Feb. 17, 1970 TAMAKI TOMITA ET AL 3,495,539

ROTARY PUMP

Filed Dec. 15, 1967 8 Sheets-Sheet 1

FIG. 1

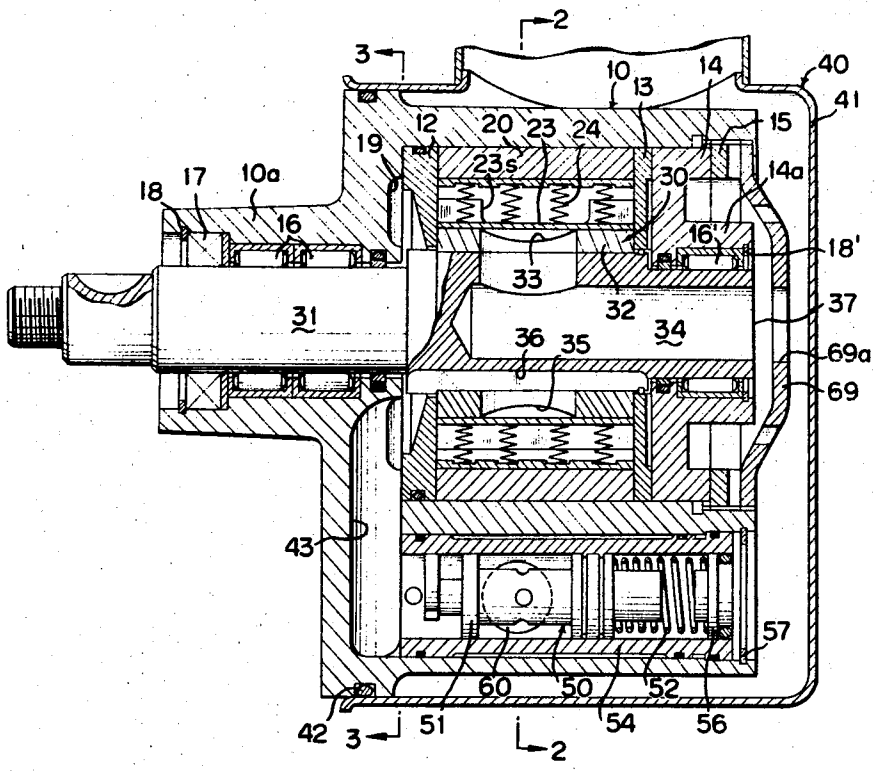


FIG. 2

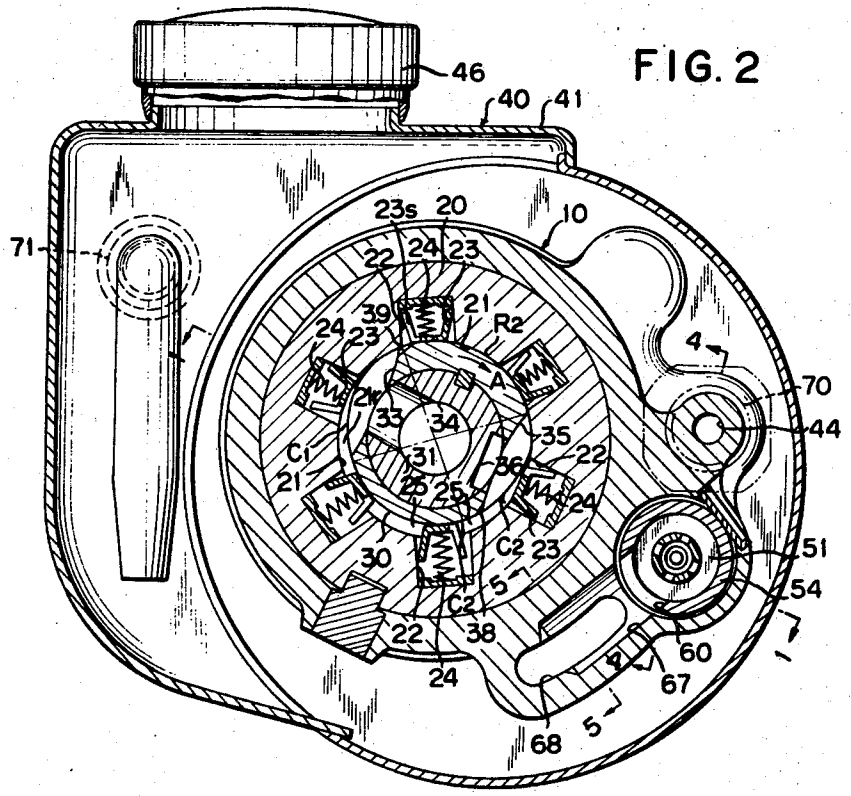

46
40
41
71
23s
20
10
22
24
23
21
R2
39
4
70
44
A
33
34
35
C1
31
36
22
21
25
24
30
C2
23
5
C2
22
38
60
67
24
68
51
54
1

FIG. 3

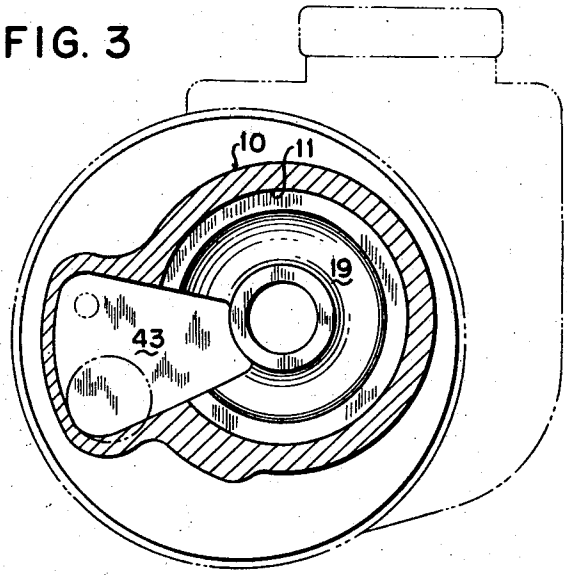

10
11
19
43

FIG. 4

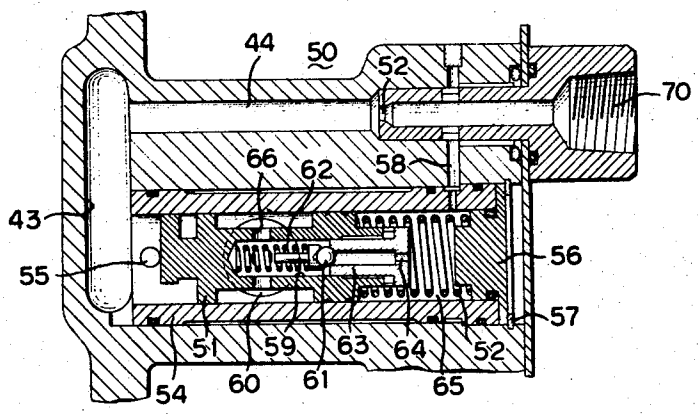

44
50
52
70
66
62
58
43
55
56
57
51
59
63
64
52
54
60
61
65

FIG. 5

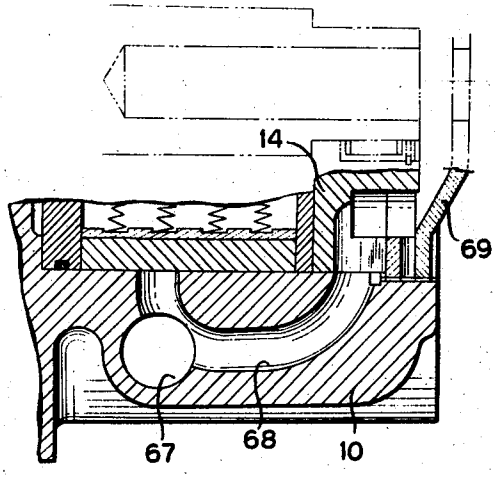

14
69
67
68
10

FIG. 6

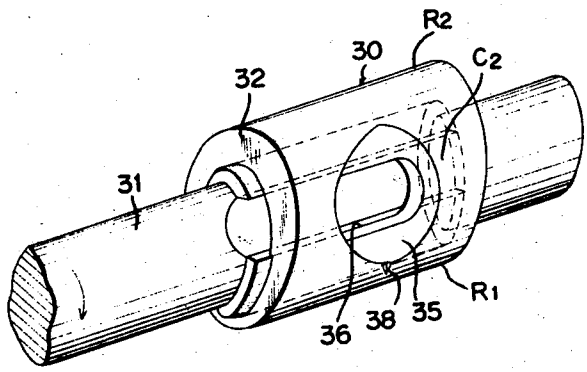



FIG. 7

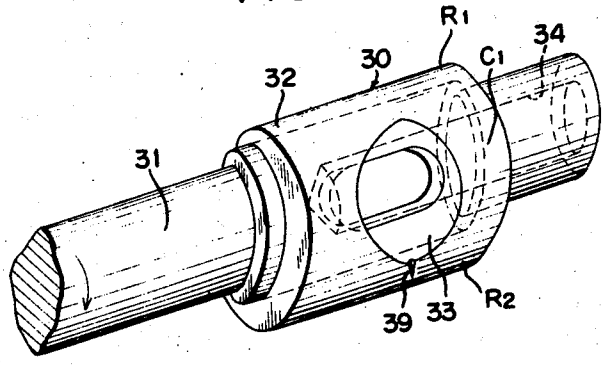


FIG. 8

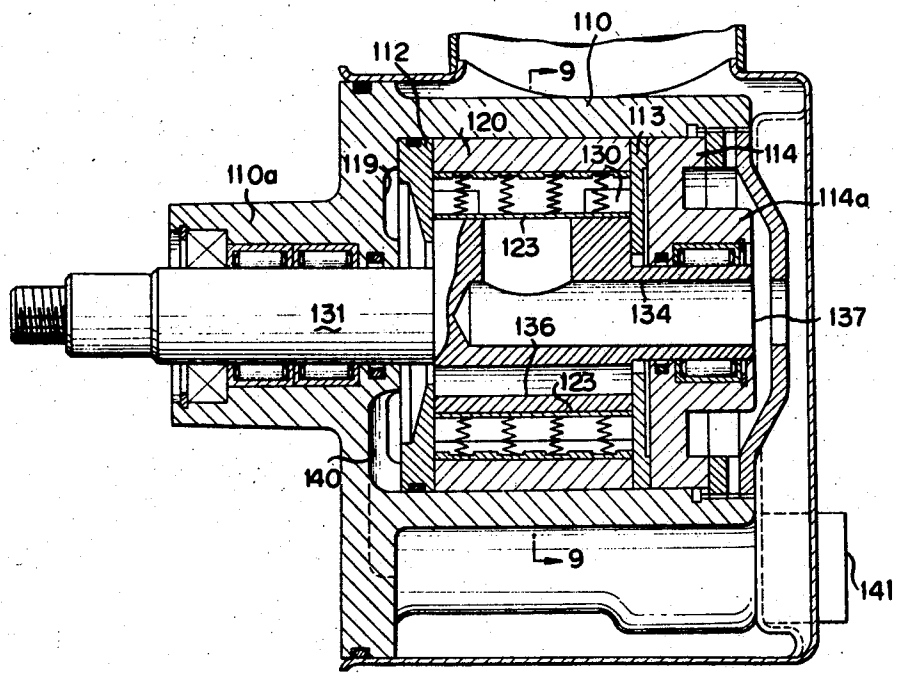

110
112
9
120
113
114
110a
119
130
114a
123
131
136
134
137
123
140
9
141

FIG. 11

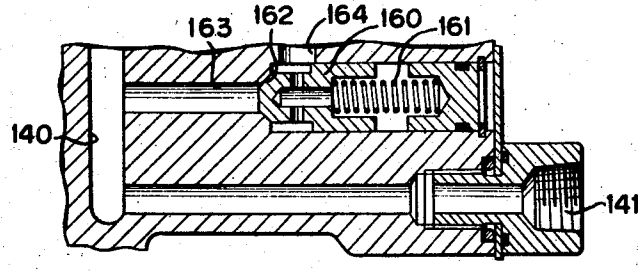

163
162
164
160
161
140
141

FIG. 9

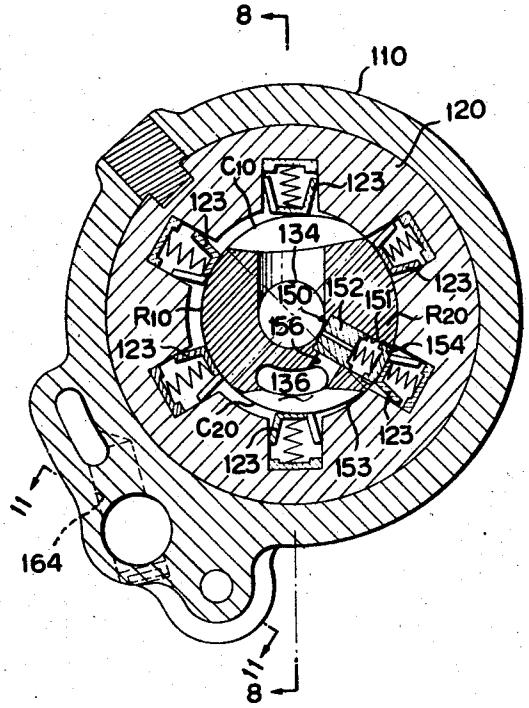

8
110
120
C10
123
123
134
123
150
152
151
R10
156
R20
123
154
136
C20
123
123
153
11
164
11
8

FIG. 10

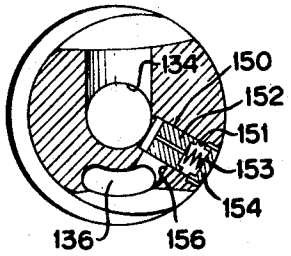

134
150
152
151
153
154
136
156

FIG. 12

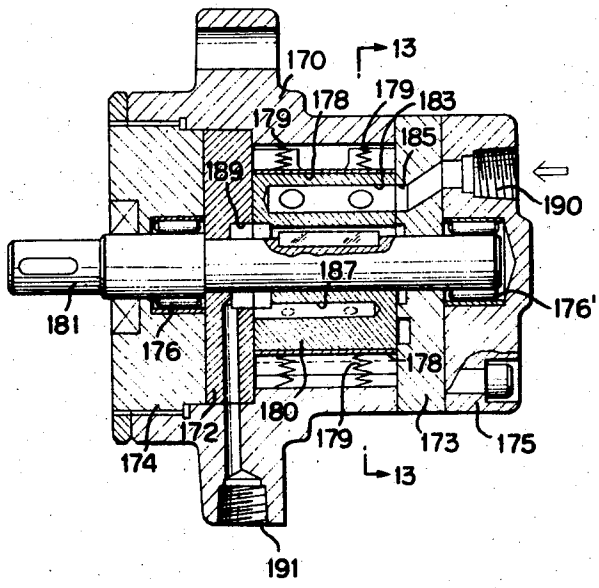

170
13
178
179
179
183
185
189
190
187
181
176
176'
178
180
172
174
179
173
175
13
191

FIG. 13

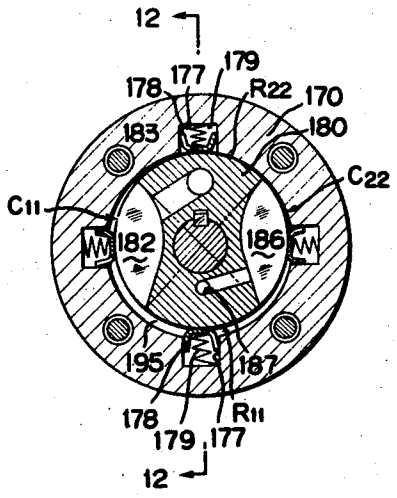

12
179
177
178
R22
170
183
180
C22
C11
182
186
195
187
178
R11
179
177
12

FIG. 14

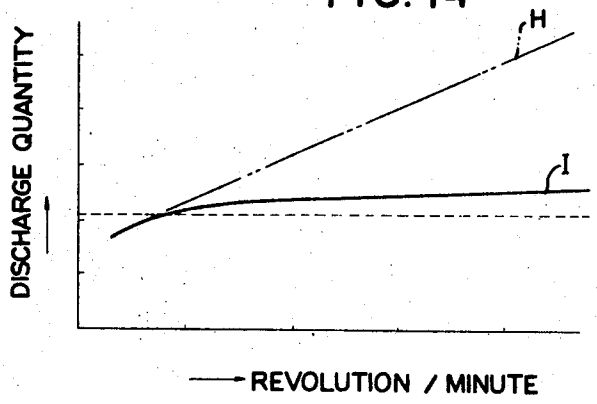
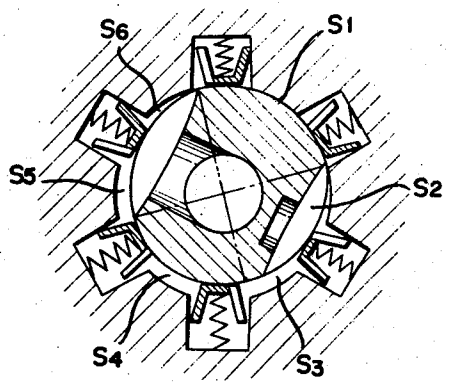
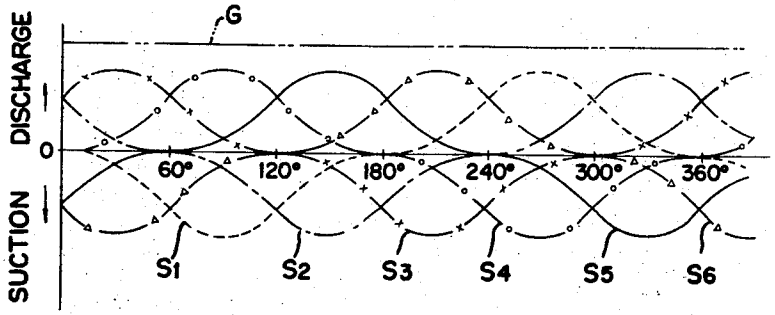



FIG. 15

(I)




(II)



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3,495,539

# ROTARY PUMP

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Int. Cl. F04c *1/00, 3/00*
U.S. Cl. 103—123 9 Claims

## ABSTRACT OF THE DISCLOSURE

A rotary pump is provided in which a driving shaft is rotatably mounted in a cylindrical casing member and is provided with a concentric suction channel and a discharge channel in the outer peripheral surface thereof, the shaft carrying an eccentric rotor thereon which forms a chamber between the rotor and the casing member, the rotor being provided with two diametrically opposite outer surfaces concentric with the shaft but with different radii and first and second cam surfaces which are located between the opposite outer surfaces and have openings in communication with the suction and discharge openings respectively. A plurality of abutments are loosely received in radial grooves circumferentially disposed with uniform spacing in the inner peripheral surface of the casing member, each abutment being resiliently urged against the outer peripheral surface of the rotor.

This invention relates to a rotary pump. It relates especially to an improvement in a type of rotary pump, which comprises a cylindrical casing member and a rotor housed therein, a plurality of sealing elements distributed within said cylindrical casing member so as to act elastically on the outer periphery of said rotor and to define a plurality of separated operating spaces within said cylindrical casing member, a non-circular profiled cam surface formed on the outer periphery of said rotor in order to increase and decrease periodically the volumes of said operating spaces, and means to connect said operating spaces periodically with the suction device and the discharge device provided on the outer periphery of said rotor.

In a conventional rotary pump of this type, an eccentric rotor is generally used and this eccentric rotor cooperates with rollers distributed within a casing member, and its suction and discharge devices are provided in said casing member. However, these known rotary pumps have some inconveniences in the sealing means, and the suction and discharge devices, whereby the leakage of fluid in the high pressure region has often been experienced, and further, the harmful vibrations or noises due to the pulsation of the discharge pressure of fluid has also been provoked. Moreover, as is always experienced in general kind of pumps, there is too much excess fluid for the pump delivery when the pump works with high speed or the hydraulic motor supplied by the pump is running under no load, whereby considerable power has been wasted for delivering the large quantity of unnecessary fluid. For example, in the case of application to the power steering gear of an automobile, the hydraulic motor of the power steering gear is running substantially under no load, when the engine is rotating at high speed, i.e. the vehicle is running at high speed, whereas the pump is forced to rotate at high speed to deliver the large quantity of fluid. Thus, the output of the engine is substantially lowered by the power consumed for such pump operations, which gives rise to discussion for power loss of the engine.



The object of this invention is to improve such type of rotary pumps as above-mentioned so as to obtain their good performances in the high pressure region and at the high speed rotation thereof, and thereby to lower the power loss of the engine. By this improvement, it is possible to provide a suitable rotary pump, especially in the application to the power steering unit of an automobile.

In order to accomplish this object, according to the principal characteristic of the invention, the operational fluid is compressed previously before discharge and is led slowly into the discharge channel, in order to make uniform the quantity of pump delivery and to avoid the pulsation of the discharge pressure of the pump, and means is provided to enhance the elastic following performances of the sealing elements is assured.

Moreover, according to this invention, the excess fluid produced at high speed rotation of the pump or under no load working of the hydraulic motor escapes through a lower flowing resistanct, whereby the power loss of the engine is considerably lowered. In a variant of this invention, there is provided a novel flow control valve actuated by centrifugal force for escaping said excess fluid.

In the following text, the present invention will be described in detail, in connection with the accompanying drawings, in which:

FIG. 1 shows a longitudinal sectional view of an example of the invention, along the line **1—1** of FIG. 2;

FIGS. 2 and 3 are sectional views along the lines **2—2** and **3—3** of FIG. 1 respectively;

FIGS. 4 and 5 are sectional views along the lines **4—4** and **5—5** of FIG. 2 respectively;

FIGS. 6 and 7 are perspective views of a pump-rotor at two different angular positions respectively,

FIG. 8 shows a longitudinal sectional view of a variant of the invention, along the line **8—8** of FIG. 9;

FIG. 9 is a sectional view along the line **9—9** of FIG. 8;

FIG. 10 is a sectional view showing the operation of a flow control valve;

FIG. 11 is a partial sectional view along the line **11—11** of FIG. 9;

FIG. 12 is a longitudinal sectional view of another variant of the invention, along the line **12—12** of FIG. 13;

FIG. 13 is a sectional view along the line **13—13** of FIG. 12;

FIG. 14 is a diagram showing the discharge characteristics;

FIG. 15 is a schematic drawing showing the suction cycles and the discharge cycles of a rotary pump according to the invention.

Now, a first example of the present invention will be explained with reference to FIGS. 1 to 7.

In a housing **10**, there are housed a cylindrical casing member **20** and a rotor **30**, and two discs **12** and **13** are applied respectively on each end surfaces of said cylindrical casing member **20** and said rotor **30**, and by a blocking member **14** clamped by a ring nut **15**, said two discs **12** and **13** are maintained in their respective positions. The outer housing **10** and the blocking member **14** are respectively projected outwards and coaxially with said outer housing **10**, and within these

projected cylindrical portions **10***a* and **14***a*, there are provided bearings **16**, **16'** and a seal member **17** fixed by snap rings **18**, **18'** in order to support a driving shaft **31** of the rotor **30**. One end of this shaft **31** is connected with an engine (not shown) by usual means, and on the central portion of this shaft **31**, there is fixed a cam ring **32**, the profile of which will be hereinafter explained, in order to constitute the rotor **30** having a non-circular profile. Thus, between the inner peripheral surface of the cylindrical casing member **20** and the rotor **30**, there is formed a chamber **21***r* having a crescent-shaped section. Moreover, in the inner peripheral surface of said casing member **20**, there are provided a plurality of radial grooves **22** distributed circumferentially with equal spacing, and within each said groove **22**, an abutment **23** with a compressed spring **24** is loosely received. These abutments **23** are pressed uniformly against the outer peripheral surface of the rotor **30** by the actions of the springs **24** and by the aid of contacts with side walls of said receiving groove **22**, whereby said crescent-shaped chamber **21***r* can be separated into a plurality of individual working spaces **25**. In this example, there are provided six abutments **23**, and six individual spaces **25** can be separated. It is desirable that each abutment **23** is formed by bending sheet metal in U shape in order to obtain its excellent following performance in the high speed region. Moreover, a notch **23***s* is provided on one side of each U shaped abutment, whereby an effective sealing pressure is always applied on the rotor **30** in the high pressure region, by the aid of back pressure due to the discharge pressure fluid led into the back side of said abutment.

On the outer peripheral surface of the cam ring **32** which constitutes the rotor **30**, there are formed a first cam surface $C_1$ having a radius gradually decreasing according to its rotation, and a second cam surface $C_2$ having a radius gradually increasing according to its rotation, in order to increase and decrease periodically the volumes of the respective working spaces **25** sealed and separated by said abutments **23**. These two cam surfaces $C_1$ and $C_2$ are connected to each other by the intermediary of a surface $R_1$ having a small fixed radius and a surface $R_2$ having a large fixed radius. This surface $R_2$ has a radius of curvature almost the same as the radius of the inner peripheral surface of the cylindrical casing member **20** so as to assure a sliding fit therein. The connections between the surfaces $R_1$, $R_2$ and the cam surfaces $C_1$, $C_2$ are effected without abrupt change of curvature. These surfaces $R_1$ and $R_2$ serve to avoid the instantaneous and abrupt change of the radial displacement of the abutment **23** from the center outwards and vice versa. The abutments **23** are guided by these surfaces $R_1$ and $R_2$ in such manner that each abutment **23** begins its radial displacement after staying some time on the members $R_1$ and $R_2$, whereby jumping movement of said abutment **23** is avoided and the sealing ability thereof in the high speed region is enhanced.

As shown in FIG. 7, a recess **33** formed on the central portion of the first cam surface $C_1$ is connected with a suction channel **34** machined in the center of the driving shaft **31**. This constitutes a suction device. As shown in FIG. 6, a recess **35** formed on the central portion of the second cam surface $C_2$ is connected with a discharge channel **36** machined on the outer peripheral surface of the driving shaft **31**, said channel **36** being connected with a fluid collecting groove **19** machined in the side wall of the outer housing **10**. This constitutes a discharge device. Owing to these devices, the suction and discharge of fluid are effected advantageously by the aid of centrifugal action produced by the rotation of the rotor. The suction is carried out, with advantage, by the aid of greater centrifugal action from the axis of the driving shaft where said suction channel is located, whereas the discharge is carried out against far less centrifugal action from the outer peripheral surface of the driving shaft where said discharge channel is located. Through such discharge device, the discharged fluid is led outside of the discs **12** and **13**, and by the fluid pressure acting on said discs **12** and **13** from outside, the side surfaces of the rotor are sealed and no leakage of the fluid takes place, and thereby any axial thrust by the rotor **30** is balanced.

The opening edges of the recesses **33** and **35**, which constitute the suction device and the discharge device, are extended circumferentially on the outer peripheral surface of the rotor, so that the angular aperture of the recess **33** in this extended direction at the center of rotation of the rotor is almost the same as that of the first cam surface $C_1$ at the center of rotation. Accordingly, it is possible to connect the respective working spaces separated by the abutments **23** (the left side spaces of FIG. 2) with the suction channel **34** in sequence, so as to effect the suction operation.

On the opening edge of the recesses **33** towards the surface $R_2$, there is provided a convergent groove **39** extending into said surface $R_2$ in order to avoid the confinement of the fluid within the working space which is going into the suction phase of the working cycle after having finished the discharge phase. The angular aperture of the recess **35** at the center of rotation of the rotor is almost the same as that of the second cam surface $C_2$ at the center of rotation, whereby the respective working spaces (the right side spaces of FIG. 2) are connected with the discharge channel **36** in sequence. On the opening edge of the recess **35**, there is provided a convergent groove **38** in order to lead the fluid in the working space which is going into the discharge phase slowly into the discharge channel **36**, whereby the pulsating variations of the discharge pressure are avoided.

In the pump constructed as mentioned above, the sealing actions are effected substantially on the two diametrically opposed portions of the surface $R_1$ and the surface $R_2$ of the rotor **30**, and thereby the space within the casing member is divided into two spaces, i.e. a space for suction phase and a space for discharge phase. The space for suction phase (the left side space of FIG. 2) surrounding the first cam surface $C_1$ is connected with the suction channel **34** by means of the recess **33**, and the space for discharge phase (the right side space of FIG. 2) surrounding the second cam surface $C_2$ is connected with the discharge channel **36** by means of the recess **35**. When the motor **30** is rotated clockwise as shown by the arrow A, the volume of each working space adjacent the recess **33** is gradually increased to suck the fluid from the suction channel **34**, and the volume of each working space adjacent the recess **35** is gradually decreased to discharge the fluid into the discharge channel **36**. Each working space effects, per revolution of the rotor **30**, the suction phase and the discharge phase alternately. However, as shown in FIG. 15, the working cycles of the various working spaces are out of phase with one another, according to the rotation of the rotor, towards the direction of the arrow A. The quantities of fluid discharged from the working spaces are joined together and delivered in almost constant volume G as shown in FIG. 15 II. Both opening edges of the recesses **33** and **35** serve to transfer the time of connection of the suction channel and the discharge channel.

An outer cover **41** encloses the housing **10** and constitutes a fluid reservoir **40**, the opening end of which is sealed by a ring **42** (FIG. 1) inserted between said cover **41** and the outer housing **10** in order to reserve the fluid therein, and a cap **46** is provided on the top of the reservoir **40**.

The fluid is sucked from a suction port **37** opened in a reservoir **40** into the suction channel **34**.

The fluid discharged from the discharge channel **36** is led through the fluid collecting groove **19** and a depressed

channel 43 into a channel 44 machined within a flow control device 50 (FIG. 4).

This flow control device 50 serves to keep constant the pump delivery. As shown in FIG. 4, a spool valve 51 is inserted in a sleeve 54 provided within the outer housing 10 so as to avoid any deformation or thermal stress. The spool valve 51 is pressed by a spring 52 provided between one end of said valve and a support member 56 fixed on one end of said sleeve 54 by a snap ring 57 so as to close a by-pass port 60 and abut against a stopper 55 provided at the other end of said sleeve 54. The hydraulic pressure prevailing in the depressed channel 43 and acting on a left-hand end surface of the valve 51 is also led through the channel 44, an orifice 53 and a channel 58 so as to act on a right-hand end surface of the valve 51. Thus, by displacement of the valve 51, the escaping area of the port 60 is changed, and thereby the discharge quantity is controlled in such manner that a constant delivery is always obtained through a delivery port 70. Moreover, in a hollow space within the spool valve 51, there is provided a control valve 61 pressed by a spring 62 on the valve seat 63 having a hole 64 opening into a chamber 65 at a right-hand end of the valve 51. When the pressure in said chamber 65 led from the channel 44 through the orifice 53 and the channel 58 becomes higher than the predetermined value, said control valve 61 is opened and the fluid escapes from openings 66 to the escaping port 60. Thus, the pressure in said chamber 65 is lowered and the spool valve 51 is displaced so as to open the by-pass port 60 to exhaust the fluid into by-pass channels 67 and 68, whereby the overpressure of the fluid is avoided. The fluid exhausted into the escaping channels 67 and 68 is led along the side surface of the blocking member 14 and a guide plate 69 to the vicinity of a suction port 37.

As an opening 69*a* is provided in the guide plate 69 almost axially with the suction port 37, the fluid supplied from the reservoir 40 can flow together with the fluid exhausted from said port 60 into the suction port 37, whereby better efficiency of suction is obtained. Moreover, in this case, all the excess fluid is returned through a very small flowing resistance from the escaping port 60 to the suction port 37, and there is no need of delivery of unnecessary high pressure fluid, whereby any power loss during the high speed rotation of the engine or under no load driving is lowered.

Now referring to FIGS. 8, 9 and 10, a first variant of the invention will be explained.

As shown in the first example of the invention, in a housing 110, there are housed a cylindrical casing member 120, a rotor 130, two side discs 112, 113 and a blocking member 114; and within projected cylindrical portions 110*a* and 114*a* of said housing 110 and said blocking member 114, a driving shaft 131 on the rotor 130 is supported by bearings. On the outer peripheral surface of the rotor 30, there are provided similar cam surfaces $C_{10}$, $C_{20}$ and similar surfaces $R_{10}$, $R_{20}$ having respective fixed radius as shown in the first example; and between the casing member 120 and said rotor 130, a similar crescent-shaped chamber is formed. A plurality of abutments 123 provided in the inner periphery of the casing member 120 act on the rotor 130 and separate said crescent-shaped chamber into several independent working spaces. The rotor 130 is equipped with a similar suction device and a similar discharge device as shown in the first example.

In this variant, there is provided a channel connecting a suction channel 134 and discharge channel 136 with each other, and the effective area of this connecting channel is controlled by a flow control valve 150 in proportion to the number of revolutions per minute of the rotor 130. This valve 150 serves to adjust said area of the connecting channel in proportion to the centrifugal force produced by the rotation of the rotor, in order to keep constant the pump delivery which is apt to increase in response to the number of revolutions per minute of the rotor, by returning the excess fluid into the suction channel 134. This flow control valve 150 is constituted by a spool valve member 152 and a compression spring 153 inserted together into a valve hole 151 machined within the rotor 130 from its outer periphery towards its center, the opening of said hole being closed by a screwed plug 154. One end of said hole 151 is opened into the suction channel 134, and a channel 156 connected with the discharge channel 136 is opened on the side wall of said hole 151. The valve hole 151 which connects the suction channel 134 and the discharge channel 136, as shown in FIG. 9, is normally closed by the valve member 152. However, in response to the increasing number of revolutions per minute of the rotor 130, as shown in FIG. 10, the channel 156 is opened by the valve member 152, displacing towards the outer periphery against the action of the spring 153 due to the centrifugal action. The passing area of this channel 156 is increased in proportion to the number of revolutions per minute of the rotor, the increasing excess fluid is exhausted into the suction channel 134, and the discharge from a collecting groove 119 can be kept almost constant, as shown by the curve I in FIG. 14. Since the excess fluid flows in the short channel, the flow resistance is very low, and the power loss can be further reduced. The discharged fluid which is thus kept almost constant is led from the collecting groove 119 through a channel 140 to a delivery port 141. Said discharged fluid is also led to a pressure escaping valve 160 by a branch channel from the channel 140. The pressure escaping valve 160 is, as shown in FIG. 11, normally pressed on a valve seat 162 by a spring 161, and a by-pass channel 163 is closed. When the pressure of the discharged fluid in the channel 140 becomes higher than the predetermined pressure value, the channel 163 is opened by pressing the valve 160 against the action of the spring 161. As explained in the first example, the fluid exhausted from a side channel 164 is led to a suction port 137 and sucked again.

After the quantity of the discharged fluid has been thus adjusted, said fluid is supplied from the delivery port of the pump to any external apparatus, for example, hydraulic motors or the like and the fluid after work is returned into a reservoir, thus the circulation of the fluid being completed.

Further, a second variant of the invention will be explained with reference to FIGS. 12 and 13.

A rotor 180 is lodged in a housing 170, and side discs 172, 173 slidably applied on each side surface of said rotor 180 are fixed to the housing 170 by blocking members 174, 175. A driving shaft 181 keyed to the rotor 180 is supported by bearings 176, 176′ equipped within said blocking members 174, 175. Along the inner periphery of said housing 170, there are provided radially four receiving grooves 177 distributed circumferentially with equal spacing, and within each said groove 177, an abutment 178 with compressed spring 179 is received. These abutments 178 can be pressed on the outer peripheral surface of the rotor 180, and thereby four separated operating spaces 195 are formed within the housing 170. As explained in the first example, on the outer periphery of the rotor 180, there are formed a similar first cam surface $C_{11}$ and a similar second cam surface $C_{22}$ in order to increase and decrease periodically the volumes of the respective operating spaces 195. Also, there are provided a surface $R_{11}$ having a small fixed radius and a surface $R_{22}$ having a large fixed radius in order to connect said two cam surfaces. A recess 182 formed on the first cam surface $C_{11}$ is connected with a suction channel 183 having an opening on the side surface of the rotor 180. Opposite this opening, there is provided an annular groove 185 on the side disc 173, said groove being connected with a suction port 190 on the blocking member 175. A recess 186 formed on the second cam surface $C_{22}$ is connected with a discharge channel 187 having an opening on the other side surface of the rotor 180. Opposite this opening, there is provided an annular groove 189 on the other side disc 172, said groove being connected with a delivery port 191 on the housing 170. The angular apertures of the respec-

tive surfaces $R_{11}$ and $R_{22}$ at the center of rotation must be larger than that of the adjacent abutments at the center of rotation. If the former is less than the latter, the operating space **195** in the suction cycle and the operating space **195** in the discharge cycle are connected with each other, whereby the desired opeartion cannot be effected.

Since each space **195** separated by the abutments **178** between the housing **170** and the rotor **180** repeats to work periodically for the suction phase and the discharge phase, the recesses **182, 186** and the annular grooves **185, 189** serve to transfer the connections of the suction channel **183** and the discharge channel **187.**

As is clear from the above-mentioned descriptions, according to the present invention, the means of sealing by the abutments, and the devices of suction and discharge are improved, so as to avoid the fluid-leakage in the region of high pressure and the pulsations of the discharge pressure. Moreover, by providing the flow control device, the excess discharged fluid is returned to the suction port through the low flow resistance, whereby the power consumption for the pump is remarkably lowered, and a specially suitable pumping system for the power steering unit of an automobile is obtained.

It is to be understood that without departing from the scope of the invention, different variants can be deduced from the examples explained hereinbefore.

What we claim is:

**1.** A rotary pump comprising a housing, a cylindrical casing member mounted in said housing, a driving shaft rotatably mounted in said cylindrical casing member and having a concentric suction channel therein and a discharge channel on the outer peripheral surface thereof, a rotor fixed on said driving shaft eccentric thereto to form a chamber between said rotor and cylindrical casing member, said rotor being provided with two dimetrically opposite outer surfaces concentric with said shaft but with different radii, and first and second cam surfaces which are located between said opposite outer surfaces and have openings in communication with said suction and discharge channels respectively, and a plurality of abutments loosely received in radial grooves disposed with uniform circumferential spacing in the inner peripheral surface of said casing member, each of said abutments being resiliently pressed against the outer peripheral surface of said rotor.

**2.** A rotary pump as claimed in claim **1,** wherein each of said abutments is provided with a notch to apply an additional sealing pressure against the outer peripheral surface of said rotor by the aid of back pressure due to the discharge pressure fluid led into the back side of said abutment therethrough.

**3.** A rotary pump as claimed in claim **1,** wherein said first cam surface is provided with a first recess for connecting a portion of said chamber corresponding to said first cam surface with said suction channel, and said second cam surface is provided with a second recess for connecting a portion of said chamber corresponding to said second cam surface with said discharge channel, whereby sealing actions are substantially effected by said abutments corresponding to said opposite outer surfaces.

**4.** A rotary pump comprising a housing, a cylindrical casing member mounted in said housing, a driving shaft rotatably mounted in said cylindrical casing member and having a concentric suction channel therein and a discharge channel on the outer peripheral surface thereof, a rotor fixed on said driving shaft eccentric thereto to form a chamber between said rotor and cylindrical casing member, said rotor being provided with two diametrically opposite outer surfaces concentric with said shaft but with different radii, and first and second cam surfaces which are located between said opposite outer surfaces and have openings in communication with said suction and discharge channels respectively, and a plurality of abutments loosely received in radial grooves disposed with uniform circumferential spacing in the inner peripheral surface of said casing member, each of said abutments being resiliently pressed against the outer peripheral surface of said rotor and provided with a notch to apply an additional sealing pressure against the outer peripheral surface of said rotor by back pressure due to the discharge pressure fluid led into the back side of said abutment therethrough, said first cam surface being provided with a first recess for connecting a portion of said chamber corresponding to said first cam surface with said suction channel, said second cam surface being provided with a second recess for connecting a portion of said chamber corresponding to said second cam surface with said discharge channel, whereby sealing actions are substantially effected by said abutments corresponding to said opposite outer surfaces.

**5.** A rotary pump as claimed in claim **4,** wherein said rotor is further provided with a first convergent groove extending from said first recess towards one of said opposite outer surfaces and a second convergent groove extending from said second recess towards the other of said opposite outer surfaces.

**6.** A rotary pump comprising a housing, a cylindrical casing member mounted in said housing, a driving shaft rotatably mounted in said cylindrical casing member and having a concentric suction channel therein, a rotor integral with said driving shaft and eccentric thereto to form a chamber between said rotor and cylindrical casing member, said rotor being provided with two diametrically opposite outer surfaces concentric with said shaft but with different radii, and first and second cam surfaces located between said opposite outer surfaces, a discharge channel disposed in parallel with said driving shaft in said rotor, a plurality of abutments loosely received in radial grooves disposed with uniform circumferential spacing in the inner peripheral surface of said casing member, said abutments being resiliently pressed against the outer peripheral surface of said rotor, said first cam surface being provided with a first recess for sequentially and periodically connecting spaces between said abutments with said suction channel, said second cam surface being provided with a second recess for sequentially and periodically connecting said spaces between said abutments with said discharge channel, and a spool valve member slidably mounted in said rotor to control fluid flow through a by-pass passage from said discharge channel to said suction channel in response to the centrifugal action produced by the rotation of said rotor.

**7.** A rotary pump as claimed in claim **6,** wherein each of said abutments is provided with a notch to apply an additional sealing pressure against the outer peripheral surface of said rotor by the aid of back pressure due to the discharge pressure fluid led into the back side of said abutment therethrough.

**8.** A rotary pump comprising a housing, a driving shaft rotatably mounted in said housing, a rotor fixed on said driving shaft eccentric thereto to form a chamber between said rotor and housing, said rotor being provided with two diametrically opposite outer surfaces concentric with said shaft but with different radii, first and second cam surfaces located between said opposite outer surfaces, and a suction and a discharge channel disposed in said rotor in parallel relationship with said shaft, a plurality of abutments loosely received in radial grooves which are disposed with uniform circumferential spacing in the inner peripheral surface of said housing, said abutments being resiliently pressed against the outer peripheral surface of said rotor, said first cam surface being provided with a first recess for sequentially and periodically connecting spaces between said abutments with said suction channel, said second cam surface being provided with a second recess for sequentially and periodically connecting said spaces between said abutments with said discharge channel.

**9.** A rotary pump as claimed in claim **8,** wherein each of said abutments is provided with a notch to apply an

additional sealing pressure against the outer peripheral surface of said rotor by the aid of back pressure due to the discharge pressure fluid led into the back side of said abutment therethrough.


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