

(19) World Intellectual Property
Organization
International Bureau



(43) International Publication Date
13 October 2005 (13.10.2005)

PCT

(10) International Publication Number
WO 2005/094480 A2

(51) International Patent Classification: Not classified

(21) International Application Number:
PCT/US2005/009593

(22) International Filing Date: 21 March 2005 (21.03.2005)

(25) Filing Language: English

(26) Publication Language: English

(30) Priority Data:
60/555,524 23 March 2004 (23.03.2004) US

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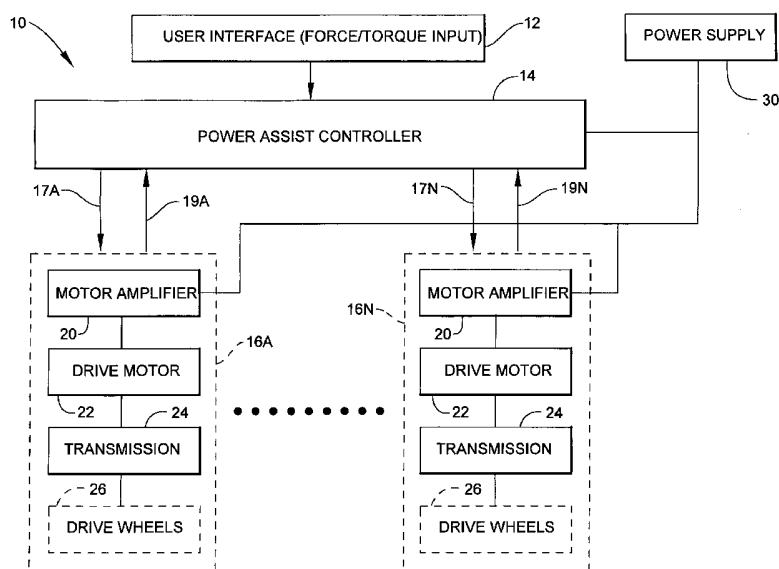
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(81) Designated States (unless otherwise indicated, for every kind of national protection available): AE, AG, AL, AM, AT, AU, AZ, BA, BB, BG, BR, BW, BY, BZ, CA, CH, CN, CO, CR, CU, CZ, DE, DK, DM, DZ, EC, EE, EG, ES, FI, GB, GD, GE, GH, GM, HR, HU, ID, IL, IN, IS, JP, KE, KG, KP, KR, KZ, LC, LK, LR, LS, LT, LU, LV, MA, MD, MG, MK, MN, MW, MX, MZ, NA, NI, NO, NZ, OM, PG, PH, PL, PT, RO, RU, SC, SD, SE, SG, SK, SL, SM, SY, TJ, TM, TN, TR, TT, TZ, UA, UG, US, UZ, VC, VN, YU, ZA, ZM, ZW.

(84) Designated States (unless otherwise indicated, for every kind of regional protection available): ARIPO (BW, GH, GM, KE, LS, MW, MZ, NA, SD, SL, SZ, TZ, UG, ZM, ZW), Eurasian (AM, AZ, BY, KG, KZ, MD, RU, TJ, TM), European (AT, BE, BG, CH, CY, CZ, DE, DK, EE, ES, FI, FR, GB, GR, HU, IE, IS, IT, LT, LU, MC, NL, PL, PT, RO, SE, SI, SK, TR), OAPI (BF, BJ, CF, CG, CI, CM, GA, GN, GQ, GW, ML, MR, NE, SN, TD, TG).

[Continued on next page]

(54) Title: POWER ASSIST DEVICE



(57) Abstract: A power assist device to assist a user in effecting movement of a wheeled device. The power assist device has a user interface configured to be gasped by a user. The user interface measures up to six degrees of freedom forces and torque applied to the user interface and converts measured force and torque into electronic signals. The power assist device includes a controller having a processing resource to process the electronic signals. The controller provides at least one velocity reference signal representing a desired velocity. The velocity servo has an input for receiving the velocity reference signal and outputs a velocity feedback signal representing an actual velocity. The processing resource is configured to process the electronic signals outputted by the user interface and the velocity feedback signal to generate the velocity signal.



Published:

— without international search report and to be republished
upon receipt of that report

For two-letter codes and other abbreviations, refer to the "Guidance Notes on Codes and Abbreviations" appearing at the beginning of each regular issue of the PCT Gazette.

1 DESCRIPTION

2
3 POWER ASSIST DEVICE

4
5 Technical Field

6 The present invention generally relates to a power
7 assist device.

8 Background Art

9 Power assist devices and vehicles are known in the
10 art and are described in U.S. Patent Nos. 6,459,962,
11 6,302,226, 6,230,831, 6,202,773, 5,818,189 and 4,339,013.

12
13 Disclosure of the Invention

14 The present invention is directed to a power assist
15 device that is applicable to such devices as lawn mowers,
16 carts, hospital gurneys, etc. In accordance with the
17 invention, the power assist device comprises a user
18 interface and a power assist controller. The user
19 interface measures the force and/or torque interaction
20 between the user and the power assist device. The input
21 force and/or torque applied by the user to the user
22 interface is measured and converted to electronic signals
23 that are inputted into the power assist controller. The
24 power assist device further comprises at least one
25 velocity servo that has a drive motor, a transmission and
26 a motor driven wheel. The input force imparted by the

1 user on the user interface directly acts on the power
2 assist device, either in parallel with or through the
3 drive motor and transmission, and works in conjunction
4 with the motive power developed by the motor driven
5 wheels.

6 In accordance with the invention, the power assist
7 controller operates on the user input force and/or torque
8 using one or more control algorithms to produce velocity
9 references for the velocity servo. The algorithms employ
10 one or more mathematical models of "ideal" devices,
11 characterized by a desired mass parameter and desired
12 drag law and value. The mathematical model of the ideal
13 device is used in conjunction with the measured user
14 input forces and/or torques and the actual velocity to
15 produce a desired velocity for the velocity servos and
16 can also command an additive torque (feed-forward
17 torque). Standard servo loop technology, using an
18 electric motor, or hydraulic or pneumatic actuators, is
19 employed to minimize the error between desired velocity
20 and actual velocity for each velocity servo.

21 A power supply provides electrical power to the user
22 interface, the power assist controller and each velocity
23 servo. The power supply may be configured as a battery
24 bank, or an engine driven alternator in conjunction with
25 a voltage regulator and battery.

1 Thus, in one embodiment, the present invention is
2 directed to a power assist device to assist a user in
3 effecting movement of a wheeled device, comprising a user
4 interface configured to be grasped by a user and measure
5 up to six degrees of freedom forces and torque applied to
6 the user interface by the user and convert the measured
7 forces and torque into electronic signals. The power
8 assist device includes a controller having a processing
9 resource to process the electronic signals. The
10 controller provides at least one velocity reference
11 signal representing a desired velocity. The power assist
12 device further comprises at least one velocity servo
13 having an input for receiving the at least one velocity
14 reference signal wherein the at least one velocity servo
15 outputs a velocity feedback signal representing an actual
16 velocity. The processing resource of the controller is
17 configured to process the electronic signals outputted by
18 the user interface and the velocity feedback signal to
19 provide the at least one velocity signal.

20 In a related aspect, the present invention is
21 directed to a method for assisting a user in effecting
22 movement of a wheeled device comprising the steps of
23 providing a wheeled device having a user interface
24 configured to be grasped by a user and at least one
25 velocity servo for assisting in the movement of the
26 wheeled device wherein the user interface is configured

1 to measure forces and torque applied thereto by a user,
2 and measuring up to six degrees of freedom forces and
3 torque applied to the user interface. The method further
4 comprises the steps of converting the measured forces and
5 torque into electronic signals, processing the electronic
6 signals to provide at least one velocity reference signal
7 representing a desired velocity, and inputting the at
8 least one velocity reference signal into the at least one
9 velocity servo. The method further comprises generating
10 a velocity feedback signal representative of an actual
11 velocity of the wheeled device, and processing the
12 velocity reference signal and the velocity feedback
13 signal to minimize error between these signals and
14 provide an updated velocity reference signal.

15

16 Brief Description of the Drawings

17 The features of the invention are believed to be
18 novel. The figures are for illustration purposes only and
19 are not drawn to scale. The invention itself, however,
20 both as to organization and method of operation, may best
21 be understood by reference to the detailed description
22 which follows taken in conjunction with the accompanying
23 drawings in which:

24 FIG. 1 is a block diagram of the power assist device
25 of the present invention;

1 FIG. 2 is a diagram illustrating three-axis force
2 measurement provided by a sensor of the power assist
3 device of the FIG. 1;

4 FIGS. 3A and 3B are diagrams illustrating forces
5 that are applied to a handle that is part of a user
6 interface of the power assist device of the present
7 invention;

8 FIG. 4A is a top plan view of a flexure device
9 utilized by the power assist device of the present
10 invention;

11 FIG. 4B is a cross-sectional view along line 4B-4B
12 of FIG. 4A;

13 FIG. 4C is a diagram illustrating the movement of a
14 flexure member shown in FIG. 4A in response to a force
15 applied thereto;

16 FIG. 5 is a block diagram of a power assist
17 controller shown in FIG. 1;

18 FIG. 6A is a diagram illustrating a two-wheel
19 vehicle and a corresponding base coordinate frame;

20 FIG. 6B is a side view taken along line 6B-6B of
21 FIG. 6A;

22 FIG. 7 is a diagram illustrating a corresponding
23 sensor coordinate frame in conjunction with the two-wheel
24 vehicle and base coordinate frame shown in FIG. 6A;

25 FIG. 8 is a block diagram illustrating the algorithm
26 implemented by a central processing unit shown in FIG. 5;

1 FIG. 9 is a block diagram of a dynamic model shown
2 in FIG. 5;

3 FIG. 10 is a diagram illustrating two-dimensional
4 drive velocity allocation;

5 FIG. 11 is a block diagram illustrating a power
6 generation scheme to supply power to the power assist
7 device in accordance with one embodiment of the present
8 invention;

9 FIG. 12A is a flow chart illustrating a power assist
10 supervisory control methodology in accordance with
11 another embodiment of the invention;

12 FIG. 12B is a diagram illustrating hand truck pitch
13 angles;

14 FIGS. 13A and 13B are diagrams that illustrate the
15 coordinate frame and force definitions used with respect
16 to a two-wheeled device;

17 FIG. 14 is a plot that shows three curves for the
18 pitch moment as a function of pitch angle for identical
19 load weights with different locations for the center of
20 gravity (CG) of a two-wheeled device;

21 FIG. 15 is a plot that shows experimental data for
22 the moment, as measured at the instrumented handle of the
23 two-wheeled device, as a function of pitch angle for
24 three different loads;

1 FIG. 16 is a diagram that shows the additional
2 dimensions needed to specify the moment supplied by the
3 user of the two-wheeled device; and

4 FIG. 17 is a block diagram that illustrates
5 implementation of the method of the present invention on
6 a two-wheeled device wherein such implementation is based
7 on the linear model case.

8

9 Modes For Carrying Out The Invention

10 The power assist device of the present invention can
11 be applied to many single and multi-wheel devices such as
12 wheelbarrows, lawn mowers, hand trucks, carts, hospital
13 gurneys, etc.

14 Referring to FIG. 1, there is shown a block diagram
15 of power assist device 10 in accordance with one
16 embodiment of the present invention. Power assist device
17 10 generally comprises user interface 12, power assist
18 controller 14 and velocity servos 16A-N. Thus, it is to
19 be understood that although FIG. 1 shows up to N velocity
20 servos 16, there can be just one velocity servo, or there
21 can be a plurality N velocity servos. User interface 12
22 measures the force and/or torque interaction between the
23 user and power assist device 10. As will be described in
24 the ensuing description, user interface 12 is part of or
25 integral with the handle or handles of a wheeled device
26 with which power assist device 10 is used. The handle or

1 handles are grasped by a user or operator. User
2 interface 12 converts the measured input force and/or
3 torque applied to the handle by the user into electronic
4 signals. These electronic signals are inputted into
5 power assist controller 14. Power assist controller 14
6 operates on the user input force and/or torque using one
7 or more control algorithms to produce velocity reference
8 signals 17A-N that are inputted into corresponding
9 velocity servos 16A-N, respectively. The algorithms
10 employ one or more mathematical models of "ideal"
11 devices, characterized by a desired mass parameter and
12 desired drag law and value. Such suitable algorithms are
13 described in U.S. Patent No. 6,459,962, the disclosure of
14 which patent is incorporated herein by reference. Each
15 velocity servo 16A-N outputs a corresponding velocity
16 feedback signal 19A-N, respectively. The mathematical
17 model of the ideal device is used in conjunction with the
18 measured user input forces and/or torques and the actual
19 velocity (e.g. velocity feedback signals 19A-N) to
20 produce a desired velocity (e.g. velocity reference
21 signals 17A-N) for velocity servos 16A-N. The
22 mathematical model of the ideal device is also used in
23 the presence of an additive torque (feed-forward torque).

24 In one embodiment of the invention, each velocity
25 servo 16A-N utilizes standard servo loop technology to
26 minimize the error between desired velocity and actual

1 velocity for each velocity servo 16A-N. In one
2 embodiment of the invention, each velocity servo 16A-N
3 comprises motor amplifier 20, electric drive motor 22,
4 transmission 24, drive wheels 26 (i.e. motor driven
5 wheels) or hydraulic or pneumatic actuators (not shown).
6 Thus, the input force imparted by the user on user
7 interface 12 directly acts on the power assist device
8 either in parallel with or through the electric drive
9 motor 22 and transmission 24, and works in conjunction
10 with the motive power developed by the drive wheels 26.
11 Power supply 30 provides electrical power to user
12 interface 12, power assist controller 14 and each
13 velocity servo 16A-N. Power supply 30 can be configured
14 as a battery bank or an engine driven alternator in
15 conjunction with a voltage regulator and battery.

16 Referring to FIG. 1, user interface 12 measures the
17 forces and/or torque interaction between the user and the
18 power assist device and converts the measured forces
19 and/or torque to electronic signals that are inputted to
20 power assist controller 14. The forces and/or torques
21 measured by user interface 12 are processed by power
22 assist controller 14 to generate the velocity reference
23 signals 16A-N that are inputted into velocity servos 16A-
24 N, respectively. In one embodiment, user interface 12
25 measures the six degrees of freedom (DOF) forces and
26 torques applied by the user or operator. The user or

1 operator grasps the handle (which is part of user
2 interface 12), and the forces and/or torques applied to
3 the handle are measured by a force/torque sensor of user
4 interface 12. (The term "sensor handle" is used in the
5 ensuing description to refer to a handle having the
6 aforementioned sensor). For devices in which pitch and
7 roll can vary (i.e. devices without rigid frames or with
8 fewer than three wheels on the ground), the pitch and
9 roll of the device must be measured in order to transform
10 the measured forces and moments into a frame that is
11 fixed relative to the gravity vector. In many cases, the
12 full six DOF forces and torques applied at the sensor
13 handle will not be required by power assist controller
14 14. In most power assist applications, measurement of
15 the two or three components of orthogonal forces at the
16 sensor handle will meet the requirements of the
17 application. For devices with rigid or suspended frames
18 and three or more wheels on the ground (such as a cart or
19 hospital gurney), the force in the forward/aft direction
20 must be measured to control forward/aft motion and either
21 the lateral force at the sensor, the moment about the
22 vertical axis of the sensor, or both, must be measured to
23 control steering. If the device has lateral mobility,
24 then the lateral force at the sensor will be measured to
25 control the lateral motion. For a fixed stability device
26 of this nature, pitch and roll measurements are not

1 needed since the sensor remains nearly fixed in
2 orientation relative to gravity. For devices with two
3 wheels, such as a hand truck, the net force in the
4 forward/aft direction must be measured in the presence of
5 variable orientation relative to gravity. Typically,
6 this involves transforming two axes of force at the
7 sensor handle into a horizontal force using a measure of
8 the pitch of the hand truck. Steering control is enacted
9 through either lateral force at the handle or moment
10 about the vertical axis at the handle. In general, the
11 device pitch measurement will be required to transform
12 the handle sensor measurements. In the most general
13 case, the sensor that is used to measure the forces and
14 torques at user interface 12 is configured to measure the
15 three orthogonal components of force at the handle of
16 user interface 12. Other measurement configurations can
17 be developed from the general case via reconfiguration of
18 the basic elements and methodologies described herein.

19 Referred to FIG. 2, there is shown a simplified
20 representation of a sensor handle that provides the
21 necessary measurement of three-axis forces. Handle 100
22 is attached to triangular handle plate 101. At each
23 vertex of the triangular handle plate 101, a spring
24 element 103 is coupled to the matching vertex of an
25 opposing triangular base plate 102. A total of three
26 spring elements 103 connect the handle plate 101 to the

1 base plate 102. In practice, base plate 102 is attached
2 to the power assist device. If spring elements 103 are
3 constrained to provide motion only in the direction of
4 positive and negative X axis as shown, then forces in the
5 positive X direction applied to handle 100 will generally
6 compress spring elements 103. Forces applied in the
7 negative X axis direction will extend spring elements
8 103. Forces applied to handle 100 in the positive
9 vertical direction (positive Z axis direction) will cause
10 the upper spring elements to compress, and the lower
11 spring element to extend due to the moment imparted by
12 the vertical force. Forces applied to handle 100 in the
13 negative vertical direction will cause the reverse, i.e.,
14 the upper spring elements will extend and the lower
15 spring element will compress. Finally, lateral forces
16 applied at handle 100 in the positive or negative Y axis
17 direction will cause compression of the upper spring
18 element in the direction of the lateral force and
19 extension of the other upper spring element due to the
20 moment imparted by the lateral force. Combinations of
21 forces applied in the X, Y and Z axis directions will in
22 general cause spring elements 103 to compress or extend
23 depending on the magnitude and direction of the net
24 applied force. If the displacement exhibited during
25 compression or extension of the spring elements 103 at
26 each vertex of handle plate 101 is measured at various

1 levels of known applied force, then a calibration can be
2 developed for each spring element 103. This calibration
3 would provide a relationship between applied force and
4 displacement for each spring element 103. Using the
5 calibration for each spring element 103 when an unknown
6 force is applied to handle 100, the measured
7 displacements of the three spring elements 103 will yield
8 the forces at each of the attachment points for spring
9 elements 103. For a given geometry of handle 100 and
10 handle plate 101, knowledge of forces at the three
11 locations will allow computation of the three orthogonal
12 force components.

13 Referring to FIGS. 3A and 3B, handle 100 and handle
14 plate 101 are illustrated in isometric view. The three
15 known forces at the vertices of the handle plate 101 are
16 labeled as F_1 , F_2 and F_3 . FIG. 3B shows a side view of
17 handle 100 and handle plate 101, and the forces F_1 , F_2 ,
18 and F_3 acting on handle plate 101. The external forces
19 applied to handle 100 by the user are illustrated as F_x
20 and F_z , representing the two orthogonal components of the
21 external force to be measured. For the purposes of
22 facilitating understanding of this aspect of the
23 invention, it is assumed that the third component of
24 force, F_y , is zero (this assumption in no way affects the
25 validity or generality of the approach). The handle
26 geometry is identified as follows: d_1 is the height

1 between the application points of the two top forces F_1
2 and F_2 , and the normal to handle plate 101 through the
3 external force, d_2 is the height between the bottom force
4 application point and the normal to handle plate 101
5 through the external force, and w is the distance of the
6 plate normal to external force. The external forces F_x
7 and F_z are computed using statics as shown by equations
8 (1) and (2):

9

$$10 \quad F_x = F_1 + F_2 + F_3 \quad (1)$$

$$11 \quad F_z = (F_3 \times d_2 - (F_1 + F_2) \times d_1) \div w \quad (2)$$

12

13 In the foregoing discussion, the assumption was made that
14 spring elements 103 are constrained to motion only in the
15 direction of the positive and negative X axis direction
16 (see FIG. 2). This requires that the compliance of the
17 spring elements 103 be infinitely stiff in the Y axis and
18 Z axis directions. Further, if handle plate 101 is rigid
19 with fixed dimensions between spring element attachment
20 points, the motion of handle plate 101 will be limited to
21 pure translation motion with all spring attachment points
22 moving the same distance. This is due to the fact that
23 any differential in motion between the spring attachment
24 points results in a lengthening or shortening of the
25 distances between attachment points. Physically, this
26 can occur only if the handle plate dimensions change

1 between attachment points, or motion of the spring
2 attachment points in the Y and Z axis directions is
3 permitted. Additionally, differential motion between the
4 spring attachment points will require that the angle
5 between spring elements 103 and handle plate 101 be
6 unconstrained, for example, as would be the case if the
7 attachments to handle plate 101 were made using ball
8 joints. One method to allow differential motion of the
9 spring element attachment points to handle plate 101 in
10 the presence of the infinitely stiff springs in the off-
11 axis directions and the fixed dimensions of handle plate
12 101 is to provide for some nominal compliance at the
13 attachment point. Using compliant elements, similar to
14 engine isolation mounts, between spring elements 103 and
15 handle plate 101 permit the necessary dimensional changes
16 and rotational freedom.

17 Referring to FIGS. 3A and 3B, spring elements 103
18 are required to exhibit desired compliance
19 characteristics for the intended direction of motion yet
20 provide near infinite compliance for off-axis motions.
21 In accordance with the invention, a flexure device that
22 exhibits these characteristics is shown in FIGS. 4A and
23 4B. The flexure device comprises housing 104, flexure
24 member 105, force attachment 106 and flexure root
25 attachment 107. Housing 104 defines recess 108 within
26 which flexure member 105 flexes. Specifically, flexure

1 member 105 flexes under the influence of externally
2 applied force F . Flexure member 105 is a blade type
3 element with the blade root attached to housing 104 via
4 attachments 107. Flexure member 105 has end 105a. The
5 force is directed to end 105a of flexure member 105 via
6 force attachment 106. Recess 108 allows flexing of
7 flexure member 105 without contacting housing 104. The
8 thickness T of flexure member 105 is selected to provide
9 the desired compliance characteristics. The height h of
10 flexure member 105 is selected to provide the desired
11 high off-axis stiffness (force stiffness and torsional
12 stiffness). Housing 104 serves to provide a method for
13 attaching the root of flexure member 105 as well as a
14 means of providing overload protection. Referring to
15 FIG. 4C, the inner recess 108 in housing 104 is designed
16 to limit the travel of flexure member 105 at some
17 predetermined force, and therefore displacement value.
18 This overload protection mechanism prevents plastic
19 deformation of flexure member 105 or damage to the
20 sensing elements under loads that exceed the nominal
21 sensing range. Specifically, housing recess 108 is
22 shaped to permit flexure member 105 to flex without
23 contacting housing 104 for a range of deflections up to
24 displacement d_0 . At displacement d_0 , occurring at a force
25 of F_0 , flexure member 105 contacts the housing walls
26 defining housing recess 108 for a fraction of the length

1 of flexure member 105. For applied forces greater than
2 F_0 , the effective flexure length is diminished, and the
3 stiffness increases considerable. Thus, the shape of
4 recess 108 can also be used to shape the resulting
5 compliance characteristics of flexure member 105. A
6 typical force-versus-displacement relationship for
7 flexure member 105 is also shown in FIG. 4C. At
8 displacement d_0 , the compliance (slope) changes due to the
9 effective shortening of the flexure length. This
10 characteristic is very desirable for the power assist
11 application since the sensor can be made very sensitive
12 at low forces (high sensitivity defined as large
13 displacement per applied force), yet provide a wide
14 sensing range (at reduced sensitivity) for large forces.
15 The displacement d of flexure end 105a can be measured
16 using commercially available non-contact displacement
17 measurement technology, such as linear hall-effect
18 devices, optical devices, or capacitive measurement
19 devices. For example, in one embodiment, one or more
20 magnets can be mounted to flexure end 105a, and a linear
21 hall-effect integrated circuit can be mounted to housing
22 104 in proximity to the magnets. Such a configuration
23 results in a near linear voltage-versus-displacement
24 relationship. As a result of calibrating the transducer
25 output voltage or current as a function of known values
26 of applied weights, a calibration curve for each of the

1

2 flexure based spring elements can be developed.

3 Programming the calibrations into a programmable central

4 processor unit (CPU), which is described in the ensuing

5 description, permits the sampled voltages or currents

6 from each flexure sensor to be turned into a force

7 magnitude value. The direction of the applied force for

8 small motions of a typical flexure is normal to the

9 direction of flexure motion. Thus, given a fixed

10 mounting configuration for the flexure members, the

11 direction of the applied force is fixed relative to the

12 frame or body of the power assisted device.

13 Referring to FIG. 5, there is shown a block diagram

14 of power assist controller 14. Power assist controller

15 14 is similar to the power assist controller described in

16 U.S. Patent No. 6,459,962, which patent was cited in the

17 foregoing description. However, power assist controller

18 14 is configured to operate in three dimension scenarios,

19 covering devices that are capable of six degrees of

20 freedom (DOF) motion in 3-D space rather than planar

21 motion alone. Power assist controller 14 generally

22 comprises central processing unit 200 (CPU), user

23 interface force transducer 202, orientation transducer

24 203, motor velocity transducer(s) 204, and dynamic model

25 205. In one embodiment, central processing unit 200

26 further comprises external computer 209 that is connected

1 to CPU 200 via wireless Ethernet link. The purpose of
2 computer 209 is described in the ensuing description.
3 Power supply 30 (see FIG. 1) supplies power to CPU 200
4 and associated control equipment. CPU 200 provides
5 velocity reference signals 17A-N which are inputted into
6 velocity servos 16A-16N, respectively, as described in
7 the foregoing description. In a preferred embodiment,
8 CPU 200 comprises a programmable microprocessor. CPU 200
9 provides the control system operation. Circuitry for the
10 control system is preferably housed in an electronics
11 control box (not shown) which is integrated with or
12 mounted to the frame of power assist device 10. CPU 200
13 interfaces to user interface force transducers 202 and
14 orientation sensor 203, and computes velocity reference
15 signals 17A-N for velocity servos 16A-N in response to
16 measured forces and/or torque at user interface 12.
17 Dynamic model 205 provides dynamic representation of a
18 "reference device" with desired mass and damping
19 properties and is used to generate the velocity
20 references from the user interface forces. The closed
21 loop velocity control uses the error between the velocity
22 reference provided by dynamic model 205 and the measured
23 velocity of motor 22 (see FIG. 1) to control the output
24 of motor 22 such that the error component is reduced to
25 zero under steady state conditions.

1 Referring to FIG. 5, in one embodiment, CPU 200 is
2 interfaced to external computer 209 to provide a two-way
3 wireless data path for reprogramming the software of CPU
4 200, as well as for monitoring the real time operating
5 conditions of the control system. Preferably, external
6 computer 209 comprises the software and hardware
7 necessary for generating the executable control program
8 that runs in CPU 200. In one embodiment, external
9 computer 209 comprises an executable program that is used
10 to provide a real time display of the operating
11 conditions of the control system in graphical or tabular
12 form, for monitoring and control of real time control
13 system parameters such as gains, and for recording the
14 operating conditions for data analysis and review at a
15 later time.

16 A single three dimensional (3-D) mathematical model
17 is used to represent the dynamics of the desired
18 reference device. The 3-D model represents the dynamic
19 behavior of the device in a fixed coordinate frame
20 attached to the device. Referring to FIGS. 6A and 6B,
21 the coordinate frame, referred to as base coordinate
22 frame 211, has an orientation that is fixed relative to
23 an earth reference. A common reference frame used for
24 this representation is a Cartesian coordinate frame with
25 the XY plane normal to the local gravity vector and the Z
26 axis positive in the down direction. Selection of the

1 fixed location of the frame relative to the power assist
2 device is not critical, although it is usually chosen to
3 simplify the dynamic model representation. Figure 6A
4 shows a device having two wheels 210. The base
5 coordinates of base coordinate frame 211 are chosen with
6 the Y axis through the center of rotation of wheels 210,
7 and the X axis at the mid-point of the wheel-to-wheel
8 spacing. All measured forces applied to the two-wheeled
9 device through the user interface 12 are resolved into a
10 set of equivalent forces and torques acting on the 3-D
11 model in the base coordinate frame 211. The forces and
12 torques will generate translational and rotational
13 accelerations (respectively) on the 3-D model based on
14 the model's mass and moment of inertia and the net forces
15 acting on the 3-D model. The net forces are the vector
16 sum of the forces applied via user interface 12 (see FIG.
17 1) and the parasitic drag forces resulting from the
18 desired damping values contained in the model.

19 As described in the foregoing description, the force
20 transducers in user interface 12 are comprised of one or
21 more flexure based sensors (e.g. the flexure device shown
22 in FIGS, 4A and 4B) that generate a voltage or current
23 signal as a function of the applied force. The
24 relationship between force and associated output voltage
25 or current is determined by bench or laboratory
26 calibration using weight standards. A single flexure

1 device will provide measure of a linear force at a known
2 geometric relationship to the flexure structure. Thus,
3 the flexure device shown in FIGS. 4A and 4B is preferably
4 located on power assist device at a known and measurable
5 location and orientation relative to some fixed
6 reference(s) on the wheeled device. Thus, the flexure
7 device output will provide measurement of a single force
8 magnitude with known location and orientation relative to
9 the fixed reference(s) on power assist device 10.

10 Referring to FIG. 7, sensor coordinate frame 212 is
11 used to represent the forces F , indicated by reference
12 number 213, applied via user interface 12. Sensor
13 coordinate frame 212 is a Cartesian coordinate frame that
14 remains attached to the sensors, maintaining a fixed
15 orientation relative to user interface 12. As
16 illustrated in FIG. 7, the axes for the sensor coordinate
17 frame 212 are labeled as X_s , Y_s , and Z_s . As the device
18 orientation changes relative to the fixed orientation
19 base coordinate frame 211, the orientation of the sensor
20 coordinate frame 212 also changes relative to the base
21 coordinate frame 211. This is illustrated in FIG. 7 for
22 two values of device pitch angle θ_1 and θ_2 . At the first
23 value of device pitch angle θ_1 corresponding to the
24 vertical orientation, the orientation of the sensor
25 coordinate frame 212 is identical to the orientation of

1 the base coordinate frame 210. As the pitch angle
2 changes to the second pitch angle θ_2 , the orientation of
3 the sensor coordinate frame 212 changes relative to the
4 orientation of base coordinate frame 210. In order to
5 represent the applied forces F , indicated by reference
6 number 213, in base coordinate frame 211, the data must
7 be transformed from sensor coordinate frame 212. A three
8 dimensional coordinate transformation is used to
9 translate and rotate the applied force F 213 from sensor
10 coordinate frame 212 to base coordinate frame 211. In
11 general, the transformation will be a function of the
12 orientation provided by the orientation transducer 203.

13 Referring to FIG. 5, orientation transducer 203 is
14 an accelerometer-based instrument that responds to the
15 local gravity vector, providing a pitch and roll angle
16 relative to the vertical reference. If the
17 accelerometer-based instrument is subject to other
18 accelerations in addition to gravity, the accelerometers
19 in the transducer can lead to dynamic errors in pitch and
20 roll measurement. Compensation for such errors can be
21 accomplished using turning rate sensors about the pitch
22 and roll axes, then combining the accelerometer
23 measurements and turning rate measurements using a Kalman
24 filter. This technique is used by several commercially
25 available orientation sensors. The transducer raw
26 accelerations and turning rates are inputted to CPU 200

1 via analog-to-digital converters (see FIG. 8), then
2 processed by CPU 200 to produce the pitch and roll angles
3 relative to the vertical reference. Simplified versions
4 of this configuration based on uncompensated acceleration
5 measurements may be used in cases where the acceleration
6 errors are small. Such simplified configurations may
7 also be based on other static tilt measurement methods,
8 such as electrolytic transducers, pendulum type
9 transducers, and others.

10 Referring to FIG. 8, there is shown a block diagram
11 of the algorithm that resides in CPU 200. In the
12 generalized case, the 3-D dynamic model 205 (see FIG. 5)
13 operates on the six degree-of-freedom (6-DOF) forces and
14 torques defined in base coordinate system 211, producing
15 a 6-DOF velocity reference also defined in base
16 coordinate system 211. Assuming rigid body dynamics, the
17 velocity reference vector can be mapped to 3-D velocity
18 vectors at the locations of the drive actuators.
19 Depending on the actuator configuration (orientation
20 relative to the rigid body) and capability (degrees of
21 freedom), each individual velocity reference signal 17A-
22 17N (see also FIG. 1) is developed from the 3-D velocity
23 vector at each associated servo or drive location. In
24 general, each velocity servo 16A-N may not be able to
25 develop the full 3-D velocity reference due to limited
26 degrees of freedom. For example, attempting to slide a

1 power assisted hand truck laterally will generate a base
2 coordinate Y axis component of velocity reference acting
3 on the hand truck body. This will result in Y components
4 of velocity reference at each of the actuator locations
5 (wheel contact locations). However, since the Y
6 component of velocity cannot be delivered by the fixed
7 orientation wheeled actuators used on the hand truck,
8 this velocity component will be ignored (i.e. not
9 responded to). In other applications, using alternative
10 actuator configurations (e.g., steered driven wheels),
11 this velocity component may be used to provide lateral
12 motive effort. Applications such as patient gurneys or
13 pallet trucks are candidates for three degree of freedom
14 power assist control that provides power assisted lateral
15 motion in addition to power assisted fore/aft motion and
16 turning motions.

17 The ensuing description describes details of the
18 processing algorithm in FIG. 8 for the generalized 3-D
19 case. Orientation sensor 203 is sampled by analog-to-
20 digital converter 250. An analog, anti-alias low pass
21 filter (not shown) filters signals prior to being
22 inputted into analog-to-digital converter 250 to prevent
23 aliasing of the sampled signal when sampled at a fixed
24 rate by analog-to-digital converter 250. The bandwidth
25 of the aforementioned anti-alias low pass filter is based
26 on the sample rate of analog-to-digital converter 250.

1 Once converted to digital representations, processing
2 resource 252, which is part of CPU 200, processes the
3 sampled signals to provide transducer orientation θ .
4 Specifically, processing resource 252 converts the
5 sampled signals to orientation in engineering units
6 (degrees or radians) in the case of floating point math,
7 or to scaled engineering units in the case of integer
8 math. Similarly, for each force transducer 202 in user
9 interface 12, the transducer voltage or current outputs
10 are processed with a low pass, anti-alias filter and then
11 sampled by analog to digital converter(s) 254.
12 Processing resource 256, which is part of CPU 200,
13 processes the sampled representations to provide forces
14 $\overline{fs_i}$. Specifically, processing resource 256 converts the
15 resulting sampled representations to forces in
16 engineering units (newtons) in the case of floating point
17 math, or to scaled engineering units in the case of
18 integer math. The resulting force variables are
19 representations of the forces applied at user interface
20 12 and represented in sensor coordinate frames 212. In
21 general, there will be one sensor coordinate frame 212
22 per force transducer 202. Forces $\overline{fs_i}$ and the calculated
23 orientation θ are inputted into processing resource 258
24 which is part of CPU 200. Processing resource 258 then
25 transforms each force $\overline{fs_i}$ from a sensor coordinate frame

1 representation to the single base coordinate frame
2 representation 211 (see FIG. 7). In the transformation
3 process, a force in sensor coordinates may, in general,
4 result in forces and torques in the base coordinate
5 frame, thus the force representation in base coordinates
6 is a 6-DOF vector of forces and torques. After all force
7 transducer values are sampled and transformed to base
8 coordinates, processing resource 260 sums the 6-DOF
9 vectors to yield the effective net forces and torques $\overline{f_{net}}$
10 acting on the device (i.e. cart, hand truck, gurney,
11 etc.). Dynamic model 205 operates on the 6 DOF
12 force/torque vector $\overline{f_{net}}$ defined relative to the base
13 coordinate system, to produce a 6 DOF velocity reference
14 vector $\overline{V_{ref}}$ also defined in the base coordinate system.
15 This 6 DOF velocity reference vector $\overline{V_{ref}}$ is then inputted
16 into drive velocity allocation processor 261. Referring
17 to FIG. 9, the dynamic model 205 incorporates mass $\overline{M}$ and
18 damping $\overline{B}$ properties for the desired behavior of power
19 assisted device 10. The 1/s term represents numerical
20 integration that operates on the acceleration vector to
21 produce the velocity reference vector $\overline{V_{ref}}$. The
22 aforementioned $\overline{M}$ and damping $\overline{B}$ properties are defined
23 for each of the six degrees of freedom for the 3-D device
24 model. Referring to FIG. 9, the block diagram shown

1 therein represents six equations wherein each equation
 2 corresponds to one of the six degrees of freedom of the
 3 3-D Dynamic Model. The quantities represented in FIG. 9
 4 are represented by equations 3-6:

$$6 \quad \bar{f}_{net} = \begin{pmatrix} f_{net,x} & f_{net,y} & f_{net,z} & m_{net,x} & m_{net,y} & m_{net,z} \end{pmatrix}^T \quad (3)$$

7
 8 wherein equation (3) represents the 6 DOF vector of
 9 forces and torques in the base coordinate frame;

$$11 \quad \bar{V}_{ref} = \begin{pmatrix} v_{ref,x} & v_{ref,y} & v_{ref,z} & \omega_{ref,x} & \omega_{ref,y} & \omega_{ref,z} \end{pmatrix}^T \quad (4)$$

12
 13 wherein equation (4) represents the velocity reference
 14 vector in the base coordinate frame, with translational
 15 velocities, v , and rotational velocities, ω .

$$17 \quad \bar{M}^{-1} = \begin{pmatrix} 1/M & 0 & 0 & 0 & 0 & 0 \\ 0 & 1/M & 0 & 0 & 0 & 0 \\ 0 & 0 & 1/M & 0 & 0 & 0 \\ 0 & 0 & 0 & 1/J & 0 & 0 \\ 0 & 0 & 0 & 0 & 1/J & 0 \\ 0 & 0 & 0 & 0 & 0 & 1/J \end{pmatrix} \quad (5)$$

18
 19 wherein equation (5) is a 6x6 diagonal matrix containing
 20 the inverse of the PA model mass and inertia.

21

$$\bar{B} = \begin{pmatrix} B_{t,x} & 0 & 0 & 0 & 0 & 0 \\ 0 & B_{t,y} & 0 & 0 & 0 & 0 \\ 0 & 0 & B_{t,z} & 0 & 0 & 0 \\ 0 & 0 & 0 & B_{r,x} & 0 & 0 \\ 0 & 0 & 0 & 0 & B_{r,y} & 0 \\ 0 & 0 & 0 & 0 & 0 & B_{r,z} \end{pmatrix} \quad (6)$$

2

3 wherein equation (6) is a 6x6 diagonal matrix containing
 4 the translational damping terms, B_t , and rotational
 5 damping terms, B_r .

6 The indicated form of the inverse "mass" matrix
 7 shows identical mass terms ($1/M$) for the translational
 8 degrees of freedom, and identical inertia terms ($1/J$) for
 9 the rotational degrees of freedom. In a real physical
 10 system, the mass terms would be identical. However, in
 11 practice, maintaining independent terms for the masses
 12 permits the power assist model to be "adjusted" on a per
 13 axis basis. For example, in an application with full x-y
 14 motion capabilities (e.g., a floor polishing machine),
 15 the x component of mass could be made larger than the y
 16 component of mass to "suppress" the x axis (forward/aft)
 17 motions relative to the y axis (side to side) motions.

18 The drive velocity allocation algorithm converts the
 19 six DOF velocity reference vector $\overline{V_{ref}}$ defined in the base
 20 coordinate system to velocity references (e.g. velocity
 21 reference signals 17A) for each of the actuators (i.e.
 22 servo) defined in the system. This is accomplished by

1 assuming rigid body dynamics with known geometry of the
2 velocity servos or actuators relative to the base
3 coordinate system. For rigid body dynamics, the velocity
4 at any point on the body can be described by the sum of
5 the velocity of a reference point on the body, in this
6 case the origin of the base coordinate system, and the
7 velocity due to rotation of the body about the same
8 reference point. In the general case, the velocity
9 reference is the sum of the translation velocity plus the
10 rotational velocity caused by the off- axis rotations.

11 An example of the dynamic model equations and
12 velocity allocation for a two-wheel device are described
13 below. In this example, it is assumed that each of the
14 two velocity servos are capable of delivering velocities
15 in the base coordinate X axis direction only. Under this
16 assumption, the velocity reference vector X translational
17 component ($v_{ref,x}$) and Z axis rotational component ($\omega_{ref,z}$)
18 are the only elements of the velocity reference vector of
19 which can generate Velocity Servo X axis motions. Due to
20 the diagonal characteristic of the mass and damping
21 matrices of the dynamic model, the force/torque vector X
22 force component ($f_{net,x}$) and Z axis moment ($m_{net,z}$) can
23 generate the velocity reference vector X translational
24 component ($v_{ref,x}$) and Z axis rotational component ($\omega_{ref,z}$).

1 In this case, the Dynamic Model reduces to equations (7)
 2 and (8):

$$3 \quad v_{ref,x} = \left(\frac{1}{Ms + B_{t,x}} \right) * f_{net,x} \quad (7)$$

$$4 \quad \omega_{ref,z} = \left(\frac{1}{Js + B_{r,z}} \right) * m_{net,x} \quad (8)$$

6
 7 Referring to FIG. 10, there is shown a diagram
 8 illustrating drive velocity allocation. The drive
 9 velocity allocation reduces to the two dimensional case
 10 shown. FIG. 10 shows a view of the base coordinate XY
 11 plane (in the direction of the positive Z base coordinate
 12 axis), and the X axis translation velocity $V_{ref,x}$ and the Z
 13 axis rotational velocity $\omega_{ref,z}$. The individual wheel
 14 velocity references are calculated from the X axis
 15 translational velocity $V_{ref,x}$ and Z axis rotational
 16 velocity $\omega_{ref,z}$ and are represented by the set of equations
 17 (9):

$$18 \quad \begin{aligned} 19 \quad v_{ax,l} &= v_{ref,x} + r_{ax,l} * \omega_{ref,z} \\ v_{ax,r} &= v_{ref,x} - r_{ax,r} * \omega_{ref,z} \end{aligned} \quad (9)$$

20
 21 In the event any velocity reference exceeds the maximum
 22 value, it is limited to the maximum value. In this case,

1 all other non-saturated velocity references are limited
2 in magnitude to preserve the direction of the original
3 net force vector in base coordinates.

4 As described in the foregoing description and as
5 shown in FIGS. 1 and 5, power assist device 10 contains
6 one or more velocity servos 16A-N, each comprising motor
7 amplifier 20, electric drive motor 22, transmission 24
8 and drive wheel 26. Each velocity servo 16A-N receives a
9 velocity reference signal 17A-N, respectively, from power
10 assist controller 14, and operates on the velocity
11 reference signals 17A-N so as to minimize the error
12 between actual motor velocity and velocity reference.
13 The actual motor velocities produced by velocity servos
14 16A-N are represented by velocity feedback signals 19A-N,
15 respectively. In a typical embodiment, motor amplifier
16 20 employs standard servo control techniques to provide
17 closed loop velocity control of the motor velocity. The
18 motor velocity servo loop regulates the motor torque to
19 maintain the desired velocity reference in the presence
20 of external disturbances to the velocity, such as drag,
21 friction and changing gravity loads due to sloped
22 terrain. As described in the aforementioned U.S. Patent
23 No. 6,459,962, each motor amplifier 20 operates on a
24 corresponding velocity reference signal (e.g. velocity
25 reference signal 17A) and actual motor velocity (e.g.
26 velocity feedback signal 19A) using a proportional plus

1 integral (PI) compensator to develop the current loop
2 (i.e., torque) reference. Motor amplifier 20 utilizes a
3 closed loop current control inside the closed loop
4 velocity control to maintain the current (torque) at the
5 desired value. The current loop employs a proportional
6 only or proportional plus integral control operating on
7 the current error (difference between the current
8 reference and actual current) to develop the motor
9 voltage command. The motor voltage command signal is a
10 pulse width modulated duty cycle, which is then used to
11 control the power amplifier stage of motor amplifier 20.
12 In one embodiment, this power amplifier stage comprises a
13 PWM H-bridge amplifier.

14 The measured motor velocity may be obtained with a
15 velocity sensor, either an optical encoder or
16 conventional tachometer mounted to the motor shaft. In
17 the case of brushless motors, the sensors used for motor
18 commutation (typically optical or magnetic hall-effect)
19 may be used for providing the measured motor velocity.
20 Optionally, sensor-less measurement may be employed, in
21 which case the motor velocity is calculated using an
22 estimate of the motor back-EMF (BEMF). This estimate is
23 based on measurements of motor current and voltage, and a
24 mathematical model of the motor's electrical and magnetic
25 parameters. Drive motor 22 can be any actuator suitable
26 for providing the required motive power for the device.

1 Typically, brush or brushless DC motors, hydraulic
2 motors, or pneumatic motors may be used depending upon
3 the specific requirements of the application. The
4 details of motor amplifier 20 are similar in each case,
5 although the power amplifier stage of motor amplifier 20
6 may be different depending on the motor type used. For
7 example, a brush motor will employ a conventional four-
8 transistor H bridge, with commutation provided by the
9 motor, whereas a brushless motor will employ one H bridge
10 per motor phase, and the commutation is provided by motor
11 amplifier 20 in conjunction with motor mounted
12 commutation sensors. The motor is directly connected to
13 transmission 24 which is typically a reduction gearbox.
14 The output of transmission 24 is connected to and drives
15 drive wheel 26. The specific configuration of
16 transmission 24 depends upon on the specific torque and
17 speed requirements of the device. Standard reduction
18 gear designs may include worm gears, helical gear sets or
19 planetary gear sets. The transmission may also
20 incorporate a belt and pulley reduction stage, either
21 alone or in conjunction with gear reduction. Alternative
22 transmission configurations may also be used depending on
23 the specific requirements of the application. An example
24 would be a continuously variable transmission (CVT) of
25 which many different configurations exist as known art.
26 The combination of a low friction motor and a low

1 reduction ratio transmission provides minimal drive-train
2 drag in the cases where the motor is un-powered, such as
3 during periods where the battery power is low, or when
4 power assist device 10 is turned off, or the device is
5 free-wheeled without the power assist function enabled.

6 Referring to FIG. 1, each velocity servo 16 can be
7 designed to provide motive effort in a fixed (single)
8 direction using conventional rotational servos coupled to
9 wheels (or tracks), or in continually variable directions
10 using a directional (steered) driven wheel. The
11 directional servo is applicable for devices in which 3-
12 DOF operation is desired, permitting independent lateral
13 motions in addition to the forward-reverse, and turning
14 motions. U.S. Patent No. 6,474,434, entitled "Drive
15 Wheel", the disclosure of which is incorporated herein by
16 reference, describes a device that provides the described
17 rotating as well as turning motions for a single velocity
18 servo. In using a directional velocity servo such as
19 this, the velocity reference from dynamic model 205 (see
20 FIG. 5) will consist of a velocity reference vector
21 defined in base coordinates, with both magnitude and
22 direction. The magnitude component will dictate the
23 rotational speed of the wheel whereas the direction
24 component will dictate the steering direction for the
25 velocity servo.

1 Referring to FIGS. 1 and 5, power supply 30 supplies
2 the necessary electrical power to power assist controller
3 14, user interface 12, and velocity servos 16.

4 Electrical power is supplied from a power source such as
5 a battery bank, or an engine driven alternator in
6 conjunction with a voltage regulator and battery. If
7 only a battery bank is used as the electrical power
8 source, then it is preferable that a rechargeable battery
9 with high discharge capacity at high current draw be
10 used. Typical battery technologies that meet these
11 requirements include wet cell lead acid or Nickel Metal
12 Hydride batteries. The battery capacity can be chosen to
13 match the desired operating time of the device assuming
14 some nominal operating duty cycle. When the battery
15 capacity is discharged to the point the device is no
16 longer operational, the battery can be swapped for a
17 fully charged battery, or charged in place.

18 In devices wherein a fueled prime mover (fueled
19 engine) is included in the system, for example in a lawn
20 mower, the prime mover may be used to provide the
21 electrical power. Referring to FIG. 11, the fueled prime
22 mover 400 provides mechanical power to an alternator or
23 generator 401 via a rotating shaft 400A. The alternator
24 or generator 401 can be integrated with the prime mover
25 or mounted to or in close proximity to the prime mover
26 400. Typical small horsepower fueled engines incorporate

1 magnets that are embedded in the prime mover's flywheel,
2 rotating in close proximity to alternator coils attached
3 to the housing around the flywheel. For larger
4 horsepower prime movers, the alternator or generator is
5 usually mounted to the engine, and driven from the engine
6 rotating shaft via pulley and belt arrangement.

7 Alternator or generator 401 provides alternating or DC
8 voltage to voltage regulator 402. The voltage regulator
9 402 may incorporate a solid state rectifier (diode
10 rectifier) to convert the alternating voltage output of
11 the alternator to a DC voltage. Voltage regulator 402
12 provides regulated output voltage to battery 403 in the
13 presence of a changing electrical load on battery 403.
14 Standard regulation techniques such as shunt or series
15 regulation are employed to provide the voltage regulation
16 function. The battery voltage is controlled to maintain
17 safe operating voltages on battery 403, as well as to
18 provide optimum voltage levels for battery charging.

19 Referring again to FIG. 11, battery 403 provides an
20 electrical buffer between the load and the prime mover
21 driven alternator or generator 401. If alternator or
22 generator 401 is sized to meet or slightly exceed the
23 average electrical load conditions of power assist device
24 10, then it should be possible to maintain the battery in
25 a fully charged state. During periods when the
26 electrical demand of the load exceeds the power level of

1 the alternator or generator output, the battery charge is
2 diminished. During periods when the electrical demand of
3 the load is less than the power level of the alternator
4 or generator output, the battery charge is replenished.

5 Power Assist Supervisory Control

6 For many classes of power assisted devices, the
7 power assist algorithm can be enabled at system power up,
8 and remain enabled until shut down by the operator, or by
9 device specific automatic safety shutdowns (e.g.,
10 hardware faults). Regulatory requirements or prudent
11 design for safe operation may dictate that certain
12 classes of devices incorporate some form of "dead-man"
13 switch to activate and deactivate the power assist
14 mechanism.

15 There are classes of power assist devices that do
16 not fit into either of these two categories, and that
17 require some form of "supervisory" control for the
18 automatic activation and deactivation of the power assist
19 mechanism. These classes of power assist devices include
20 devices that exhibit intrinsically unstable operation
21 around one or more points of operation. A typical
22 example is a loaded, two-wheeled hand truck, which
23 exhibits unstable operation at all pitch angles other
24 than the vertical angle and the balance point angle. In
25 accordance with another embodiment of the invention, the
26 control algorithm implemented by power assist controller

1 14 (see FIG. 8) is modified to assist in the safe
2 initiation and termination of the power assist operation
3 of intrinsically unstable devices such as those described
4 in the foregoing description. In the case of a two-
5 wheeled hand truck, one scenario is to prevent power
6 assisted operation until the operator has rotated the
7 hand truck to a comfortable operating angle, and to
8 terminate power assisted operation when the pitch angle
9 of the hand truck gets too close to vertical. The
10 following describes one specific supervisory control
11 methodology that addresses this class of devices. In
12 order to facilitate understanding of this methodology,
13 the algorithm implemented by power assist controller 14
14 is configured for use with the hand truck model.
15 However, it is to be understood that the methodology is
16 applicable and extendable to other configurations
17 exhibiting different stability characteristics.
18 Referring to FIG. 12A, there is shown a flowchart of the
19 supervisory control methodology for the automatic
20 activation and deactivation of power assisted operation.
21 The first step 501 is the power-up step with the hand
22 truck being in a vertical position. In steps 502 and
23 503, the supervisory control will establish zero velocity
24 reference for each wheel velocity servo, and enable each
25 wheel velocity servo. Additionally, the power assist
26 algorithm will not respond to force inputs from the

1 handle sensor. The net effect is to maintain zero wheel
2 speed via the velocity servos. As the operator pulls
3 back on the handle of a loaded hand truck, the velocity
4 servos will maintain zero wheel speed and prevent any
5 rolling motion of the hand truck. Thus, the applied
6 force to the handle will result in a pitch moment about
7 an axis through the wheel axles resulting in the rotation
8 of the hand truck and load. As the hand truck pitch
9 angle changes, step 504 continually monitors the pitch
10 angle, and when the angle exceeds some fixed value p_0 ,
11 step 505 implements the dynamic model 205 (see FIG. 5)
12 which will generate the velocity references from the
13 applied forces at user interface 12. Step 506 determines
14 when the pitch angle exceeds a predetermined fixed value
15 p_1 . When the pitch angle is less than p_1 , the power assist
16 dynamic model is deactivated and step 503 ramps the
17 velocity references to zero. As shown in FIG. 12B, angle
18 p_1 will be very close to the vertical so as to allow the
19 power assist to remain active for a wide range of pitch
20 angles, yet provide a controlled shutdown of the assisted
21 operation when the user returns the hand truck to the
22 vertical position for unloading or loading.

23 The supervisory control methodology described will
24 further benefit the user by allowing them to rotate the
25 hand truck to and from the vertical position without the
26 need to place the foot on the wheel axle, as is customary

1 practice with non-power assisted hand trucks, in order to
2 retain the bottom of the unit from rotating.

3 Thus, in accordance with a further embodiment, the
4 present invention is applied to two-wheeled devices such
5 as two-wheeled carts and hand-trucks. A two-wheeled
6 vehicle such as a hand-truck in a static (motionless)
7 state is unstable (i.e. it will fall over) except for
8 three conditions:

9 a) the vehicle is lying flat; or

10 b) the vehicle is completely upright and resting on its
11 nose plate; or

12 c) the vehicle is tilted at a specific angle called the
13 "balance point", an angle of tilt where the net
14 gravitational forces create zero net moment about
15 the hand-truck axle.

16 When the hand-truck is tilted at angles closer to the
17 operator than the balance point, the hand-truck will tend
18 to fall toward the operator thereby approaching condition
19 (a) above. When the hand-truck is tilted beyond the
20 balance point, the hand-truck will move toward the
21 upright position, thereby approaching condition (b). The
22 balance point, while stable, represents a point with
23 negative static stability. Any perturbation from the
24 balance point will result in movement away from the
25 balance point.

1 These problems result from the fact that the
2 operator must apply forces to the handle of the two-
3 wheeled vehicle in order to balance the vehicle. These
4 forces are in addition to the forces the operator must
5 apply in order to make the vehicle move or change speed,
6 including starting, maintaining or changing speeds, and
7 slowing to a stop. In accordance with the invention, the
8 following method is used to solve the aforementioned
9 problems related to two-wheeled vehicles. For purposes
10 of illustrating this embodiment, the two-wheel vehicle is
11 a hand-truck.

12 First, upon start-up, the operator must tilt the
13 hand-truck over a sufficient angular range so that
14 parameters of the combination of the hand-truck and load
15 can be automatically determined.

16 During operation, the forces applied by the operator
17 are transformed into an appropriate coordinate frame and
18 the tilt angle of the hand-truck is measured
19 continuously.

20 In the next step, the net forces required at the
21 current angle of tilt to balance the hand-truck and load
22 are computed using the model of the hand-truck and the
23 load.

24 Next, the forces required for balance are subtracted
25 from the total forces applied by the operator to produce
26 an estimate of the forces intended to propel the hand-

1 truck. This difference is then used to drive the power
2 assist algorithm described in the foregoing description
3 (see FIG. 8). As a result of this method, the operator
4 can remain stationary with the hand-truck at any tilt
5 angle and the hand-truck will remain stationary even
6 though a non-zero set of forces is applied to the
7 instrumented handle. These forces can be substantial
8 depending on the load. This method does not use the
9 control system to automatically balance the system. The
10 method actually mimics a manual system, but with lower
11 operator effort. Thus, although the method would require
12 the operator to balance the system as for a manual hand-
13 truck, the operator input forces required to propel or
14 stop the hand-truck are significantly reduced. Thus,
15 this method identifies a model of the combined hand-truck
16 and load, and then uses the model in real time to remove
17 the operator-input forces to balance the hand-truck from
18 the force estimates that are sent to the power assist
19 algorithm

20 In order to implement this embodiment of the
21 invention, a processing system, drive system and power
22 supply module are attached to the two-wheeled vehicle.
23 The processing system pre-processes the data signals
24 defining the forces needed to balance and drive the two-
25 wheeled vehicle and then implements the power assist
26 algorithm and method, as described in the foregoing

1 description and shown in FIG. 8, using the pre-processed
2 data.

3 Each step of the method relating to two-wheeled
4 vehicles described above is now described in detail.

5 a) Model Identification

6 Prior to implementing the power assist algorithm
7 described in the foregoing description with respect to a
8 hand-truck, the hand-truck operator must move the hand-
9 truck through a range of tilt angles so that the static
10 properties of the hand-truck and load can be determined.
11 This model can then be used to remove the forces required
12 to balance the hand-truck from the forces applied by the
13 operator for vehicle movement. FIGS. 13A and 13B show the
14 coordinate frame and force definitions used in the
15 calculations and processing. The coordinate frame is
16 centered on a point on the wheel below the axle and the X
17 axis points outward and the Z axis points down parallel
18 to the frame. Critical quantities are:

19 P, the angle of pitch from the vertical;
20 Θ , the complement of P;
21 $\Theta=90^\circ$ corresponds to straight up;
22 L_1 , the Z coordinate of the CG;
23 L_2 , the height of the hand truck;
24 L_3 , the X coordinate of the frame;
25 L_4 , the distance along the X axis from the frame
26 to the center of gravity (CG).

1 The total moment, which varies with the pitch angle, is
2 described by equation 10:

$$3 \quad M = WL_1 \cos \theta - W(L_3 + L_4) \sin \theta \quad (10)$$

4 wherein "W" is the weight, and L_1 , L_3 , and L_4 describe the
5 offsets of the center-of-mass of the combined hand-truck
6 and load.

7 Startup begins with the hand-truck at the stable,
8 upright position. If this condition is not met, the
9 power assist algorithm and method will not be executed.
10 The operator must then tilt the hand-truck toward himself
11 well past the balance point (typically approximately
12 60°). During this motion, user interface 12 records the
13 pitch angle and the handle sensor forces. Referring to
14 FIGS. 13A and 13B, the coordinate system is placed at the
15 bottom of the wheel below the axle. The position of the
16 center of gravity, CG, is described by L_1 , L_3 , and L_4 . The
17 angle θ describes the angle of tilt and has a value of
18 90° when the hand-truck is upright and is zero when the
19 hand-truck frame is parallel to the floor. These
20 measurements are then used to create the model for the
21 torque required to balance the hand-truck. In the
22 simplest implementation, we approximate equation (10)
23 above by equation (11):

24

$$25 \quad M = \alpha \theta + \beta \quad (11)$$

1
2 Given measurements of M and θ over a sufficient range
3 (over approximately 60° , including the balance point),
4 the coefficients α and β can be identified by linear
5 least squares. The resulting linear relationship between
6 pitch angle and moment represents an approximation to the
7 actual behavior. We can also identify the groups of
8 terms in equation (10) by linear least squares to provide
9 a more precise model of the moment as a function of θ .
10 We can rewrite equation (10) as equation (12):

11

$$12 \quad M = [-W(L_3 + L_4) \quad WL_1] \begin{bmatrix} \sin \theta \\ \cos \theta \end{bmatrix} = [\delta \quad \varepsilon] \begin{bmatrix} \sin \theta \\ \cos \theta \end{bmatrix} \quad (12)$$

13

14 While this equation is nonlinear, it is linear in the
15 parameters δ and ε , which can be computed by linear least
16 squares. The additional computational burden over
17 fitting the linear model is very minor. Equation (12)
18 also shows an important aspect of this method.
19 Specifically, only the grouped terms δ and ε are
20 identified, not the actual lengths and weight. Thus, the
21 weight of the load and the location of its center of
22 gravity are not known. However, at this step in the
23 method, it is known that the center of gravity of the
24 load lies somewhere along a line passing through both the

1 balance point and the origin. The angle of this line is
2 defined by equation (13):

$$3 \qquad \theta_b = \tan^{-1} \left(\frac{\varepsilon}{\delta} \right) \qquad (13)$$

4
5 However, the weight of the load could be large with its
6 center of gravity at a small distance from the origin, or
7 it could be a relatively lighter weight with its center
8 of gravity farther away from the origin. Statically,
9 these are the same if the product of the distance from
10 the origin and the weight is equal. However, dynamically
11 they are different. Different loads will result in
12 moment-pitch angle relationships with different balance
13 points and different shaped curves. FIG. 14 shows three
14 such curves for identical load weights with different
15 locations for the center of gravity (CG). The weight of
16 the load is 50 kg. In each case, the center of gravity
17 (CG) is 0.4 meters forward from the origin (L_3+L_4) but the
18 vertical distances (L_1) of the center of gravity (CG) are
19 0.0 meters (low), 0.5 meters (medium) and 1.0 meters
20 (high). The curves shown in FIG. 14 are not linear and
21 intersect 0.0 moment (the balance point) at greatly
22 different angles. These functions resemble straight
23 lines about the balance point, indicated by (*).
24 Accordingly, the moment required to balance the hand-

1 truck at a reasonable angle (for example 45 degrees)
2 varies in both magnitude and sign.

3 Referring to FIG. 15, there is shown a plot that
4 shows experimental data for the moment, as measured at
5 the instrumented handle as a function of pitch angle for
6 three different loads. Each load has different weight
7 and center of gravity, with the balance point (moment=0)
8 ranging from about 10° to about 90°. The linear and
9 nonlinear model fits are also shown in FIG. 15. The
10 start-up of the test is shown in the measured moment
11 curve on the left side. This start-up response is
12 ignored when the model parameters are computed. The
13 nonlinear model fits the data better and most
14 importantly, it predicts the balance point better in one
15 of the three conditions (a), (b) and (c) discussed in the
16 foregoing description.

17 b) Use of the Model in Real Time

18 In accordance with this embodiment of the invention,
19 the moment-angle relationship is used in real-time to
20 remove the component of the operator's input force that
21 balances the two-wheeled vehicle. FIG. 16 shows the
22 additional dimensions needed to specify the moment
23 supplied by the user, and FIG. 17 shows implementation of
24 the power assist method and algorithm for the linear
25 model case. FIG. 17 is a processing diagram that
26 illustrates how the moment required to balance the load

1 (M_{load}) is computed and then removed from the overall
2 measured moment inputted by the operator (M_{oper}). As
3 diagrammed in FIG. 17, the operator moment (M_{oper}) is
4 calculated from the X and Z components of the measured
5 sensor forces (F_{sx} and F_{sz} respectively) in the sensor
6 coordinate frame, and the associated moment arms X_0 and Z_0
7 from the handtruck origin (X, Z). The load moment (M_{Load})
8 illustrated in FIG. 17 uses the linear model derived from
9 the Model Identification phase described in section a)
10 above, specifically equation 11. The model parameters, α
11 and β are derived during the last "power assist" turn-on
12 cycle, and are used to generate the estimated static
13 balance moment (M_{Load}) for the load as a function of pitch
14 angle θ . The net operator propulsion force is then
15 calculated as the difference between the balance moment
16 for the load (M_{Load}) and the measured operator generated
17 moment (M_{oper}). Without loss of generality, the load
18 moment could be computed using the non-linear model
19 derived from the Model Identification phase described in
20 section a) above, specifically equation 12.

21 The principles, preferred embodiments and modes of
22 operation of the present invention have been described in
23 the foregoing specification. The invention which is
24 intended to be protected herein should not, however, be
25 construed as limited to the particular forms disclosed,

1 as these are to be regarded as illustrative and exemplary
2 rather than restrictive. Variations and changes may be
3 made by those skilled in the art without departing from
4 the spirit of the invention. Accordingly, the foregoing
5 detailed description should be considered exemplary in
6 nature and not as limiting the scope and spirit of the
7 invention as set forth in the attached claims.

8 What is claimed is:

9
10
11
12
13

CLAIMS

1

2

3 1. A power assist device to assist a user in
4 effecting movement of a wheeled device, comprising:

5

6 a user interface configured to be grasped by a user and
7 measure up to six degrees of freedom forces and
8 torque applied to said user interface by the user
9 and convert said measured force and torque into
10 electronic signals;

11

12 a controller having a processing resource to process
13 said electronic signals, said controller providing
14 at least one velocity reference signal representing
15 a desired velocity;

16

17 at least one velocity servo having an input for
18 receiving said at least one velocity reference
19 signal, said at least one velocity servo outputting
20 a velocity feedback signal representing an actual
21 velocity; and

22

23 said processing resource of said controller being
24 configured to process said electronic signals
25 outputted by said user interface and said velocity

1 feedback signal to provide said at least one
2 velocity signal.

3

4 2. The power assist device according to claim 1
5 wherein the power assist device includes comprises a
6 wheeled device.

7

8 3. The power assist device according to claim 1
9 further including a power supply to power said user
10 interface, said controller and said at least one
11 velocity servo.

12

13 4. The power assist device according to claim 1
14 wherein said at least one velocity servo comprises
15 means to minimize error between said velocity
16 reference signal and said velocity feedback signal.

17

18 5. The power assist device according to claim 1
19 wherein said user interface comprises a sensor
20 handle.

21

22 6. The power assist device according to claim 5
23 wherein said sensor handle is configured to measure
24 three orthogonal forces at said sensor handle.

25

1 7. The power assist device according to claim 6
2 wherein said sensor handle comprises flexure device
3 means to generate electrical signals representing
4 applied force and torque.

5

6 8. The power assist device according to claim 2
7 wherein said processing resource of said controller
8 is configured to implement a three-dimensional, six
9 degree-of-freedom mathematical model to represent
10 the desired dynamics of the wheeled device.

11

12 9. The power assist device according to claim 1
13 wherein said user interface is configured to measure
14 predetermined, initial, static balance parameters of
15 a two-wheeled vehicle.

16

17 10. The power assist device according to claim 9
18 wherein said processing resource of said controller
19 is configured to process said measured
20 predetermined, initial, static balance parameters to
21 determine the forces or torques required to balance
22 said two-wheeled vehicle.

23

24 11. The power assist device according to claim 10
25 wherein said processing resource subtracts said
26 forces or torques required to balance said two-

1 wheeled vehicle from the total forces or torques
2 applied to the user interface so as to provide
3 estimates of the user-supplied propulsion forces and
4 to prevent user- supplied balancing forces from
5 causing motion of the system.

6

7 12. A method for assisting a user in effecting
8 movement of a wheeled device comprising the steps
9 of:

10

11 providing a wheeled device having a user interface
12 configured to be grasped by a user and at least one
13 velocity servo for assisting in the movement of said
14 wheeled device, said user interface configured to
15 measure forces and torque applied thereto by a user;

16

17 measuring up to six degrees of freedom forces and
18 torque applied to said user interface;

19

20 converting said measured force and torque into
21 electronic signals;

22

23 processing said electronic signals to provide at least
24 one velocity reference signal representing a desired
25 velocity;

26

1 inputting said at least one velocity reference signal
2 into said at least one velocity servo;
3
4 generating a velocity feedback signal representative of
5 an actual velocity of the wheeled device; and
6
7 processing said velocity reference signal and said
8 velocity feedback signal to minimize error between
9 these signals and provide an updated velocity
10 reference signal.

11

12 13. The method according to claim 12 wherein said
13 step of measuring comprises generating electrical
14 signals representing applied force and torque.

15

16 14. The method according to claim 12 wherein said
17 step of processing said electronic signals to
18 provide at least one velocity reference signal
19 representing a desired velocity comprises processing
20 said electronic signals with a three-dimensional
21 mathematical, six degree-of-freedom model to
22 represent the desired dynamics of the wheeled
23 device.

24

25

26

1 15. The method according to claim 12 wherein said
2 step of measuring includes measuring predetermined,
3 initial static balance parameters of a two-wheeled
4 vehicle.

5
6 16. The method according to claim 15 wherein the
7 step of processing said electronic signals comprises
8 processing said measured predetermined, initial
9 static balance parameters to determine the forces
10 required to balance said two-wheeled vehicle.

11
12 17. The method according to claim 16 wherein said
13 step of processing said electronic signals further
14 comprises subtracting said forces required to
15 balance said two-wheeled vehicle from the total
16 forces applied to the user interface so as to
17 provide estimates of the user supplied propulsion
18 forces to propel the two-wheeled vehicle and prevent
19 user-supplied balancing forces from causing motion
20 of the two-wheeled vehicle.

21

22

1/17

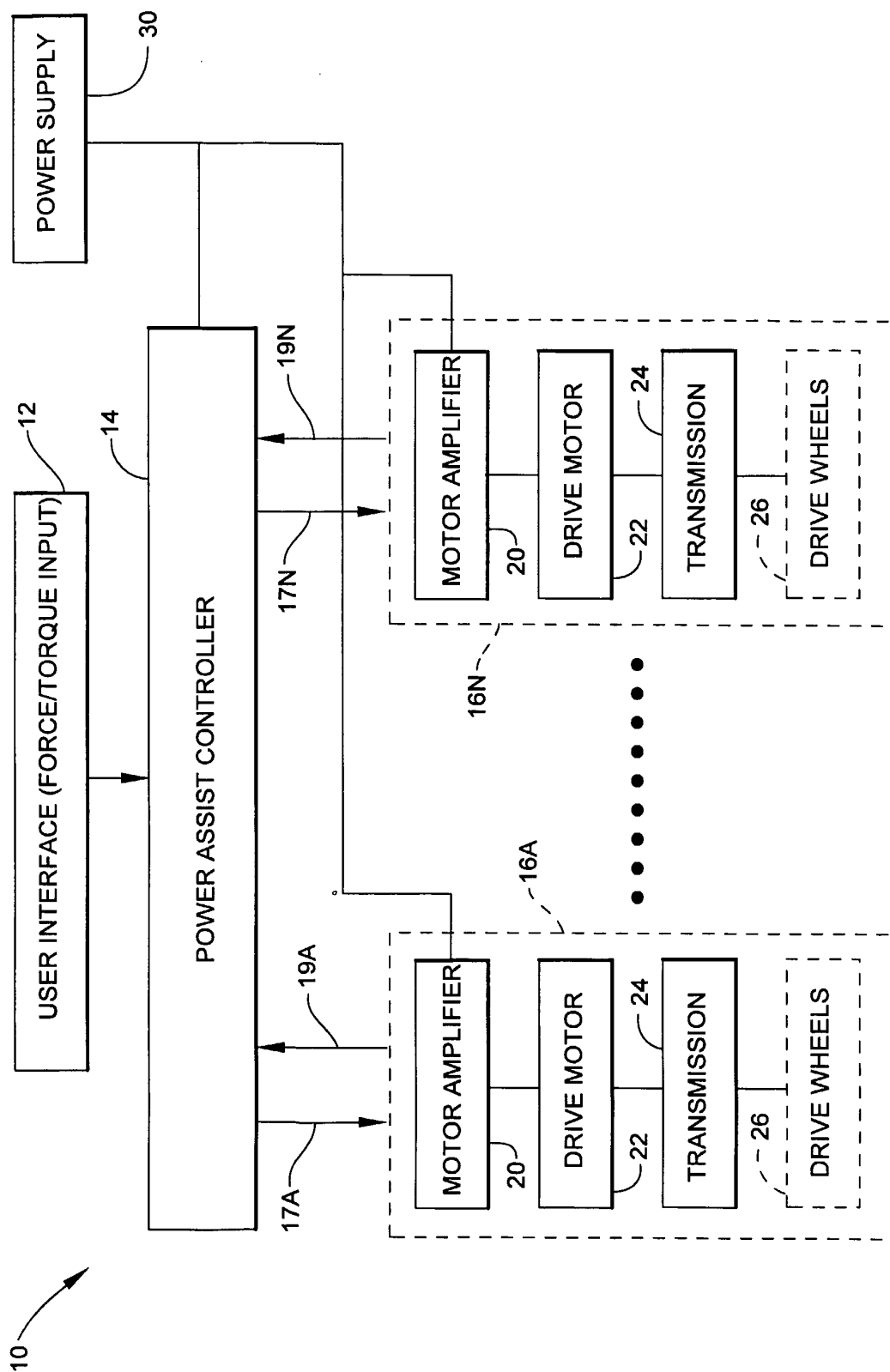


Fig. 1

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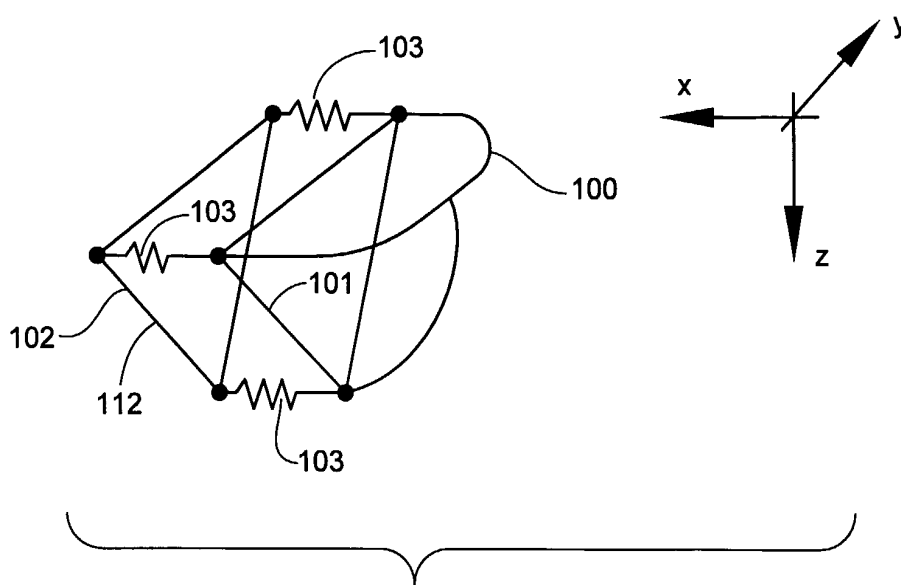


Fig. 2

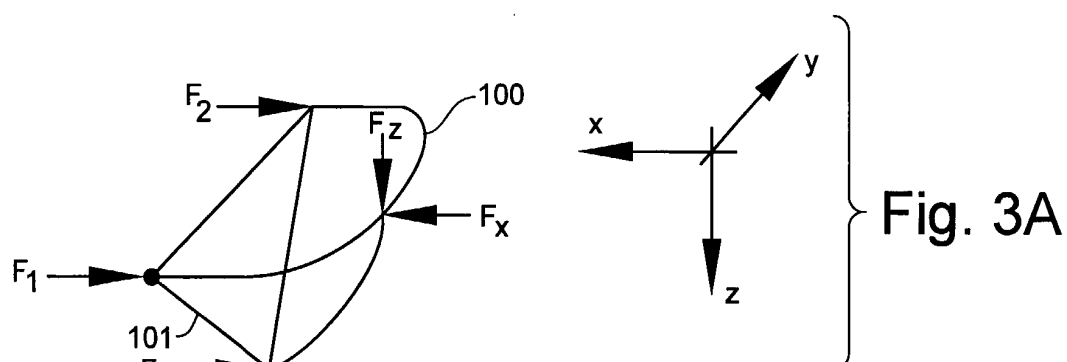


Fig. 3A

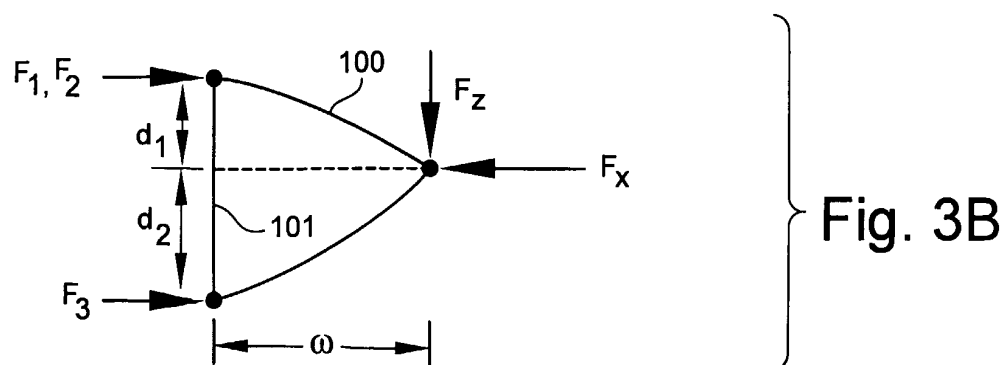


Fig. 3B

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Fig. 4A

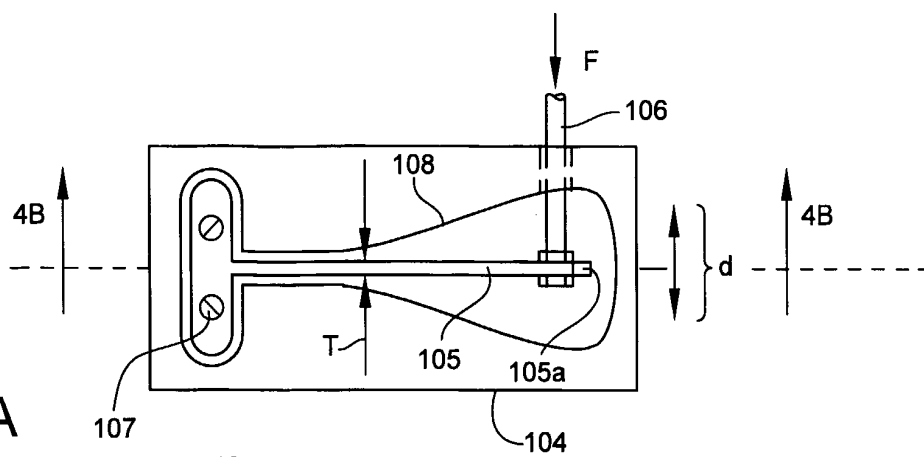


Fig. 4B

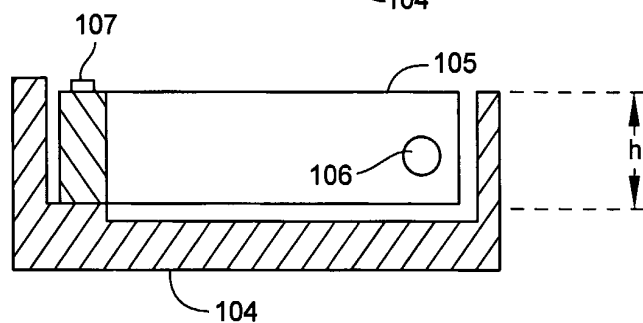
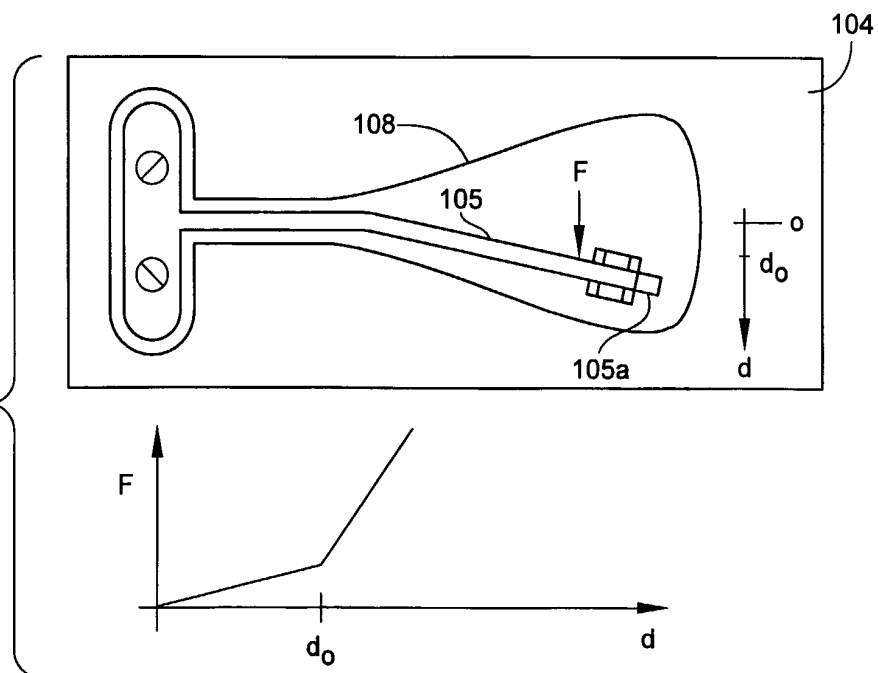


Fig. 4C



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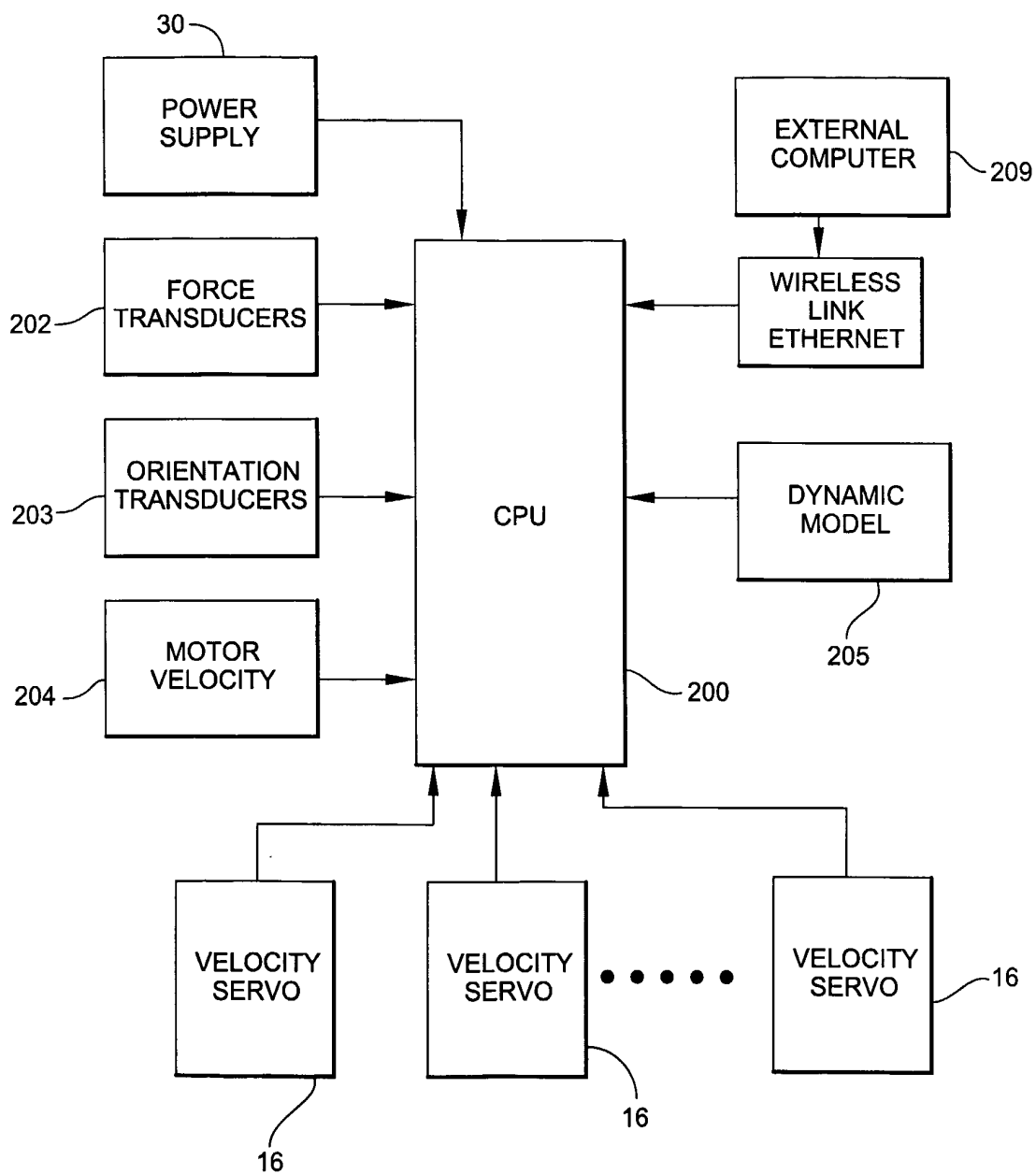


Fig. 5

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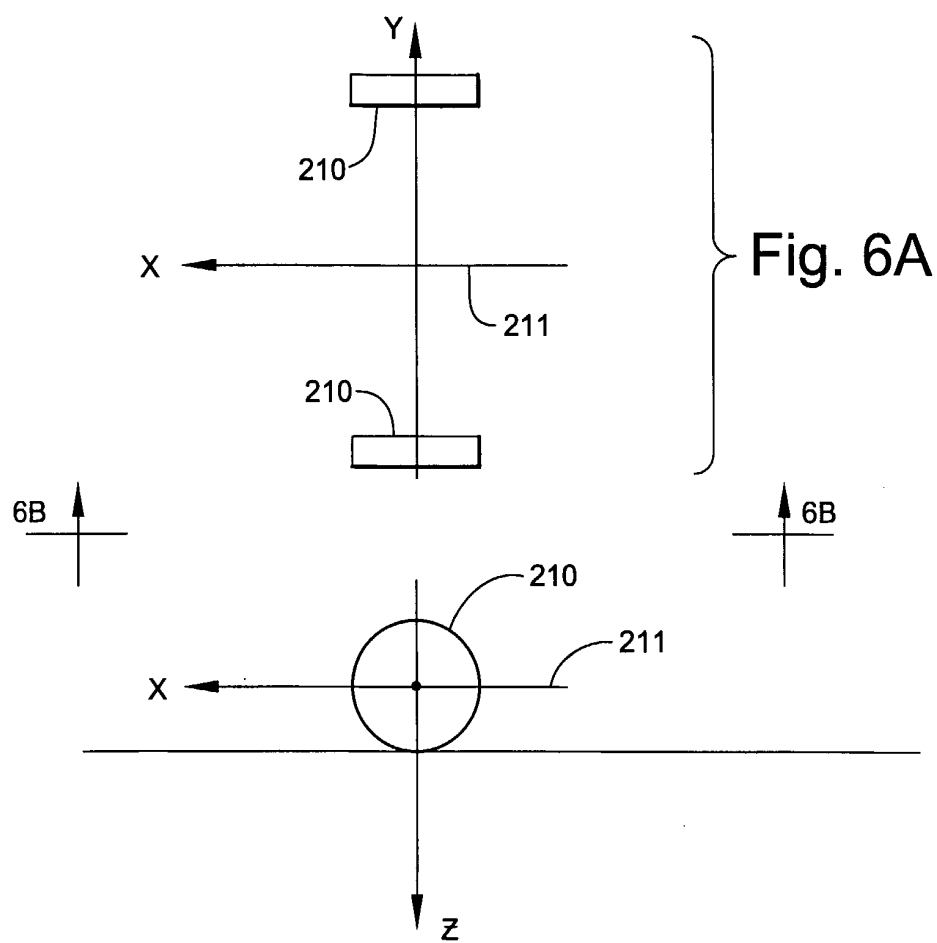


Fig. 6B

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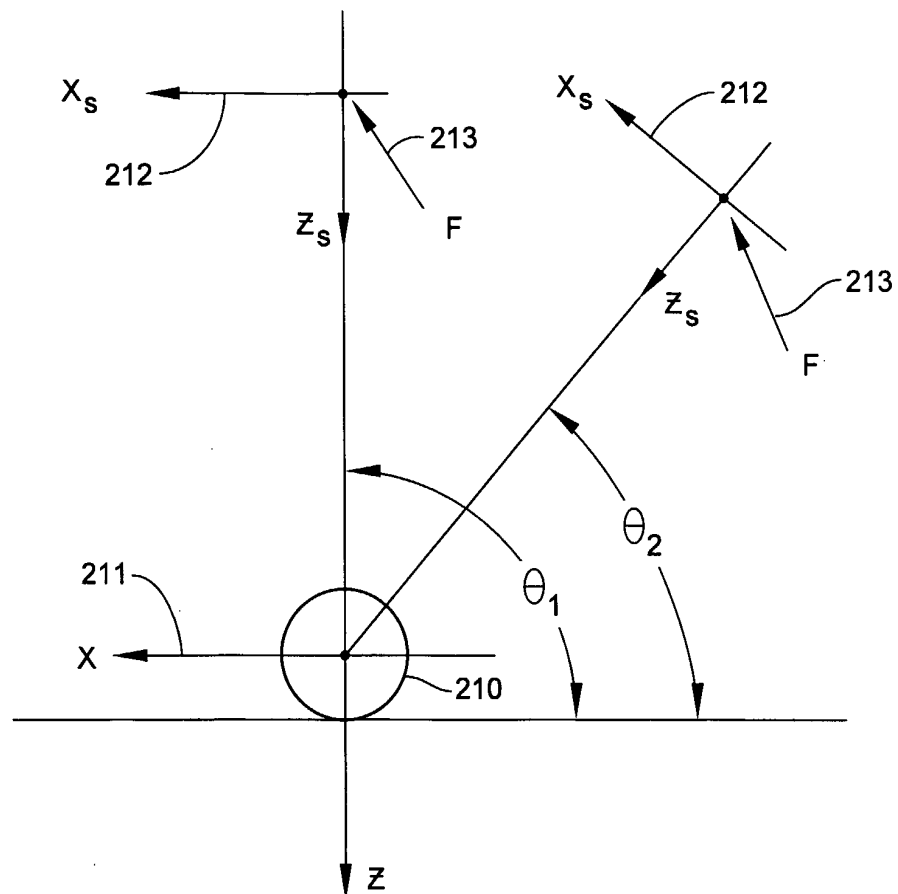


Fig. 7

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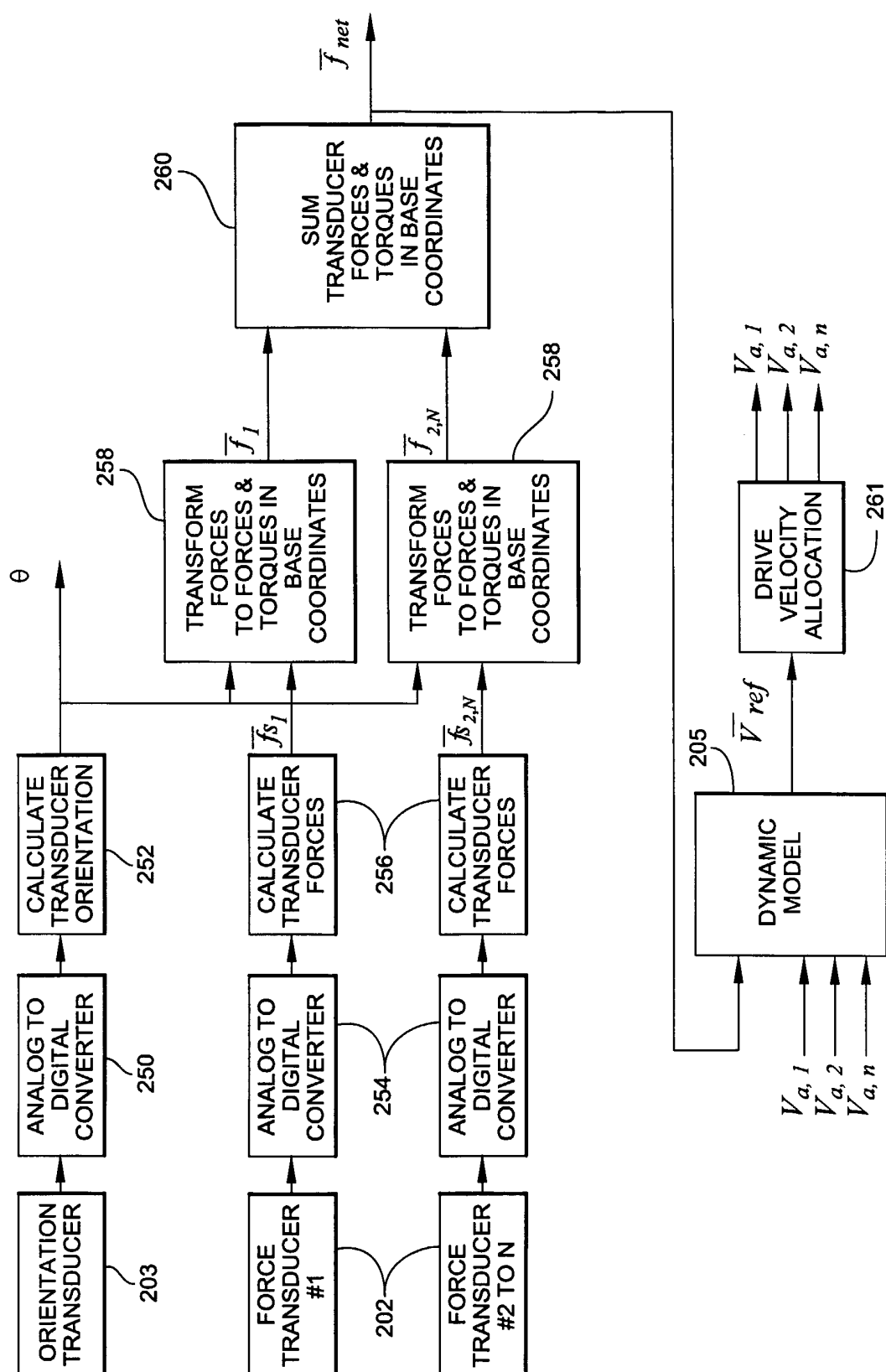


Fig. 8

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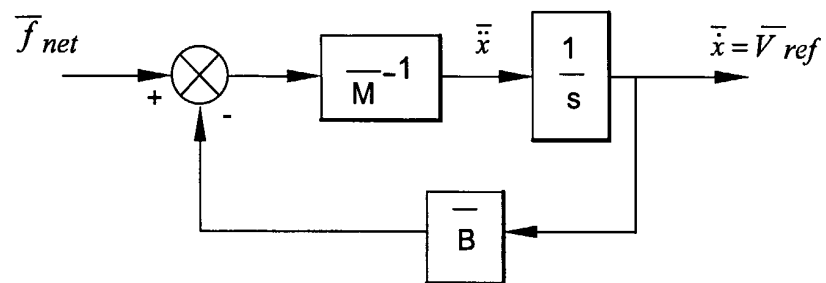


Fig. 9

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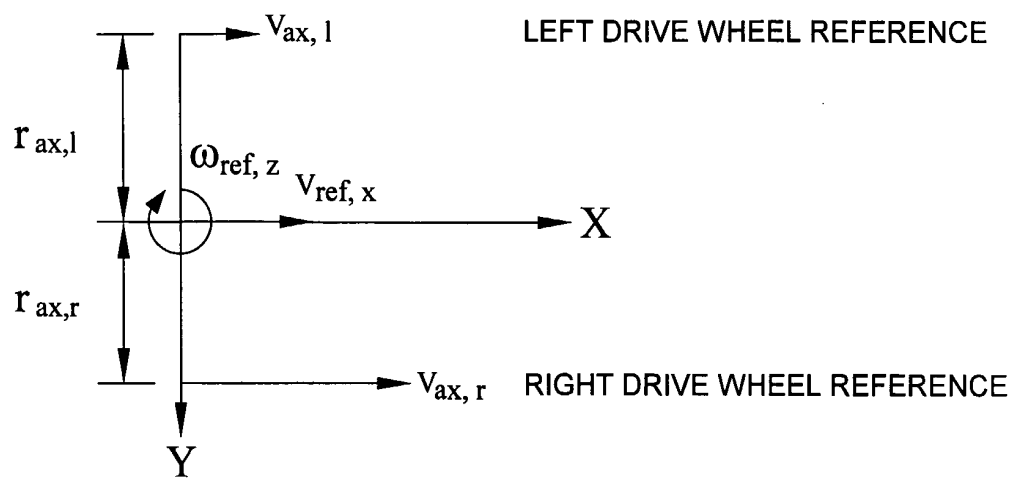


Fig. 10

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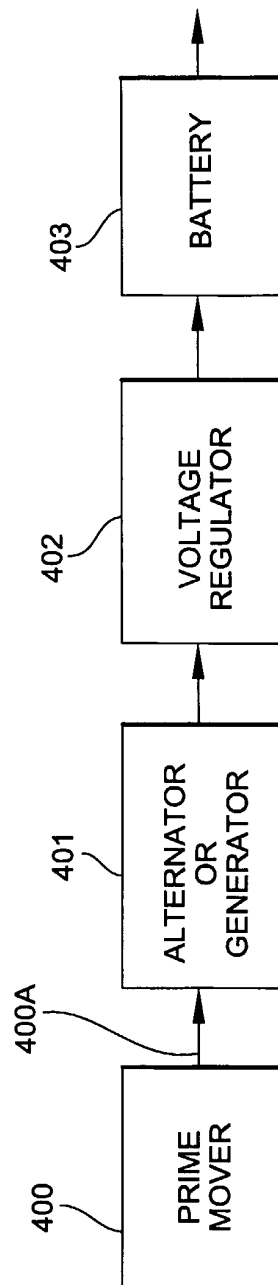
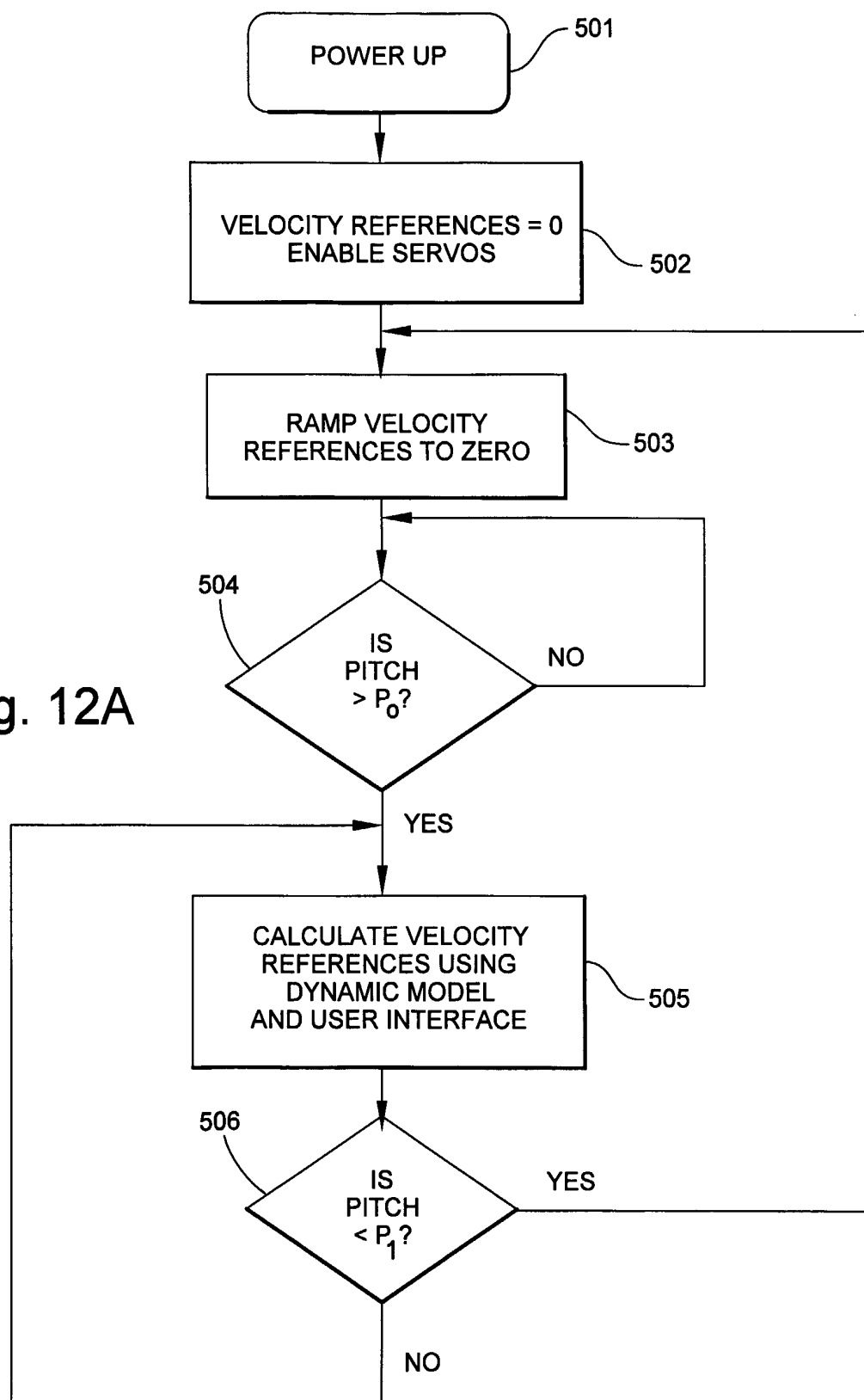


Fig. 11

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Fig. 12A



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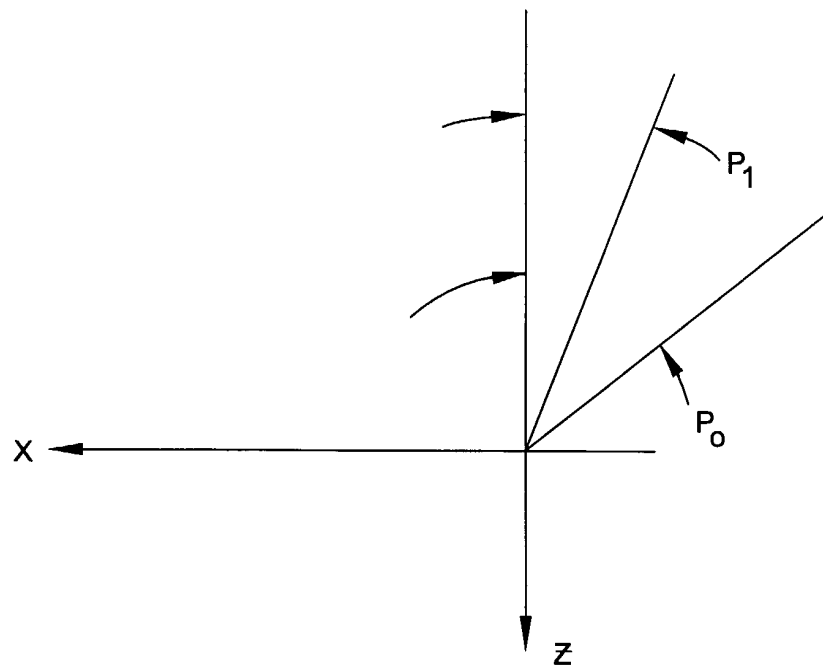


Fig. 12B

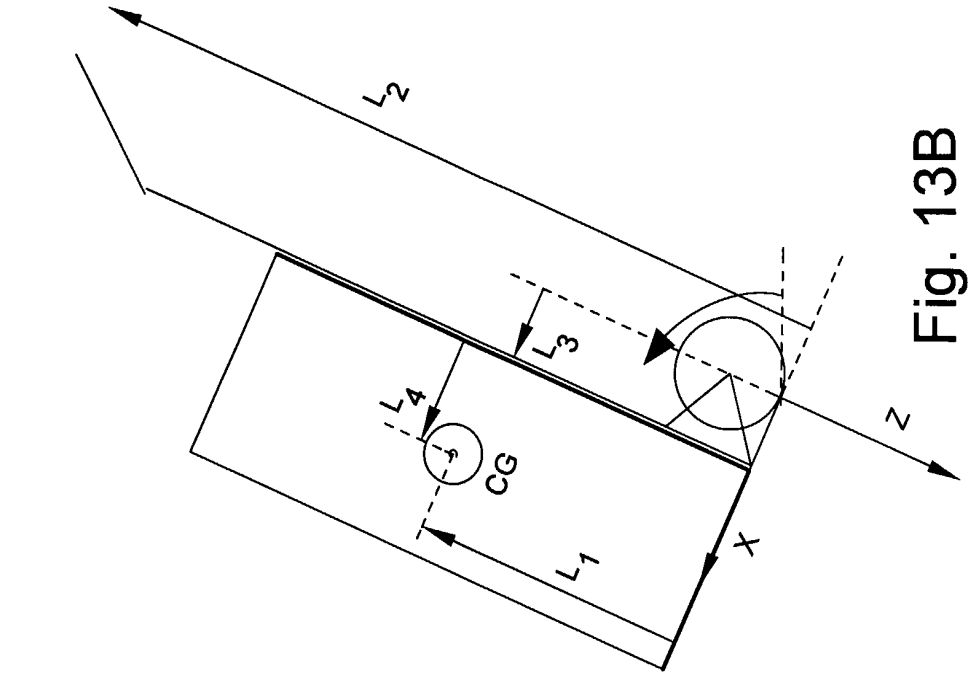


Fig. 13A

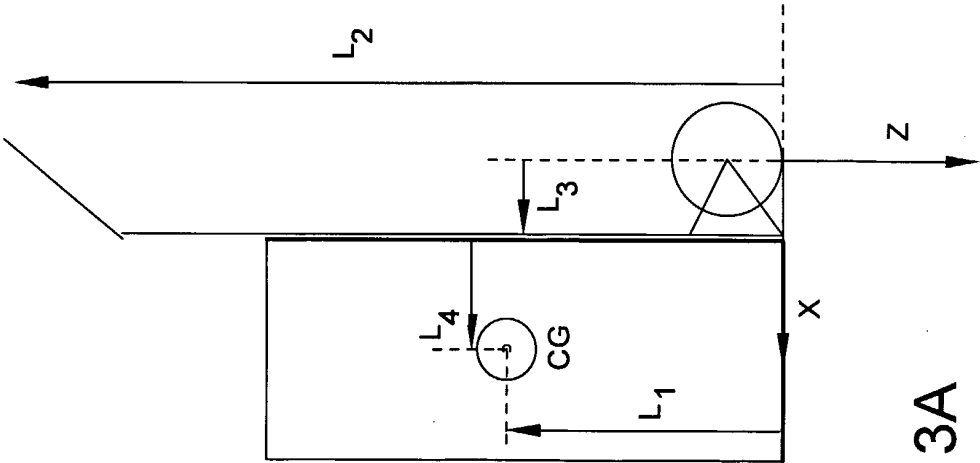


Fig. 13B

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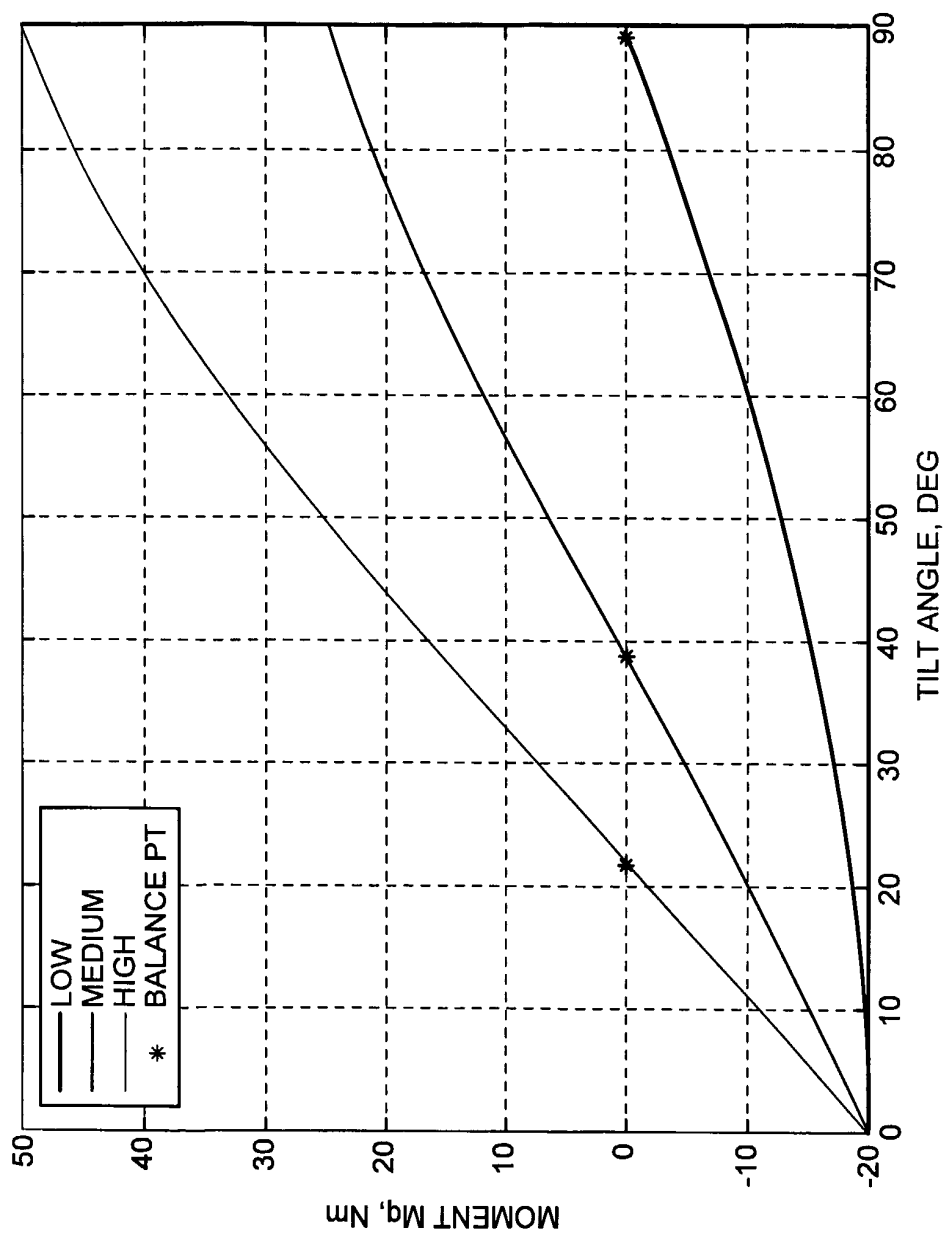


Fig. 14

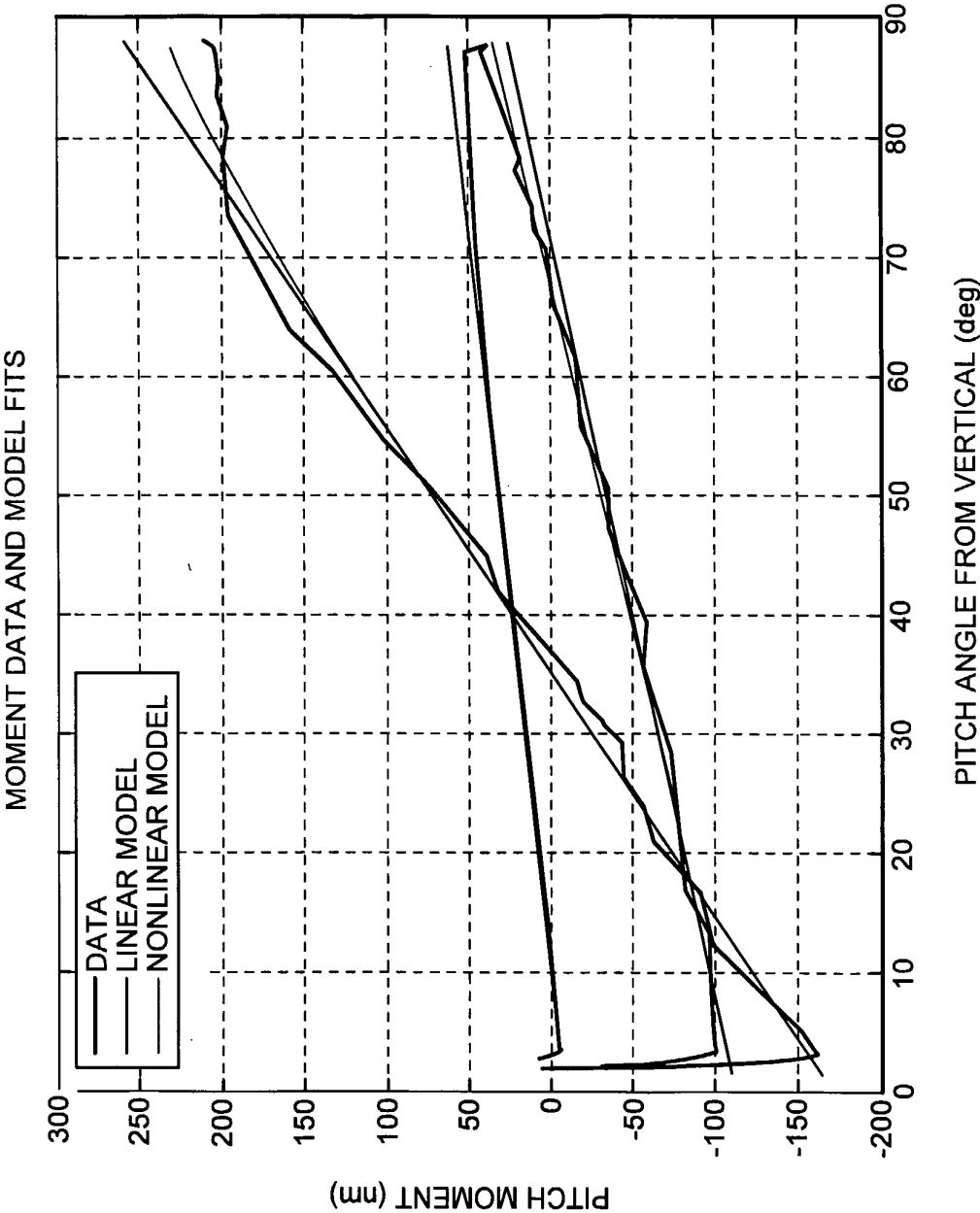


Fig. 15

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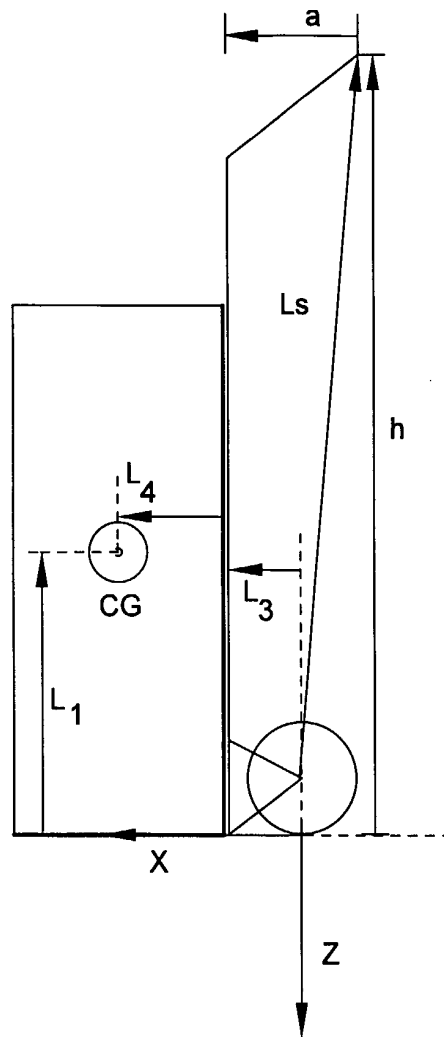


Fig. 16

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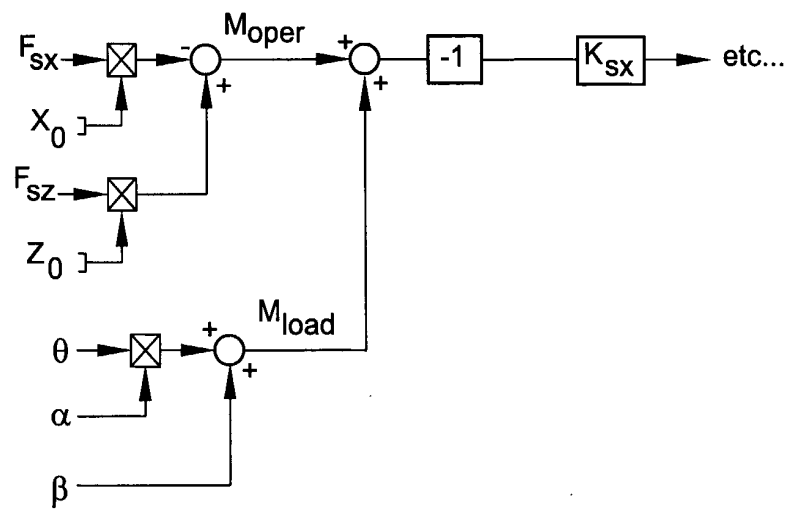


Fig. 17