3 to about −1.2; of −0.4 to −0.8, or about −0.4 to about −0.8. In certain embodiments, a suspension system of the invention has a squat curve with a slope in the end of suspension travel of −0.0002 to −1000, or about −0.0002 to about −1000; of −0.1 to −500, or about −0.1 to about −500; of −0.2 to −50, or about −0.2 to about −50; of −0.3 to −10, or about −0.3 to about −10; of −0.4 to −5.0, or about −0.4 to about −5.0; of −0.6 to −2.0, or about −0.6 to about −2.0.

Graphical kinematical squat curves are discussed. Graphical methods can be used to determine suspension kinematical layout used to attain a desired squat curve for a suspension. For shaft drive and chain drive vehicles, graphical layout is identical until factoring in the unique features of each powertrain. Any suspended wheel in a vehicle has an axle path that a wheel follows when a suspension is moved through suspension travel. The curvature of this axle path and its layout in relation to specific powertrain components define a squat curve. A squat curve is a measurement of the changing magnitude and direction of squat developed under powered acceleration as suspension system is cycled through suspension travel from its beginning uncompressed point to its ending fully compressed point. Every instantaneous point in a suspension travel has a corresponding instantaneous amount of squat present. These instantaneous squat points can be measured or graphed as a point on the squat layout line at a perpendicular distance from the lower squat layout line. When the desired instantaneous amounts of squat at instantaneous points in the suspension travel are known, squat definition points can be graphed in conjunction with each other, beginning when a suspension is in its uncompressed state and ending in its fully compressed state, and in relation to the vehicle geometry to obtain a suspension kinematical layout which will attain the desired squat curve. The squat curve beginning value is measured at the point where the suspension system is in its completely uncompressed state. As the suspension is cycled further through suspension travel towards complete compression pausing at a minimum of 5 percent total suspension travel increments, further squat points are measured and graphed versus their correlating escalating percent total suspension travel increments. Suspension travel displacement is measured in a direction parallel to and against gravity, and parallel to the instantaneous squat point measurements. Critical and known preexisting defining points such as vehicle wheelbase, powertrain location, and center of mass are graphed alongside the squat definition points to obtain a clear picture of vehicle squat performance. Vehicle graphs for obtaining and defining squat performance are always laid out with the vehicle viewed in the side elevational view.

A squat layout line is drawn parallel to and against gravitational force through center of the front wheel contact patch between the tire and the ground and terminating at further points. A squat definition point, which is taken directly from the aforementioned squat curve will be graphed on this squat layout line. A squat lower measurement definition line is drawn from the center of the driven wheel tire to ground contact patch perpendicular to and terminating on the squat layout line. Squat definition points are drawn on the squat definition line in relation to one another, and in relation to the squat lower measurement definition line. A squat definition point drawn above the squat lower measurement definition line will correlate with a squat amount. A squat definition point drawn coincident with the squat lower measurement definition line will correlate with a zero squat amount. A squat definition point drawn below the squat lower measurement definition line will correlate with a pro squat amount. A squat force vector is drawn from the center of the driven wheel tire to ground contact patch to the squat point on the squat layout line. As the suspension is moved through instantaneous measured points through suspension travel, the squat force vector is drawn with a beginning point at the center of the rear tire to ground contact patch, and an ending point at its corresponding measured instantaneous squat point graphed on the squat layout line.

Diversion in graphical method to obtain specific suspension system kinematical layouts from a desired squat curve must occur when factoring in specifics for different types of power transfer systems such as shaft drive or chain drive.

A shaft drive system generally uses a power transmission system that can transmit power via rotary motion from a power unit output shaft to a wheel shaft. The two shafts are generally fixed at close to a perpendicular angle in one plane. Power transmission systems can vary from gears to cogs to friction wheels and other types of systems, all herein referred to universally as cogs. These shaft drive systems feature a driving cog which is rotatably attached to the power unit output, a first intermediate cog, which transfers rotational motion from the driving cog to a relatively perpendicular shaft, a second intermediate cog, which transfers rotational motion from the shaft to a driven cog which is rotatably attached to the rotation axis of a wheel.

Shaft drive vehicle powertrains and suspensions typically take one of two forms. These are, a single pivot system, or a multi link system. A simple single pivot system features a driven cog that is fixed to and housed within a swinging wheel carrier link which pivots around a single pivot. In this arrangement, there is only one pivot connecting the swinging wheel carrier link to the vehicle frame structure. The rotating drive torque is acted against by the driven cog housing, which is part of the swinging wheel carrier link. Action against the drive torque in the swinging wheel carrier link causes a torque about the ling single frame pivot. The addition of this torque plus the driving force imparted through the wheel tire combination to the ground through a tire to ground contact patch totals a squat response. An instantaneous pivot location for a single pivot shaft drive system can be found at any point on a drawn squat force vector that correlates with the desired instantaneous squat response. These single pivot systems produce a linear squat curve.

A multi pivot linkage can be used to alter squat characteristics and obtain a variable squat curve in a shaft driven wheel suspension system. A multi link shaft drive suspension system isolates the torque passed through the driven cog in the system from the swinging link system. In a 4-bar variation, the driven cog is attached to a swinging wheel carrier link, which pivots at one end of a first swinging link. The first carrier manipulation link is pivotally attached to the vehicle chassis at the end opposite of the swinging wheel carrier link pivot. A torque reaction, like the one discussed in the single pivot shaft drive system works to rotate the swinging wheel carrier link against the first carrier manipulation link. A second carrier manipulation link is also attached to the swinging wheel carrier link. This second carrier manipulation link is attached to the vehicle chassis at a different location from the first swinging carrier manipulation link. The second carrier manipulation link works to inhibit free rotation of the swinging wheel carrier link against the first carrier manipulation link. To find instantaneous carrier manipulation link pivot points which will give a desired instantaneous squat amount, its correlating desired squat force vector must be graphed. The two swinging wheel carrier link pivots are next defined. Carrier manipulation link force lines are drawn so that a force line passes directly through the center of the rearward pivots which are coincident with the pivots on the swinging wheel carrier link. The carrier manipulation link force lines are drawn so that they intersect on the desired squat force vector. The first and second vehicle chassis pivots can be positioned upon the corresponding first and second carrier manipulation link force lines to attain the desired instantaneous squat response. Graphing the carrier manipulation link force lines and desired squat force vectors together overlaid at multiple points in the suspension travel will allow the designer to choose pivot point locations and kinematical suspension layout that can attain a desired variable squat curve.

A chain drive powertrain system uses a chain or belt to transmit power between two reasonably parallel shafts. Chain drive systems are very common in motorcycle, ATV, and bicycle applications because of their light weight, robustness, and simplicity in both manufacturing and use. The chain drive systems feature a driving cog and a driven cog, with the driving cog attached to a power source, and a driven cog rotatably attached to the rotation axis of a wheel. The driven wheel or wheels is/are attached to a swinging link or linkage system via a bearing or bushing system, which allows rotational motion of the driven wheel or wheels in relation to the swinging link or linkage system.

Chain drive suspensions typically take one of several forms. These include single pivot systems, multi link systems, cam/track type systems, and flexure type systems. The suspensions can also feature variable chainline type designs, which manipulate a chain force vector line through the use of a pulley system that moves with the suspension. A single pivot system uses a single pivoting suspension link to transmit force between a suspended wheel and a chassis. A multi link system uses an arrangement of pivoting suspension links to transmit force between a suspended wheel and a chassis. A cam/track type system that uses sliding elements but does not use links to attain force transfer from a wheel axle to a chassis is also possible but uncommon in practice. Flexure type systems use flexing elements to transmit power from a suspended wheel to a chassis structure. In all types of the chain driven wheel suspension system mentioned above, the driving force can be represented as a vector drawn perpendicular to the driven wheel axle path. In a chain driven suspension, driving force is always the major force component when compared to chain pull.

There are two internal forces present within a chain driven vehicle chassis that together create a squat response. These two forces are driving force, and chain pull force.

When a chain driven vehicle is accelerated, force is transferred from a power source to a driving cog. This driving cog transmits its force through a chain to a driven cog. The force direction and magnitude present in the tensioned chain are referred to as chain pull force. Fixed chainline type designs are present where at any instantaneous point, a single driving cog is fixed rotationally on a chassis structure, and a driven cog is attached to a suspension member, and force is transmitted from the driving cog to the driven cog through a chain. In this fixed chainline type design, the chainline force vector is always located at one end by the tensioned chainline tangent point where the chain is fixed in relation to the vehicle chassis structure, and by the tensioned chainline tangent point of the moving pulley at the opposite end.

In variable chainline type designs, which manipulate a chain force vector line through the use of a pulley system that moves with the suspension, the chainline force vector is always located at one end by the tensioned chainline tangent point where the chain is fixed in relation to the vehicle chassis structure, and by the tensioned chainline tangent point of the moving pulley at the opposite end. Sliding elements can also be substituted for pulleys in this application.

In the chain drive powertrain, the driven cog is rotatably attached to a wheel/tire combination. The wheel pushes against the ground resulting in friction. As the wheel rotates a driving force transmitted from the contact patch through the wheel structure and a force is imparted at the rear hub axle. This pushing force can be transferred to the chassis via a wheel suspension system, ultimately pushes the vehicle forward. This pushing force is referred to as driving force. The driving force direction is measured and represented graphically as a driving force vector drawn from the driven wheel rotation axis, perpendicular to the driven axle path, where the axle path is defined as a line that a suspended wheel rotational axis travels as a suspension is moved through suspension travel. This axle path can be a constant curvature or changing curvature line depending on suspension layout.

A simple single pivot system features a driven cog that is rotatably attached to a wheel, which is rotatably attached to a swinging wheel carrier link which pivots around a singular pivot. In this arrangement, the suspended wheel travels in a constant radius arc. To find the instantaneous swinging link pivot point for a single pivot chain drive system, which will give a desired instantaneous squat amount, its correlating desired squat force vector must be graphed. Because there is only one link in the single pivot suspension, the swinging link pivot will lie coincident with the driving force line. Desired vehicle geometry is graphed in a side view. This vehicle geometry will include the size, location, and center points of vehicle tires, powertrain component layout, and the direction of gravitational force. A squat layout line is graphed first. A desired squat force vector is drawn from the center of a rear wheel contact patch to the desired squat layout point on a squat layout line as described previously. Next, the chain force vector is graphed in relation to the powertrain components as described previously. The chain force vector is drawn so that it intersects the squat force vector. Finally, the driving force vector is drawn from the center of the rear wheel axis to the intersection point of the squat force vector and chain pull force vector. The pivot point for the single pivot swinging link suspension arm will lie at any point along the driving force vector to achieve the desired instantaneous squat amount. Graphing the chain pull force vector, and squat force vectors together overlaid at multiple points in the suspension travel will allow the designer to find driving force vectors at multiple points through the suspension travel. The crossing point of the overlaid driving force vectors for different points in the suspension travel define the single pivot point location and kinematical suspension layout that can attain the desired squat curve.

Multi link systems, cam/track (sliding link) type systems, and flexure type systems feature a driven cog that is rotatably attached to a wheel, which is rotatably attached to a swinging wheel carrier link which moves the wheel along an axle path that is defined by a multi element system. To aid the analysis of multi-element systems, it is simplest to define or measure an axle path which will guide a wheel, and then define the elements that will give the desired axle path later, as opposed to trying to define elements first and measure axle path as a byproduct later to attain a desired result. Multi element systems do not have a single hardware defined pivot point like a single fixed pivot system does. The multi element systems use combinations of links or cams to project a virtual or instantaneous pivot point. This pivot point can always be found at a point along a driving force vector, which is drawn perpendicular to a driven wheel axle path as previously described.

To find the axle path which will give a desired instantaneous squat amount, its correlating desired squat force vectors must be graphed. Desired vehicle geometry is graphed in a side view. This vehicle geometry will include the size, location, and center points of vehicle tires, vehicle ground plane, powertrain component layout, and the direction of gravitational force. A vehicle wheel suspension system always has a minimum suspension travel point, where the suspended wheel is at its zero compressed suspension travel point, and a maximum suspension travel point, where the suspended wheel is at its 100 percent compressed suspension travel point. Several overlaid graphs must be made to obtain a squat curve. The minimum increment in suspension compression displacement that can be used to graph an accurate squat curve from the graphical method using squat force vectors as presented has been found to be 5 percent of total suspension compression displacement between graphed squat force vectors. A squat layout line is graphed first. A desired squat force vector is drawn from the center of a driven wheel contact patch to the desired squat layout point on a squat layout line as described previously. Next, the chain force vector is graphed in relation to the powertrain components as described previously. The chain force vector is drawn so that it intersects the squat force vector. Finally, the driving force vector is drawn from the center of the driven wheel axis to the intersection point of the squat force vector and chain pull force vector. The instantaneous pivot point for the single pivot swinging link suspension arm will lie at any point along the driving force vector to achieve the desired instantaneous squat amount. Graphing the chain pull force vector, and squat force vectors together overlaid at multiple points in the suspension travel will allow the designer to find driving force vectors at multiple points through the suspension travel. The crossing point of the overlaid driving force vectors for different points in the suspension travel define the instantaneous pivot point movement through the suspension travel, and kinematical suspension layout that can attain the desired squat curve. For multi element systems, there are several methods that can define element layout based on a desired axle path, for example, by using kinematical analysis computer software. Software that can perform this specific function is marketed under the names SyMech, which is available from SyMech Inc, 600 Townsend Street, San Francisco, Calif., 94107, USA, and SAM, which is available from ARTAS-Engineering Software, Het Puyven 162, NL-5672 R J Nuenen, The Netherlands. This software allows a user to define an axle path, and set parameters such as mechanical element type, number of mechanical elements, and desired location of anchor components. The software will then suggest multiple link layout choices that will meet all of the set forth parameters. Graphical analysis can also be performed by hand. In a hand graphical analysis, the mechanical components of a multi element system are measured at multiple points through the suspension travel. At each point in the suspension travel, the instant center of the link system is graphed. A common 4-bar linkage suspension system features a driven cog that is rotatably attached to a driven wheel, which is rotatably attached to a swinging wheel carrier link which is pivotably attached to two separate carrier manipulation links. The swinging links are pivotably attached to a vehicle chassis at their other ends. The instant center in a 4 bar pivoting linkage system such as shown in FIG. **1***a*, is found by projecting individual link force lines through both pivots of each of the two carrier manipulation links that support the swinging wheel carrier. The two carrier manipulation link force lines are projected so that they intersect each other. This intersection point is commonly known at the instant force center. A driving force line can be drawn directly from the rotation axis of the driven wheel to this instant force center. As the carrier manipulation links rotate on their pivots, the instant center position changes in relation to the driven wheel rotation axis and the vehicle chassis. This causes the driving force line to move in relation to the chain force line. Because the squat force line is defined in part by the location of the driven wheel contact patch, and the intersection between the driving force vector and the chain force vector, a change in squat vector direction can occur. The perpendicular distance from the lower squat definition line to the point at which this squat direction vector intersects the drawn squat layout line to is measured and recorded.

Four bar sliding link suspension systems are analyzed identically to 4 bar pivoting systems, but the identification of the instant center is performed in a slightly different manner due to the constraints of the sliding link system. Four bar sliding link systems feature a driven cog that is rotatably attached to a driven wheel, which is rotatably attached to a swinging wheel carrier link which is pivotably attached to two separate sliding carrier manipulation sliding blocks. The individual carrier manipulation sliding blocks move on individual sliding rails. The instant center in a 4 bar sliding linkage system is found by projecting individual sliding link force lines centered at the pivots of each of the two carrier manipulation sliding block that support the swinging wheel carrier. The carrier manipulation sliding block force lines are projected perpendicular to the sliding rail so that the two carrier manipulation sliding black force lines intersect each other. This intersection can be referred to as the instant force center. A driving force line can be drawn directly from the rotation axis of the driven wheel to this instant force center. As the carrier manipulation sliding blocks slide on their respective sliding rails, the instant center position changes in relation to the driven wheel rotation axis and the vehicle chassis. This causes the driving force line to move in relation to the chain force line. Because the squat force line is defined in part by the location of the driven wheel contact patch, and the intersection between the driving force vector and the chain force vector, a change in squat vector direction can occur. The perpendicular distance from the the lower squat definition line to the point at which this squat direction vector intersects the drawn squat layout line to is measured and recorded.

Measurement of multi element systems to determine axle path can be done graphically, or by using measurement equipment. Using measurement equipment, the vehicle can be rigidly mounted and oriented so that the suspended wheel can be moved freely through measured points in its suspension travel while the chassis stays stationary. In a side view orientation, the horizontal and vertical distance from the suspended wheel rotation axis to a fixed point on the vehicle frame at multiple points in the suspension travel is taken. As the suspension is cycled through suspension travel, the corresponding measurements of horizontal and vertical distance form a wheel rotation axis travel path in relation to the vehicle chassis. This path is referred to as the axle path.

Analysis has shown that a vehicle with a compressible suspension system using a chain driven suspended wheel achieves the squat curve **17** of the current invention by having a layout that features a driven cog that is rotatably attached to a driven wheel, which is rotatably attached to a swinging wheel carrier link which is pivotably attached to separate upper and lower carrier manipulation links. The upper and lower carrier manipulation links are pivotably attached to a vehicle chassis at their other ends. The upper and lower carrier manipulation links rotate in the same rotational direction about their respective fixed axis at the vehicle chassis. The upper carrier manipulation link is arranged in relation to the lower carrier manipulation link so that the instant center projected by the two carrier manipulation links, when measured at zero percent suspension compression, is at a point that is beyond the outer limits of the two pivots of the lower carrier manipulation link. This condition is shown in FIGS. **1***a* and **1***c*. As the suspension is compressed towards a point of full compression, the rotation of the upper and lower carrier manipulation links in relation to each other causes the instant center of the linkage system to lie at points on the lower carrier manipulation link in between the lower carrier manipulation link fixed vehicle chassis pivot, and moving pivot attached to the swinging wheel carrier link. This condition is shown in FIGS. **1***b* and **1***d*.

Analysis has shown that a vehicle with a compressible suspension system using a shaft driven suspended wheel achieves the squat curve **17** of the current invention by having a layout that features a driven cog that is rotatably attached to a driven wheel, which is rotatably attached to a swinging wheel carrier link which is pivotably attached to separate upper and lower carrier manipulation links. The upper and lower carrier manipulation links are pivotably attached to a vehicle chassis at their other ends. The upper and lower carrier manipulation links rotate in a contra rotational direction about their fixed axes at the vehicle chassis. The upper carrier manipulation link is arranged in relation to the lower carrier manipulation link so that the instant center projected by the two carrier manipulation links, when measured at zero percent suspension compression, lies at a point on the lower carrier manipulation link in between the lower carrier manipulation link fixed vehicle chassis pivot, and moving pivot attached to the swinging wheel carrier link. This condition is shown in FIGS. **2***a* and **2***c*. As the suspension is compressed towards a point of full compression, the rotation of the upper and lower carrier manipulation links in relation to each other causes the instant center of the linkage system to lie at a point that is beyond the outer limits of the two pivots of the lower carrier manipulation link. This condition is shown in FIGS. **2***a* and **2***d*.

The present invention is not to be limited in scope by the specific embodiments described herein, which are intended as single illustrations of individual aspects of the invention, and functionally equivalent methods and components are within the scope of the invention. Indeed, various modifications of the invention, in addition to those shown and described herein, will become apparent to those skilled in the art from the foregoing description. Such modifications are intended to fall within the scope of the appended claims. All cited publications, patents, and patent applications are herein incorporated by reference in their entirety.

What is claimed is:

1. A driven wheel suspension system comprising a chassis having a driving coq rotation axis, an upper carrier manipulation link pivotally connected to the chassis at a fixed upper pivot, a swinging wheel carrier link pivotally connected to said upper carrier manipulation link at an upper link floating pivot, a lower carrier manipulation link pivotally connected to the swinging wheel carrier link at a lower link floating pivot, said lower carrier manipulation link being pivotally connected to the chassis at a lower link fixed pivot, wherein said lower link fixed pivot is above the driving cog rotation axis, and wherein said suspension system has a squat curve with a negative slope in the beginning of suspension travel and in the end of suspension travel, and wherein the slopes in the beginning of suspension travel and in the end of suspension travel are more negative than the slope of the squat curve in the interim of suspension travel, and wherein said squat curve is obtained by determining a measured squat distance at various percentages of suspension compression, said measured squat distance being determined by an intersection point of a squat force vector and a squat layout line, said squat force vector also intersecting with an intersection point of a chain force vector and a driving force vector and said squat force vector also intersecting with a drive wheel ground contact.

2. The suspension system of claim **1**, wherein the suspension system is useful for a chain driven vehicle.

3. The suspension system of claim **1**, wherein the suspension system is useful for a shaft driven vehicle.

4. The suspension system of claim **1**, wherein the suspension system is useful for a belt driven vehicle.

5. The suspension system of claim **1**, wherein the suspension system is useful for a human powered vehicle.

6. The suspension system of claim **1**, wherein said suspension system comprises a damper unit and wherein said damper unit in said suspension system is selected from the group consisting of a spring, a compression gas spring, a leaf spring, a coil spring, and a fluid, and wherein said damper unit is connected to a structure selected from the group consisting of an upper carrier manipulation link, a lower carrier manipulation link, a wheel carrier link, and a chassis.

7. The suspension system of claim **1**, wherein the suspension system comprises an upper carrier manipulation link and a lower carrier manipulation link and wherein both links are pivotally attached to the chassis of the vehicle in which said suspension system is used.

8. The suspension system of claim **7**, wherein said suspension system comprises a damper unit and wherein said damper unit in said suspension system is selected from the group consisting of a spring, a compression gas spring, a leaf spring, a coil spring, and a fluid, and wherein said damper unit is connected to said upper and lower carrier manipulation links.

* * * * *

UNITED STATES DISTRICT COURT, CENTRAL DISTRICT OF CALIFORNIA
CIVIL COVER SHEET

I (a) PLAINTIFFS (Check box if you are representing yourself)	DEFENDANTS
dw-link Incorporated	Giant Bicycle, Inc., and Giant Manufacturing Co., Ltd.
(b) Attorneys (Firm Name, Address and Telephone Number. If you are representing yourself, provide same.) Arthur Beeman, SNR Denton US LLP 525 Market Street, 26th Floor, San Francisco, CA 94105 Tel: (415) 882-5000	Attorneys (If Known) By Fax

II. BASIS OF JURISDICTION (Place an X in one box only.)

1 U.S. Government Plaintiff
X 3 Federal Question (U.S. Government Not a Party)
2 U.S. Government Defendant
4 Diversity (Indicate Citizenship of Parties in Item III)

III. CITIZENSHIP OF PRINCIPAL PARTIES - For Diversity Cases Only (Place an X in one box for plaintiff and one for defendant.)

	PTF	DEF		PTF	DEF
Citizen of This State	1	1	Incorporated or Principal Place of Business in this State	4	4
Citizen of Another State	2	2	Incorporated and Principal Place of Business in Another State	5	5
Citizen or Subject of a Foreign Country	3	3	Foreign Nation	6	6

IV. ORIGIN (Place an X in one box only.)

X 1 Original Proceeding
2 Removed from State Court
3 Remanded from Appellate Court
4 Reinstated or Reopened
5 Transferred from another district (specify):
6 Multi-District Litigation
7 Appeal to District Judge from Magistrate Judge

V. REQUESTED IN COMPLAINT: JURY DEMAND: X Yes No (Check 'Yes' only if demanded in complaint.)

CLASS ACTION under F.R.C.P. 23: Yes X No

☒ **MONEY DEMANDED IN COMPLAINT:** $ TBD

VI. CAUSE OF ACTION (Cite the U.S. Civil Statute under which you are filing and write a brief statement of cause. Do not cite jurisdictional statutes unless diversity.)
35 U.S.C. 271, 283, 284 and 285, Patent Infringement, Breach of Contract and Unjust Enrichment

VII. NATURE OF SUIT (Place an X in one box only.)

OTHER STATUTES
- 400 State Reapportionment
- 410 Antitrust
- 430 Banks and Banking
- 450 Commerce/ICC Rates/etc.
- 460 Deportation
- 470 Racketeer Influenced and Corrupt Organizations
- 480 Consumer Credit
- 490 Cable/Sat TV
- 810 Selective Service
- 850 Securities/Commodities/ Exchange
- 875 Customer Challenge 12 USC 3410
- 890 Other Statutory Actions
- 891 Agricultural Act
- 892 Economic Stabilization Act
- 893 Environmental Matters
- 894 Energy Allocation Act
- 895 Freedom of Info. Act
- 900 Appeal of Fee Determination Under Equal Access to Justice
- 950 Constitutionality of State Statutes

CONTRACT
- 110 Insurance
- 120 Marine
- 130 Miller Act
- 140 Negotiable Instrument
- 150 Recovery of Overpayment & Enforcement of Judgment
- 151 Medicare Act
- 152 Recovery of Defaulted Student Loan (Excl. Veterans)
- 153 Recovery of Overpayment of Veteran's Benefits
- 160 Stockholders' Suits
- 190 Other Contract
- 195 Contract Product Liability
- 196 Franchise

REAL PROPERTY
- 210 Land Condemnation
- 220 Foreclosure
- 230 Rent Lease & Ejectment
- 240 Torts to Land
- 245 Tort Product Liability
- 290 All Other Real Property

TORTS
PERSONAL INJURY
- 310 Airplane
- 315 Airplane Product Liability
- 320 Assault, Libel & Slander
- 330 Fed. Employers' Liability
- 340 Marine
- 345 Marine Product Liability
- 350 Motor Vehicle
- 355 Motor Vehicle Product Liability
- 360 Other Personal Injury
- 362 Personal Injury-Med Malpractice
- 365 Personal Injury-Product Liability
- 368 Asbestos Personal Injury Product Liability

IMMIGRATION
- 462 Naturalization Application
- 463 Habeas Corpus-Alien Detainee
- 465 Other Immigration Actions

TORTS
PERSONAL PROPERTY
- 370 Other Fraud
- 371 Truth in Lending
- 380 Other Personal Property Damage
- 385 Property Damage Product Liability

BANKRUPTCY
- 422 Appeal 28 USC 158
- 423 Withdrawal 28 USC 157

CIVIL RIGHTS
- 441 Voting
- 442 Employment
- 443 Housing/Accommodations
- 444 Welfare
- 445 American with Disabilities - Employment
- 446 American with Disabilities - Other
- 440 Other Civil Rights

PRISONER PETITIONS
- 510 Motions to Vacate Sentence Habeas Corpus
- 530 General
- 535 Death Penalty
- 540 Mandamus/ Other
- 550 Civil Rights
- 555 Prison Condition

FORFEITURE / PENALTY
- 610 Agriculture
- 620 Other Food & Drug
- 625 Drug Related Seizure of Property 21 USC 881
- 630 Liquor Laws
- 640 R.R. & Truck
- 650 Airline Regs
- 660 Occupational Safety /Health
- 690 Other

LABOR
- 710 Fair Labor Standards Act
- 720 Labor/Mgmt. Relations
- 730 Labor/Mgmt. Reporting & Disclosure Act
- 740 Railway Labor Act
- 790 Other Labor Litigation
- 791 Empl. Ret. Inc. Security Act

PROPERTY RIGHTS
- 820 Copyrights
- X 830 Patent
- 840 Trademark

SOCIAL SECURITY
- 861 HIA (1395ff)
- 862 Black Lung (923)
- 863 DIWC/DIWW (405(g))
- 864 SSID Title XVI
- 865 RSI (405(g))

FEDERAL TAX SUITS
- 870 Taxes (U.S. Plaintiff or Defendant)
- 871 IRS-Third Party 26 USC 7609

CV 13 - 00801 SPx

FOR OFFICE USE ONLY: Case Number: ______________________

AFTER COMPLETING THE FRONT SIDE OF FORM CV-71, COMPLETE THE INFORMATION REQUESTED BELOW.

FEB - 5 2013

UNITED STATES DISTRICT COURT, CENTRAL DISTRICT OF CALIFORNIA
CIVIL COVER SHEET

VIII(a). IDENTICAL CASES: Has this action been previously filed in this court and dismissed, remanded or closed? X No Yes
If yes, list case number(s):

VIII(b). RELATED CASES: Have any cases been previously filed in this court that are related to the present case? X No Yes
If yes, list case number(s):

Civil cases are deemed related if a previously filed case and the present case:
(Check all boxes that apply)
A. Arise from the same or closely related transactions, happenings, or events; or
B. Call for determination of the same or substantially related or similar questions of law and fact; or
C. For other reasons would entail substantial duplication of labor if heard by different judges; or
D. Involve the same patent, trademark or copyright, and one of the factors identified above in a, b or c also is present.

IX. VENUE: (When completing the following information, use an additional sheet if necessary.)

(a) List the County in this District; California County outside of this District; State if other than California; or Foreign Country, in which EACH named plaintiff resides.
Check here if the government, its agencies or employees is a named plaintiff. If this box is checked, go to item (b).

County in this District:*	California County outside of this District; State, if other than California; or Foreign Country
	dw-link Incorporated - Dukes County, MA

(b) List the County in this District; California County outside of this District; State if other than California; or Foreign Country, in which EACH named defendant resides.
Check here if the government, its agencies or employees is a named defendant. If this box is checked, go to item (c).

County in this District:*	California County outside of this District; State, if other than California; or Foreign Country
Giant Bicycle, Inc. - Ventura County, CA	Giant Manufacturing Co., Ltd. - Taichung, Taiwan

(c) List the County in this District; California County outside of this District; State if other than California; or Foreign Country, in which EACH claim arose.
Note: In land condemnation cases, use the location of the tract of land involved.

County in this District:*	California County outside of this District; State, if other than California; or Foreign Country
Plaintiff's Claim - Ventura County, CA	

*** Los Angeles, Orange, San Bernardino, Riverside, Ventura, Santa Barbara, or San Luis Obispo Counties**
Note: In land condemnation cases, use the location of the tract of land involved

X. SIGNATURE OF ATTORNEY (OR PRO PER): *[signature]* Date 2/5/2013

Notice to Counsel/Parties: The CV-71 (JS-44) Civil Cover Sheet and the information contained herein neither replace nor supplement the filing and service of pleadings or other papers as required by law. This form, approved by the Judicial Conference of the United States in September 1974, is required pursuant to Local Rule 3-1 is not filed but is used by the Clerk of the Court for the purpose of statistics, venue and initiating the civil docket sheet. (For more detailed instructions, see separate instructions sheet.)

Key to Statistical codes relating to Social Security Cases:

Nature of Suit Code	Abbreviation	Substantive Statement of Cause of Action
861	HIA	All claims for health insurance benefits (Medicare) under Title 18, Part A, of the Social Security Act, as amended. Also, include claims by hospitals, skilled nursing facilities, etc., for certification as providers of services under the program. (42 U.S.C. 1935FF(b))
862	BL	All claims for "Black Lung" benefits under Title 4, Part B, of the Federal Coal Mine Health and Safety Act of 1969. (30 U.S.C. 923)
863	DIWC	All claims filed by insured workers for disability insurance benefits under Title 2 of the Social Security Act, as amended; plus all claims filed for child's insurance benefits based on disability. (42 U.S.C. 405(g))
863	DIWW	All claims filed for widows or widowers insurance benefits based on disability under Title 2 of the Social Security Act, as amended. (42 U.S.C. 405(g))
864	SSID	All claims for supplemental security income payments based upon disability filed under Title 16 of the Social Security Act, as amended.
865	RSI	All claims for retirement (old age) and survivors benefits under Title 2 of the Social Security Act, as amended. (42 U.S.C. (g))

AO 440 (Rev. 06/12) Summons in a Civil Action

UNITED STATES DISTRICT COURT

for the

Central District of California

dw-link Incorporated,
Plaintiff(s)
v.

Giant Bicycle, Inc., and Giant Manufacturing Co., Ltd.,
Defendant(s)

Civil Action No. CV 13 - 00801 PA SPx

SUMMONS IN A CIVIL ACTION

To: *(Defendant's name and address)*

A lawsuit has been filed against you.

Within 21 days after service of this summons on you (not counting the day you received it) — or 60 days if you are the United States or a United States agency, or an officer or employee of the United States described in Fed. R. Civ. P. 12 (a)(2) or (3) — you must serve on the plaintiff an answer to the attached complaint or a motion under Rule 12 of the Federal Rules of Civil Procedure. The answer or motion must be served on the plaintiff or plaintiff's attorney, whose name and address are: Arthur Beeman
SNR Denton US LLP
525 Market Street, 26th Floor
San Francisco, CA 94105

If you fail to respond, judgment by default will be entered against you for the relief demanded in the complaint. You also must file your answer or motion with the court.

TERRY NAFISI

CLERK OF COURT

Date: FEB - 5 2013

J. MURRAY

Signature of Clerk or Deputy Clerk

(1134)

UNITED STATES DISTRICT COURT

for the

Central District of California

dw-link Incorporated, *Plaintiff(s)* v. Giant Bicycle, Inc. and Giant Manufacturing Co., Ltd. *Defendant(s)*	)	Civil Action No.

SUMMONS IN A CIVIL ACTION

To: *(Defendant's name and address)*
Giant Bicycle, Inc.
3587 Old Conejo Road
Newbury Park, CA 91320

A lawsuit has been filed against you.

Within 21 days after service of this summons on you (not counting the day you received it) — or 60 days if you are the United States or a United States agency, or an officer or employee of the United States described in Fed. R. Civ. P. 12 (a)(2) or (3) — you must serve on the plaintiff an answer to the attached complaint or a motion under Rule 12 of the Federal Rules of Civil Procedure. The answer or motion must be served on the plaintiff or plaintiff's attorney, whose name and address are: Arthur Beeman
SNR Denton US LLP
525 Market Street, 26th Floor
San Francisco, CA 94105

If you fail to respond, judgment by default will be entered against you for the relief demanded in the complaint. You also must file your answer or motion with the court.

CLERK OF COURT

Date: ______________________

____
Signature of Clerk or Deputy Clerk

Civil Action No.

PROOF OF SERVICE

(This section should not be filed with the court unless required by Fed. R. Civ. P. 4 (l))

This summons for *(name of individual and title, if any)* ______________________
was received by me on *(date)* ______________ .

☐ I personally served the summons on the individual at *(place)* ______________________
______________________ on *(date)* ______________ ; or

☐ I left the summons at the individual's residence or usual place of abode with *(name)* ______________
______________________ , a person of suitable age and discretion who resides there,
on *(date)* ______________ , and mailed a copy to the individual's last known address; or

☐ I served the summons on *(name of individual)* ______________________ , who is
designated by law to accept service of process on behalf of *(name of organization)* ______________
______________________ on *(date)* ______________ ; or

☐ I returned the summons unexecuted because ______________________ ; or

☐ Other *(specify)*:

My fees are $ __________ for travel and $ __________ for services, for a total of $ 0.00 .

I declare under penalty of perjury that this information is true.

Date: ______________

____
Server's signature

____
Printed name and title

____
Server's address

Additional information regarding attempted service, etc:

UNITED STATES DISTRICT COURT

for the
Central District of California

dw-link Incorporated,
Plaintiff(s)
v.

Giant Bicycle, Inc. and Giant Manufacturing Co., Ltd.
Defendant(s)

Civil Action No.

SUMMONS IN A CIVIL ACTION

To: *(Defendant's name and address)*
Giant Manufacturing Co., Ltd.
19 Shun-Farn Road
Taichung, Taiwan

A lawsuit has been filed against you.

Within 21 days after service of this summons on you (not counting the day you received it) — or 60 days if you are the United States or a United States agency, or an officer or employee of the United States described in Fed. R. Civ. P. 12 (a)(2) or (3) — you must serve on the plaintiff an answer to the attached complaint or a motion under Rule 12 of the Federal Rules of Civil Procedure. The answer or motion must be served on the plaintiff or plaintiff's attorney, whose name and address are: Arthur Beeman
SNR Denton US LLP
525 Market Street, 26th Floor
San Francisco, CA 94105

If you fail to respond, judgment by default will be entered against you for the relief demanded in the complaint. You also must file your answer or motion with the court.

CLERK OF COURT

Date: ______________________

____
Signature of Clerk or Deputy Clerk

Civil Action No.

PROOF OF SERVICE

(This section should not be filed with the court unless required by Fed. R. Civ. P. 4 (l))

This summons for *(name of individual and title, if any)* ______________________
was received by me on *(date)* ____________ .

☐ I personally served the summons on the individual at *(place)* ______________________
______________________ on *(date)* ____________ ; or

☐ I left the summons at the individual's residence or usual place of abode with *(name)* ____________
______________________ , a person of suitable age and discretion who resides there,
on *(date)* ____________ , and mailed a copy to the individual's last known address; or

☐ I served the summons on *(name of individual)* ______________________ , who is
designated by law to accept service of process on behalf of *(name of organization)* ____________
______________________ on *(date)* ____________ ; or

☐ I returned the summons unexecuted because ______________________ ; or

☐ Other *(specify)*:

My fees are $ ________ for travel and $ ________ for services, for a total of $ 0.00 .

I declare under penalty of perjury that this information is true.

Date: ____________

____
Server's signature

____
Printed name and title

____
Server's address

Additional information regarding attempted service, etc: