United States Patent

Suzuki

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[54] **POWER STEERING APPARATUS**

[72] Inventor: **Akira Suzuki,** Kariya, Japan

[73] Assignee: **Toyoda Koki Kabushiki Kaisha trading as Toyoda Machine Works, Ltd.,** Kariya, Aichi, Prefecture, Japan

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[21] Appl. No.: **858,380**

[30] **Foreign Application Priority Data**

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[52] **U.S. Cl.**....................................**91/372,** 91/375, 92/136

[51] **Int. Cl.**..**F15b 9/10,** F01b 9/00

[58] **Field of Search**................91/375 A, 370, 371, 372, 373; 186/79.2; 60/52 S, 52 PJ

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Primary Examiner—Paul E. Maslousky

Attorney—Holcombe, Wetherill & Brisebois

[57] **ABSTRACT**

A power steering apparatus in which relative rotation between two steering shaft members connected by a resilient coupling member actuates a flap valve mechanism, the action of which controls the hydraulic pressure of a hydraulic motor to boost the power applied by manual steering, said apparatus comprising pistons which exert pressure against flap valves so as to damp the vibration of said valves, and a surge pressure absorbing apparatus in the hydraulic pressure circuit which absorbs the surge pressure generated in said circuit.

6 Claims, 13 Drawing Figures

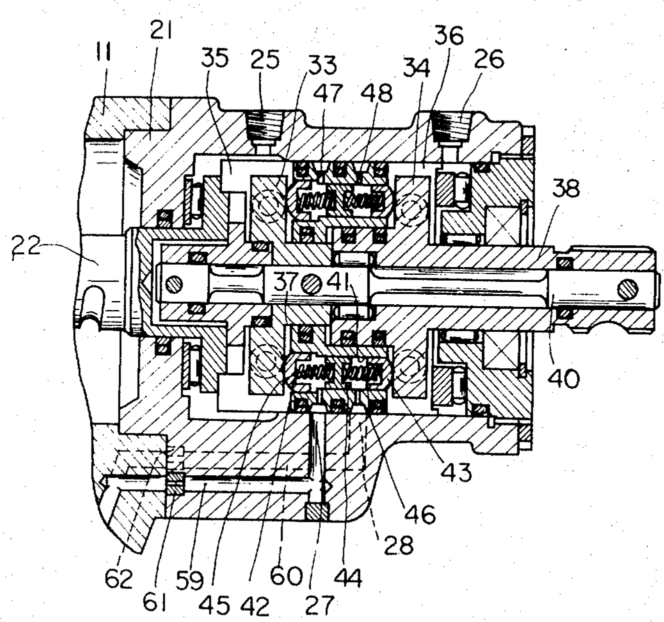

FIG. 1

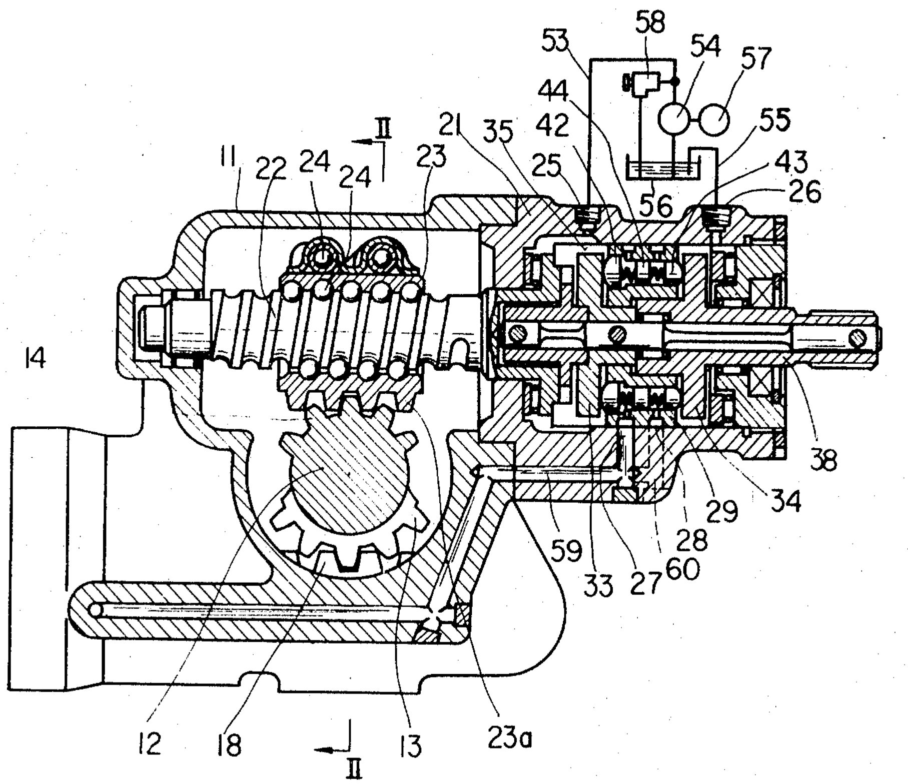
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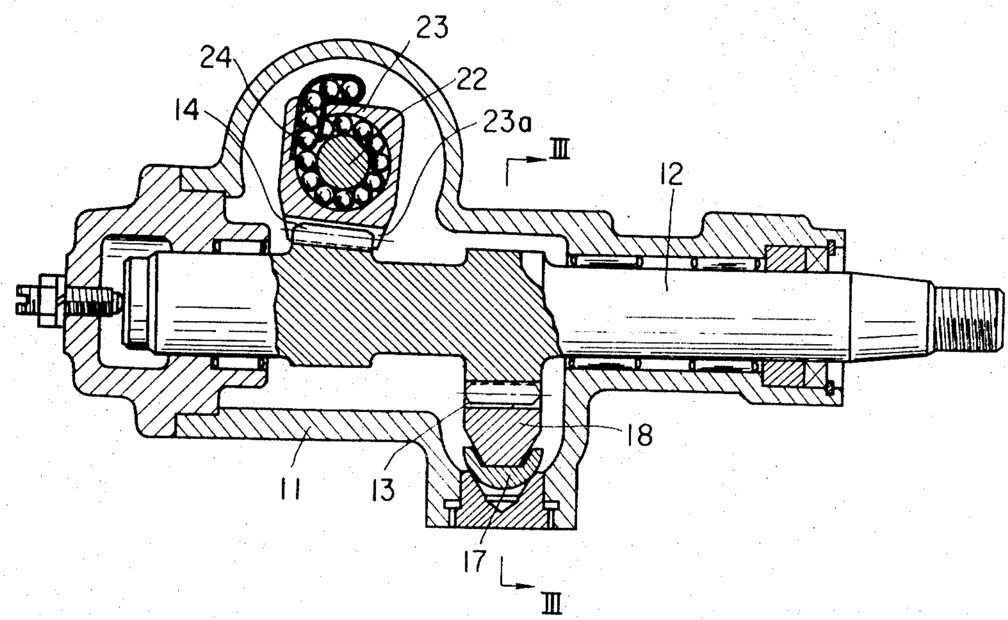
FIG. 2


23
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III
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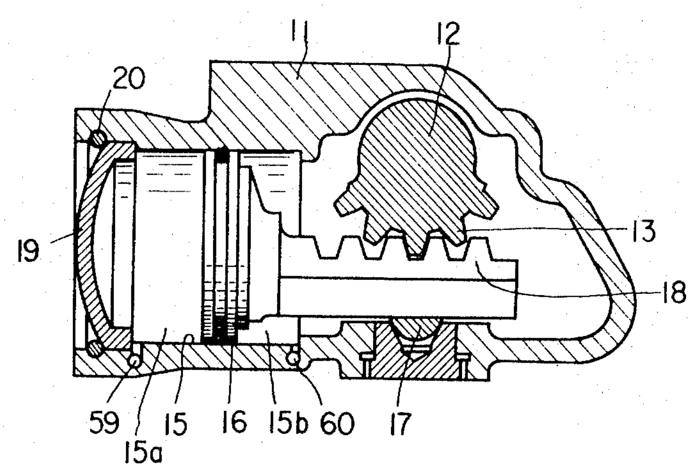
III

FIG. 3


11
12
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15a
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16
15b
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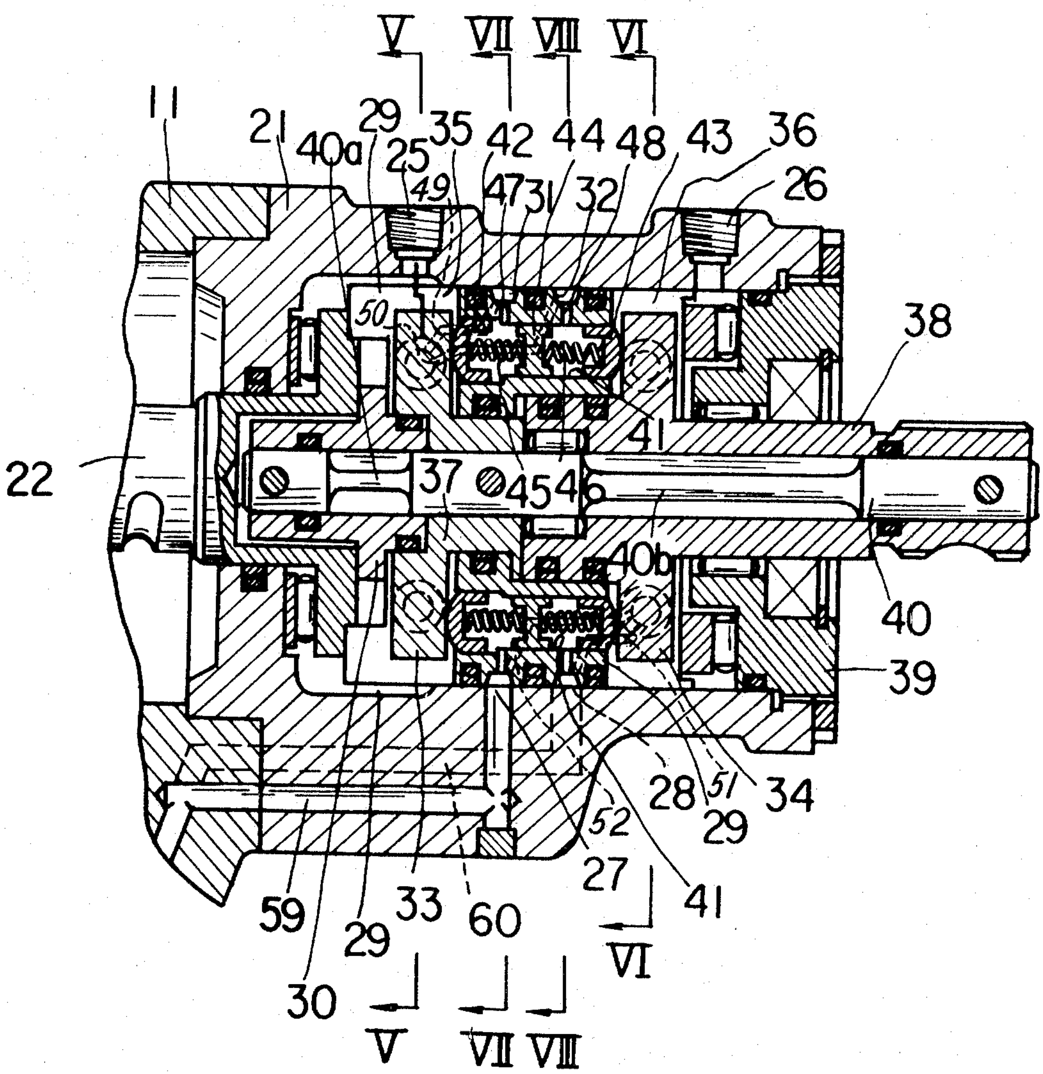
FIG. 4


V
VII
VIII
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40a
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50
38
22
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46
41
40b
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VIII

FIG. 5

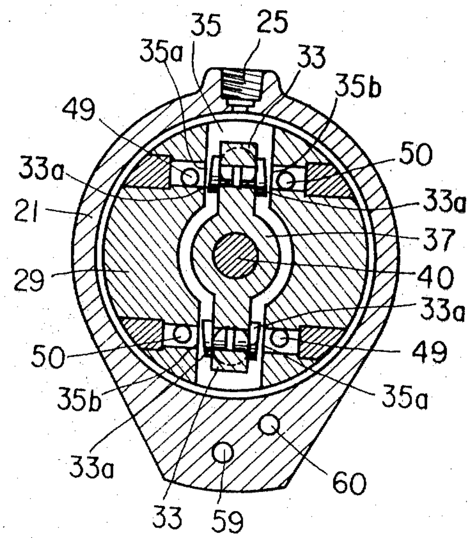
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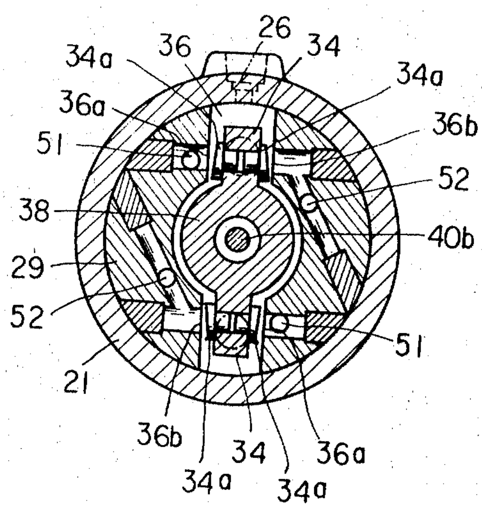
FIG. 6


26
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34a
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40b
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FIG. 7

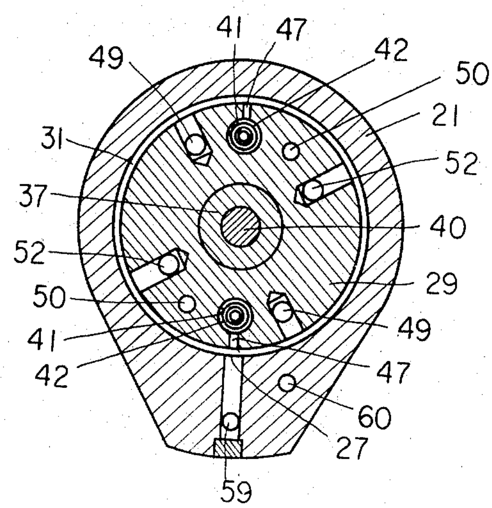
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FIG. 8

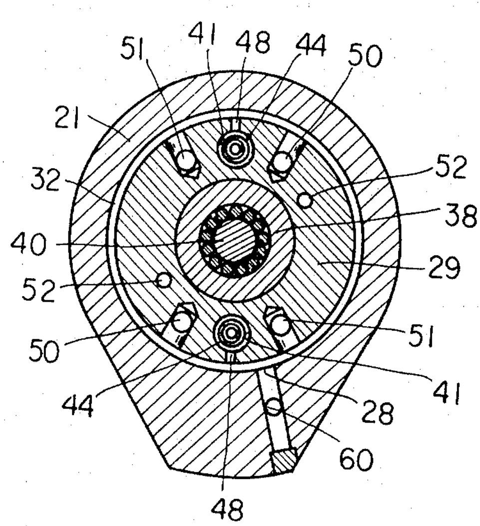
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FIG. 9

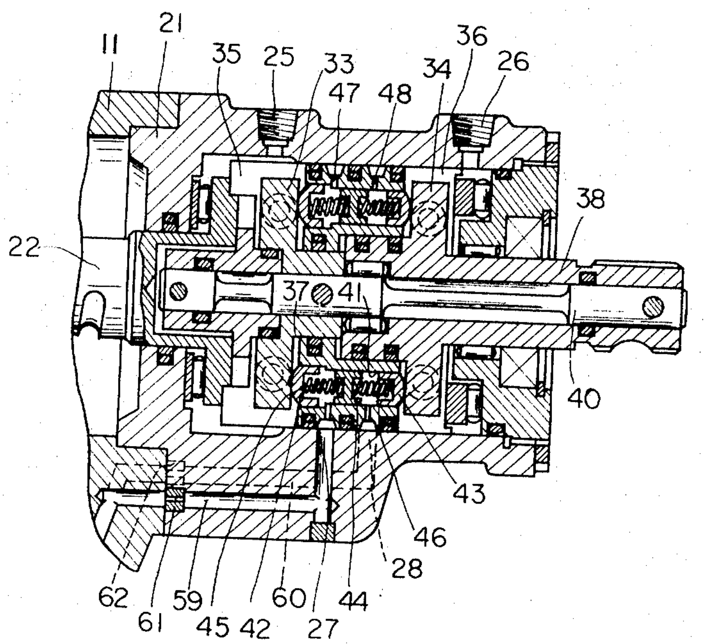
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FIG. 10

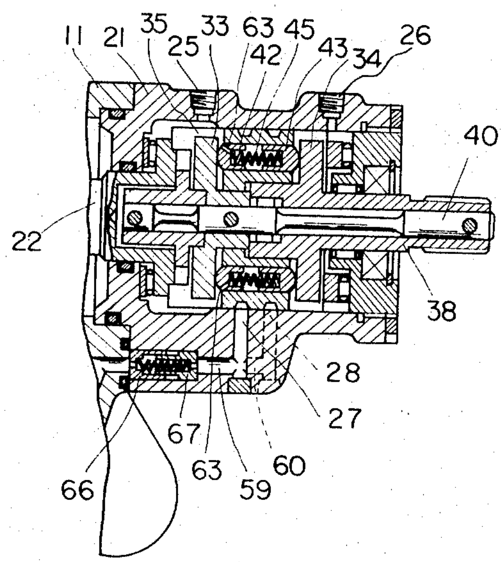

FIG. 11

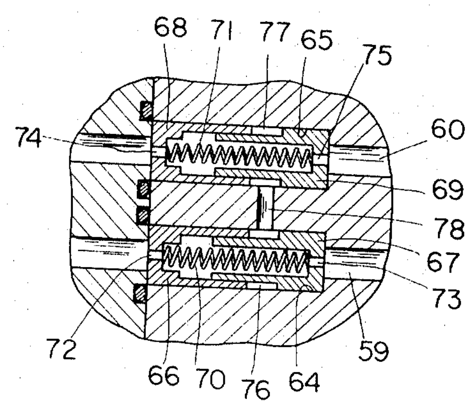
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FIG. 12

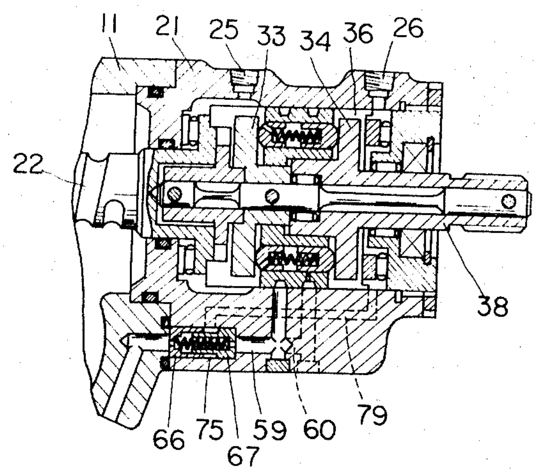
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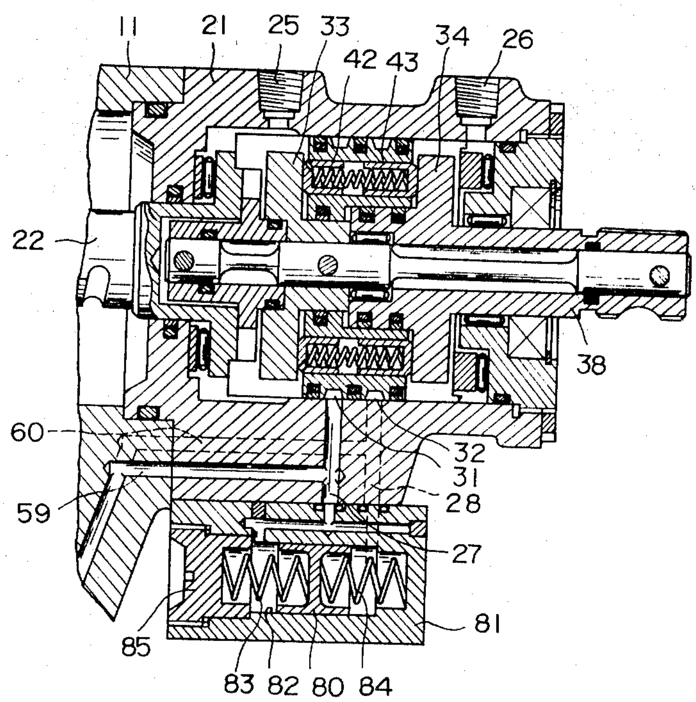
FIG. 13



POWER STEERING APPARATUS

SUMMARY OF THE INVENTION

The present invention relates to power steering apparatus, and more particularly to power steering apparatus comprising a hydraulic motor to boost the power applied by manual steering, said apparatus comprising servo valve means wherein flap valves are angularly moved by manual steering to control the pressure fluid for operating the hydraulic motor.

In power steering apparatus in which the steerable wheels are deflected by controlling the hydraulic pressure in a hydraulic circuit through the action of flap valve mechanism due to relative rotation between steering shaft members under the manual steering torque, the slightest rotational displacement of the flap valves can quickly and precisely deflect the steerable wheels. However, in such power steering apparatus, the flap valve mechanism is so sensitive that when a strong turning force acts on steerable wheels, the flap valves begin to vibrate and in consequence the stability of the valve may be impaired, an unpleasant noise may occur, and the steering operation may become unsatisfactory.

Moreover, when the flap valves vibrate or when the steering wheel is turned with a jerk, the gap between the flap valves and the main valve member will change sharply so that an abnormal surge pressure will develop in the hydraulic pressure circuit, and the steering operation of a driver who feels this surge pressure will be adversely affected.

In the present invention, which is designed to eliminate this conventional drawback, a very simple device attached to the flap valve mechanism can absorb the surge pressure generated in the hydraulic pressure circuit as well as effectively damp the vibration of the flap valves.

The present invention stabilizes the valve characteristics of the flap valve mechanism and enhances the steering performance.

The primary object of the present invention is to provide power steering apparatus comprising means for damping the vibration of flap valves.

Another object of the present invention is to provide power steering apparatus in which any surge pressure generated in the hydraulic pressure circuit is absorbed.

Still another object is to provide power steering apparatus which comprises a surge pressure absorbing device in the hydraulic pressure circuit which absorbs the surge pressure generated by a sudden action of the flap valves.

Yet another object is to provide power steering apparatus in which the vibration damping of flap valves and the absorption of surge pressure is attained by a simple mechanism.

The foregoing and other objects of the present invention, which will become more fully apparent from the following detailed description, may be achieved by means of the representative apparatus set forth in this specification and depicted in the accompanying drawings, in which:

FIG. 1 is a longitudinal section through a power steering apparatus constituting one embodiment of the present invention;

FIG. 2 is a transverse sectional view of the apparatus taken along the line II—II of FIG. 1;

FIG. 3 is a sectional view of the apparatus taken along the line III—III of FIG. 2;

FIG. 4 shows a partial enlargement of FIG. 1;

FIG. 5, FIG. 6, FIG. 7 and FIG. 8 are sectional views of the apparatus taken along the lines V—V, VI—VI, VII—VII, and VIII—VIII in FIG. 4, respectively;

FIG. 9 is a sectional view on an enlarged scale of the essential part of a power steering apparatus in another embodiment of the present invention;

FIG. 10 is a sectional view of the essential part of a power steering apparatus in still another embodiment of the present invention;

FIG. 11 is a sectional view showing part of the apparatus illustrated in FIG. 10 on an enlarged scale;

FIG. 12 is a sectional view of the essential part of a power steering apparatus in a fourth embodiment of the present invention; and

FIG. 13 is a sectional view of the essential part of a power steering apparatus in a fifth embodiment of the present invention.

In FIGS. 1, 2 and 3, reference numeral **11** indicates the main body of a power steering apparatus having a rotating shaft **12** rotatably supported therein. One end of the rotating shaft **12** is connected via a pitman arm and a connecting rod (not shown) and other connecting means to the steerable wheels of a vehicle.

Two axially spaced partial gears **13** and **14** are mounted on the rotating shaft **12**. These gears are concentric with the rotating shaft **12** and their teeth extend in opposite directions.

In the main body of the power steering apparatus is a hydraulic pressure cylinder **15**. A piston **16** (FIG. **3**) is mounted in said cylinder for sliding movement in a direction at right angles to the axis of the rotating shaft **12**. At one end of the piston **16** is a rack **18**, the bottom of which is supported by a bearing **17**.

The rack **18** meshes with the gear **13** on the rotating shaft **12**. As shown in FIG. **3**, the end of the cylinder **15** is closed by a cylinder cover **19** and a ring **20**.

A servo valve housing **21** is attached to the main body **11** of the power steering apparatus. Reference numeral **22** indicates a steering shaft, one end of which is supported in the housing **21** and the other end of which is supported in the main body **11**. The axis of the steering shaft **22** is parallel to the axis of the piston **16**. A nut **23** encircles ball bearings **24** which roll in a spiral groove in the steering shaft **22**, and rotation of the shaft **22** shifts the nut member **23** in an axial direction against small frictional resistance. The nut **23** is provided with a rack portion **23*a***, which meshes with the gear **14** on the rotating shaft **12**.

In FIG. **4**, the servo valve housing **21** is shown to be provided with a pressure fluid supply port **25** axially spaced from a discharge port **26**. Inside the servo valve housing are the supply-discharge ports **27** and **28** which communicate through conduits **59** and **60** respectively with a left compartment **15*a*** and a right compartment **15*b*** of the hydraulic pressure cylinder **15**. A main valve member **29** is rotatably mounted within the housing **21**, and one end of the main valve member **29** faces toward a coupling member **30**. Two axially spaced annular grooves **31** and **32** are cut into the periphery of the main valve member **29** and the supply-discharge ports **27** and **28** open into the grooves **31** and **32** respectively. At the two ends of the main valve member **29** are the valve chambers **35** and **36** housing first flap valves **33** and second flap valves **34**, which are rotatable with respect to the main valve member **29** (see FIGS. **5** and **6**). In the chamber **35**, a valve member **37**, having radial projections which form the first flap valves **33** is so housed as to be able to swing around the axis of the chamber. In the other chamber **36**, one end of a steering shaft **38** having radial projections forming the second flap valves **34**, is mounted to swing freely around the axis of the chamber. The steering shaft **38** is aligned with the axis of the steering shaft **22** and is resiliently connected via a double torsion-bar **40** (to be described later) to the steering shaft **22**. The steering shaft **38** is supported in a sealed bearing member **39** fastened to one end of the servo valve housing **21**, and the projecting right end of the steering shaft **38** is connected via couplings and the like (not shown) to a steering column and a steering wheel to be operated by the driver.

The steering shaft **38** with the projections forming the second flap valves **34**, the valve member **37** with the projections forming the first flap valves **33**, and the coupling member **30** engaging the steering shaft **22** are connected by pins to the double torsion-bar **40** which is coaxial with the steering shafts **38** and **22**, which are resiliently coupled together by the double torsion-bar **40**. The double torsion-bar **40** comprises resilient portion **40*a*** having a relatively small diameter and a small spring constant, and a resilient portion **40*b*** having a larger diameter and a larger spring constant than that of the smaller diameter portion **40*a***.

The valve chamber **35** which houses the first flap valves **33**, as shown in FIG. **5**, communicates with the supply port **25**, and on opposite sides of the first flap valves **33** in the chamber are the distribution ports **35a** and **35b**.

When the first flap valves **33** are in a neutral position there is a specified gap between the valve seats **33a** on opposite sides of the first flap valves and the distribution ports **35a** and **35b**.

The valve chamber **36**, which houses the second flap valves **34**, communicates with the discharge port **26** and has, as shown in FIG. **6**, a pair of jet ports **36a** and **36b** on opposite sides of the second flap valves **34**. When the second flap valves **34** are in a neutral position, there is a specified gap between the valve seats **34a** provided on opposite sides of second flap valves **34** and the jet ports **36a** and **36b**. The gap formed between the valve seats **34a** on the second flap valves **34** and the jet ports **36a** and **36b** is made slightly wider than that between the valve seats **33a** on opposite sides of the first flap valves **33** and the distribution ports **35a** and **35b**.

Cylindrical holes **41** extend axially through the main valve member **29**, as shown in FIG. **4**, and terminate adjacent the inner sides of the first and second flap valves **33** and **34**. In the cylindrical holes **41** are the slidably mounted pressure-exerting pistons **42** and **43** and a free piston **44** positioned between the pistons **42** and **43**. Springs **45** and **46** are inserted while compressed in the spaces between the pressure-exerting pistons **42** and **43** and the free piston **44**.

The spaces between the pressure-exerting pistons **42** and **43** and the free piston **44** communicate via narrow holes **47** and **48** respectively with the annular concave grooves **31** and **32** of the main valve member **29**. Thereby, the first flap valves **33** and the second flap valves **34** are subjected by the pistons **42** and **43** to pressures proportional to the pressures generated in the left compartment **15a** and in the right compartment **15b** of cylinder **15**, and unstable vibration of first and second flap valves **33** and **34** can be eliminated by the frictional force between the pressure-exerting pistons **42** and **43** and the first and second flap valves **33** and **34**, thus making the valve characteristics more stable.

When a surge pressure develops in the hydraulic pressure circuit, for instance, in conduits **59** and **60**, the free piston **44** is displaced in response to the surge pressure, thereby absorbing the surge pressure so that the driver will not feel an unpleasant impact. At the same time, the piston **42** projecting into the valve chamber **35**, which communicates with the supply port **25**, is restricted against displacement to the right in FIG. **4** by a step in the wall of the cylinder hole **41**, whereby the piston **42** is prevented from plugging the narrow hole **47**.

In FIGS. **5** to **8**, the distribution ports **35a** opening into the valve chamber **35** (FIG. **5**) communicate through fluid conduits **49** with the annular concave groove **31**, while the distribution ports **35b** communicate through fluid conduits **50** with the annular concave groove **32**. Meanwhile, the jet ports **36a** opening into the valve chamber **36** communicate through fluid conduits **51** with the annular concave groove **32**, while the jet ports **36b** communicate through fluid conduits **52** with the annular concave groove **31**.

As indicated in FIG. **1**, the supply port **25** communicates through a supply pipe **53** with a pressure fluid supply pump **54**, while the discharge port **26** communicates through a discharge pipe **55** with a fluid tank **56**. The supply pump **54** is driven by the driving means **57** such as an automobile engine, and the delivery pressure is kept at less than a predetermined value by means of a relief valve **58**.

When, in the above arrangement, the supply pump **54** is started to supply a pressure fluid through the supply port **25** into the servo valve housing **21**, the pressure fluid admitted into the valve chamber **35** passes through the distribution ports **35a** and **35b** (shown in FIG. **5**), through the fluid conduits **49** and **50**, through the annular grooves **31** and **32** to the supply exhaust ports **27** and **28** of the hydraulic pressure cylinder **15**. Said fluid also passes through the fluid conduits **51** and **52** to the jet ports **36a** and **36b**. At the neutral positions of first flap valves **33** and second flap valves **34**, both the distribution ports **35a** and **35b** and both the jet ports **36a** and **36b** are open, and accordingly the greater part of the fluid supplied passes through the jet ports **36a** and **36b** and discharge port **26** back to the fluid tank **56** to be reclaimed. Therefore, the pressures in the left compartment **15a** and in the right compartment **15b**, without rising, are balanced with each other and in consequence the piston **16** does not move.

Now, when the steering wheel is turned clockwise, since the resistance of the ground to turning of the steerable wheels is transferred to the steering shaft **22**, the smaller diameter resilient portion **40a** of the double torsion-bar **40** is first twisted, while the first flap valves **33**, together with the steering shaft **38**, are turned clockwise relative to the main valve member **29** in FIG. **5**. The distribution ports **35b** are thereby closed, while the other distribution ports **35a** are opened and in consequence the entry of pressure fluid into the distribution ports **35b** is restricted.

When the steering shaft is turned further clockwise, the large diameter resilient portion **40b** of the double torsion-bar **40** is twisted, and the second flap valves **34** are rotated clockwise with respect to the first flap valves **33**. The jet ports **36b** are thereby closed, while the other jet ports **36a** are opened and in consequence, the greater part of pressure fluid supplied to the servo valve housing **21** by the supply pump **54** is delivered through the first flap valves **33** to the distribution ports **35a** only, by way of the jet ports **36b** through the fluid conduits **49**, the annular concave groove **31**, and the fluid conduits **52**.

As stated above, the jet ports **36b** are closed by the second flap valves **34** and the flow to the discharge port **26** is restricted. Accordingly, the pressure fluid passes through the annular concave groove **31**, the port **27** and the conduit **59** to the left compartment **15a** of the hydraulic pressure cylinder **15**, thereby urging the piston **16** to the right in FIG. **3**. Thus, the fluid within the right hand compartment **15b** of the cylinder **15** is discharged into the fluid tank **56** through the conduit **60**, the port **28**, the annular concave groove **32**, the fluid conduits **51** and the wide-open jet ports **36a**. In this manner, a slight steering torque applied by the driver to the steering wheel is boosted by the piston **16** to assist it in turning the rotating shaft **12**, which turns the steerable wheels in right direction.

On this occasion, the pressure fluid supplied to the distribution ports **35a**, passing through the fluid conduits **49** and the annular concave groove **31**, is introduced through the narrow hole **47** into the cylinder holes **41** in the main valve member **29**, thereby causing the pistons **42** to press against the first flap valves **33** and at the same time causing the free pistons **44** to displace, thereby compressing the springs **46** and urging the pistons **43** against the second flap valves **34**. Thus, the frictional force developed between the pistons **42** and **43** and the first and second flap valves **33** and **34** damps the instable vibration of the first and second flap valves **33** and **34** to ensure excellent stability. Therefore the driver does not feel the vibration and accordingly an improved steering performance as well as the elimination of unpleasant noise due to vibration results.

Vibration of the flap valves **33** and **34** or a jerky turning of the steering wheel suddenly changes the gap between the distribution ports **35a** and **35b** and the valve seats **33a** and as a result an abnormal "surge" pressure develops in the hydraulic pressure circuit. The free pistons **44** move in accordance with this surge pressure and the displacement of the free pistons **44** absorbs the surge pressure. Thus, even if a surge pressure develops in the hydraulic pressure circuit, the steering performance can be maintained without the driver feeling this pressure.

FIG. **9** illustrates another embodiment of the present invention which is essentially no different from the preceding embodiment except that, in the former, the vibrations of the first and second flap valves **33** and **34** and the surge pressure are effectively damped and absorbed by an additional throttle valve installed in the hydraulic pressure circuit.

Constrictions **61** and **62** are respectively provided in conduits **59** and **60** which connect the supply-exhaust ports **27** and **28** to the right compartment **15***a* and the left compartment **15***b* of the hydraulic pressure cylinder **15**. Otherwise, the arrangement is entirely the same as shown in FIG. **4**, so identical symbols are used to indicate the identical parts and further description is omitted here. The vibrations excited in the first and second flap valves **33** and **34** are mostly attributable to the steerable wheels. For instance, when the steerable wheels of a running car are forcibly deflected the main valve member **29** is forced to deflect in response to pressure exerted through the rotating shaft **12**, the nut member **23** and the steering shaft **22**. As a result the gaps between the distribution ports **35***a* and **35***b*, and jet ports **36***a* and **36***b* of the main valve member **29** and the valve seats **33***a* and **34***a* of the first and second flap valves **33** and **34** change to cause an imbalance in the hydraulic pressure acting on both sides of the flap valves **33** and **34**, thereby exciting vibrations in the flap valves **33** and **34**. It is therefore desirable to arrange the mechanism in such a way that, even when a strong external deflecting force is exerted on the steerable wheels, the main valve member will not be greatly deflected. For this purpose, the constrictions **61** and **62** are provided in the hydraulic pressure circuit. Thus, the action of the piston **16** mechanically connected to the steerable wheels is made slightly sluggish. This arrangement permits the strong deflecting force exerted on the steerable wheels to be absorbed, with the small vibration excited in the flap valves **33** and **34** damped by the frictional force due to the push pistons **42** and **43**, and the accompanying small surge pressure being absorbed by displacements of free pistons **44**. The vibrations of flap valves **33** and **34** and the surge pressure can thereby be reliably and effectively damped.

FIG. **10** illustrates still another embodiment of the present invention, in which there is no free piston **44** between the pressure-exerting pistons **42** and **43**, and there is a surge pressure absorbing device inserted in the conduits **59** and **60**.

In the embodiment illustrated in FIG. **10**, a compressed spring **45** is inserted between two pressure-exerting pistons **42** and **43**. A narrow hole **63** is cut in the pressure-exerting piston **42** contacting the first flap valves **33**, and a pressure fluid is admitted through the supply port **25** to the back side of the pressure-exerting pistons **42** and **43**. In this manner, the first flap valves **33** and the second flap valves **34** are respectively subjected to pressure from the push pistons **42** and **43**, and because of the frictional force between the first flap valves **33** and the pressure-exerting pistons **42** and between the second flap valves **34** and the pressure-exerting pistons **43**, unstable vibrations of the first and second flap valves **33** and **34** can be eliminated to stabilize their valve characteristics. The friction generated between the both flap valves **33** and **34** and push pistons **42** and **43** is proportional to the fluid pressure in the valve chamber **35** and it offers a relatively large resistance to vibration by the flap valves **33** and **34**.

At the same time, a surge pressure absorbing device is inserted in each of the conduits **59** and **60** which connect the right and left compartments **15***b* and **15***a* of the hydraulic cylinder **15** with the ports **27** and **28**. As illustrated in FIG. **11**, valve holes **64** and **65** are provided midway in the conduits **59** and **60**, and in the valve holes **64** and **65** are slidably mounted spools **66, 67, 68** and **69**, and between the spools **66** and **67** and the spools **68** and **69** compressed springs **70** and **71** are so positioned that the forces exerted by these springs normally keep the spools **66, 67, 68** and **69** in positions at both ends of the valve holes **64** and **65**. The spools **66, 67, 68** and **69** are respectively equipped with conduits **72, 73, 74** and **75** provided with constrictions which act as throttles, and these conduits provide communication between the right and left compartments of the hydraulic cylinder **15** and the ports **27** and **28**. A space **76** is left between the spools **66** and **67** and a space **77** between the spools **68** and **69** and the spaces **76** and **77** are connected by a conduit **78**. Occasionally in the hydraulic pressure circuit an abnormal pressure, i.e., a surge pressure develops, which displaces the spool **67** or **69** to compress the spring **70** or **71**, which absorbs this pressure. As the displacement of the spool for absorbing surge pressure is accomplished by means of the fluid leaking out of the space **76** or **77** through the valve hole **64** or **65**, the speed of displacement is extremely slow, and an effective absorption of surge pressure can be counted upon.

FIG. **12** illustrates a fourth embodiment of the present invention, in which the spaces **76** and **77** formed between spools **66** and **67** and **68** and **69** (not shown) are connected through a discharge conduit **79** to the valve chamber **36** which leads to the discharge port **26**.

In this embodiment, just as hereinbefore described, displacement of the spool caused by a surge pressure absorbs the surge pressure.

FIG. **13** illustrates a fifth embodiment of the present invention, which is basically the same as the embodiment illustrated in FIG. **1** except that a free piston **80** is installed independently, separate from the pressure-exerting pistons **42** and **43**.

In this embodiment, a valve box **81**, secured to the servo valve housing **21**, is provided with a valve chamber **82** having a relatively large diameter and a free piston **80** is mounted in the chamber **82**. Compressed springs **83** and **84** are provided on both sides of the piston **80** and the open end of the valve hole **82** is closed by a plug **85**. The valve chamber **82** on the left side of the free piston **80** is connected through the port **27** to the annular concave groove **31** and the conduit **59**, while the valve chamber **82** on the right side of the free piston **80** is connected through the port **28** to the annular concave groove **32** and the conduit **60**.

In this embodiment, in which the free piston **80** is installed in the valve box **81** which is separated from the servo valve housing **21**, the valve chamber **82** can be sufficiently enlarged to receive the free piston **80** having a relatively large diameter and accordingly with a large change in the volume of the valve chamber **82** due to the displacement of the free piston **80**, it is possible to effectively absorb the surge pressure, and the values of the spring constants of compression springs **83** and **84**, which act on the free piston **80** can be so selected within a wide range that the best absorption of surge pressure may be assured.

According to the present invention, the main valve member is provided with a plurality of cylindrical holes, each of which has a pressure-exerting piston, and these pressure-exerting pistons are urged against a plurality of flap valves by means of a plurality of springs and the friction between the pressure-exerting pistons and flap valves can thereby effectively damp the vibrations of the flap valves. Meanwhile, a pressure proportional to the fluid pressure in the hydraulic cylinder which boosts the steering torque is exerted by the pressure-exerting pistons on the flap valves, so that the vibrations of the flap valves can be adequately damped.

A free piston is installed between two pressure-exerting pistons and a displacement of the free piston can absorb the surge pressure developed in the hydraulic circuit.

Moreover, surge pressure absorbing devices such as throttle valves, spools, and free pistons are provided in the hydraulic circuit so that the surge pressure in the circuit may be absorbed. Thus, the valve characteristics of the flap valve mechanism can be improved, unpleasant noise due to flap valve vibration and the driver's unpleasant feeling due to surge pressure in the hydraulic circuit can be eliminated, thereby stabilizing and enhancing the steering performance.

The present invention is not restricted to the details of the several embodiments disclosed herein, since various modifications can easily be made by a man skilled in the art without thereby departing from the basic concepts defined by the following claims.

What is claimed is:

1. In a power steering apparatus comprising

a hydraulic motor for boosting manually applied steering torque, which motor is supplied through a hydraulic circuit including supply and discharge passages leading to said motor, two rotatably steering shaft members aligned with each other and arranged to permit relative rotation therebetween, one of which shaft members is operably connected to control means in said hydraulic circuit and the other of which shaft members is operably connected to a steering wheel,

said control means comprising a main valve member aligned between said steering shaft members and mounted within a valve chamber to divide said chamber into two parts,

first and second flap valves mounted within said two valve chamber parts respectively to distribute pressure fluid to said hydraulic motor selectively through relative rotation therebetween, one of said flap valves being connected to rotate with said one shaft member and the other flap valve being driven by said one shaft member through a resilient connection,

at least two passageways in said main valve member extending between said first and second flap valve members, and a pair of axially spaced pressure-exerting pistons slidably mounted in each of said passageways in contact with said flap valves,

the improvement which comprises means for absorbing surge pressures in said hydraulic circuit including free piston means mounted in parallel with said motor between the supply and discharge passages leading to said motor, and

spring means biassing said pistons against said flap valves to dampen vibrations of said valves and biassing said free piston means in opposite directions.

2. Power steering apparatus as claimed in claim **1** further comprising at least one constriction in one of the supply and discharge passages leading to said hydraulic motor.

3. In a power steering apparatus comprising a hydraulic motor for boosting manually applied steering torque, which motor is supplied through a hydraulic circuit including supply and discharge passages leading to said motor,

two rotatable steering shaft members aligned with each other and arranged to permit relative rotation therebetween, one of which shaft members is operably connected to control means in said hydraulic circuit and the other of which shaft members is operably connected to a steering wheel,

said control means comprising a main valve member aligned between said steering shaft members and so mounted within a valve chamber as to divide said chamber into two parts,

first and second flap valves positioned within said two valve chamber parts respectively and adapted to distribute pressure fluid to said hydraulic motor selectively through relative rotation therebetween, one of said flap valves being connected to rotate with said one shaft member and the other flap valve being driven by said one shaft member through a resilient connection,

at least two passageways in said main valve member extending between said first and second flap valve members, and a pair of axially spaced pressure-exerting pistons slidably mounted in each of said passageways,

the improvement which comprises means for absorbing surge pressures in said hydraulic circuit including a free piston slidably mounted between each pair of pressure-exerting pistons, one compression spring inserted between each pressure-exerting piston and the adjacent free piston, and ducts in said main valve member between said pressure-exerting pistons and said free piston connecting the passageways in said valve member to the supply and discharge passages leading to said hydraulic motor.

4. Power steering apparatus as claimed in claim **3** further comprising at least one constriction in one of the supply and discharge passages leading to said hydraulic motor.

5. In a power steering apparatus comprising

a hydraulic motor for boosting manually applied steering torque, which motor is supplied through a hydraulic circuit including supply and discharge passages leading to said motor,

two rotatable steering shaft members aligned with each other and arranged to permit relative rotation therebetween, one of which shaft members is operably connected to control means in said said hydraulic circuit and the other of which shaft members is operably connected to a steering wheel,

said control means comprising a main valve member aligned between said steering shaft members and so mounted within a valve chamber as to divide said chamber into two parts,

first and second flap valves positioned within said two valve chamber parts respectively and adapted to distribute pressure fluid to said hydraulic motor selectively through relative rotation therebetween, one of said flap valves being connected to rotate with said one shaft member and the other flap valve being driven by said one shaft member through a resilient connection,

at least two passageways in said main valve member extending between said first and second flap valve members,

a pair of axially spaced pressure-exerting pistons slidably mounted in each of said passageways with a compression spring therebetween which urges said pistons into vibration damping contact with said flap valves,

a valve housing containing said valve chamber, and means for conducting pressure fluid from one of the parts of said valve chamber into the space formed between said two pressure-exerting pistons,

the improvement which comprises means for absorbing surge pressures in said hydraulic circuit including a valve body externally secured to said valve housing, an auxiliary valve chamber in said valve body, a free piston slidably mounted in said auxiliary valve chamber, and compression springs in said auxiliary valve chamber for urging said free piston in opposite directions, the spaces in the auxiliary valve chamber defined between said free piston and the ends of said auxiliary valve chamber being respectively connected to the supply and discharge passages leading to said hydraulic motor.

6. Power steering apparatus as claimed in claim **5** further comprising at least one constriction in one of the supply and discharge passages leading to said hydraulic motor.

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