(19) Europäisches Patentamt
European Patent Office
Office européen des brevets

(11) EP 0 797 000 A1

(12) **EUROPEAN PATENT APPLICATION**

(43) Date of publication:
24.09.1997 Bulletin 1997/39

(51) Int Cl.[6]: **F04B 49/22**, F25B 41/04, F04B 49/03

(21) Application number: **97301697.5**

(22) Date of filing: **13.03.1997**

(84) Designated Contracting States:
DE FR GB IT SE

(30) Priority: **21.03.1996 JP 64448/96**

(71) Applicant: **SANDEN CORPORATION**
Isesaki-shi Gunma, 372 (JP)

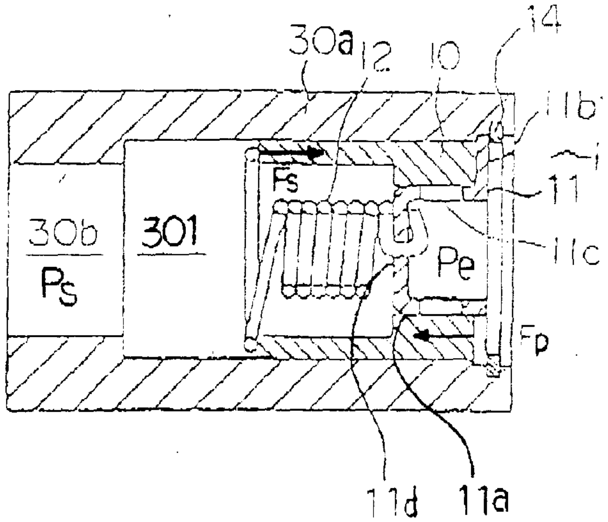
(72) Inventor: **Fujita, Masaaki,**
c/o Sanden Corporation
Isesaki-shi, Gunma 372 (JP)

(74) Representative: **Jackson, Peter Arthur**
GILL JENNINGS & EVERY
Broadgate House
7 Eldon Street
London EC2M 7LH (GB)

(54) **Starting load reducing device for refrigerant compressor**

(57) A device for reducing the starting load which occurs when starting up the compressor for automotive air conditioning system, is disclosed. The starting load reducing device is composed of a piston, a cap like member, and a tension coil spring made of shape-memory alloy disposed between them. In the actual refrigerant circuit, the starting load reducing device may be disposed anywhere between the outlet of evaporator and inlet port of compressor. When starting up the compressor, the flow of refrigerant gas from evaporator to suction chamber of the compressor is restricted to minimum value by the device. So the starting load of the compressor is suppressed effectively. After some time has lapsed, the temperature of refrigerant from the outlet of evaporator will gradually decreases to the transformation temperature of coil spring made of shape-memory alloy, due to small amount of continued operation of compressor. Temperature decrease in the refrigerant in the device will cause the martensitic transformation of the coil spring to occur, so that at this moment a full communication between the outlet of the evaporator and the suction chamber is realized . And stationary operation of the compressor enters with this elastically soft state of coil spring in the starting load reducing device. Thus, the purpose of eliminating an uncomfortable drive feeling due to the torque shock by suppressing the starting load of a compressor is attained.

Fig. 4



EP 0 797 000 A1

Description

The present invention relates to an automotive air conditioning system generally, and more particularly, to a starting load reducing device for a refrigerant compressor.

A refrigerant circuit used in an automotive air conditioning system such as shown in Figure 1, is well known generally. With reference to Figure 1, this refrigerant circuit includes a compressor 3, a condenser 4, receiver-dryer 5, an expansion valve 6, and an evaporator 7 connected with each other in series. In this refrigerant circuit, if the compressor is started when the thermal load for the air conditioning system is large, i. e., when the refrigerant temperature at the outlet of the evaporator is high, the compressor is required to perform a compression of large amount of refrigerant immediately after the compressor is powered on. As a result, not a little amount of energy of the drive source, for example an automotive engine, is consumed for the operation of the compressor 3 immediately after it is powered on. In consequence of this, a phenomenon that the rotational frequency of the drive shaft of an engine falls down sharply and immediately after the compressor was powered on, occurs. Since this unexpected and sharp fall down of rotational frequency of drive shaft (what is called torque shock) gives the car driver an uncomfortable drive feeling, it has been desired for a long time to eliminate or alleviate this torque shock.

In response to this requirement, U.S. Patent No. 4,905,477 is proposing a starting load reducing device equipped within a portion of a compressor. However, because the starting load reducing device according to that invention includes a portion in which a piston member is slid within a cylinder potion by the action of pressure in discharge chamber(high pressure), the practice of the invention requires a high precision manufacturing process that must make both of tight sealing and low friction of the sliding potion compatible simultaneously.

It is a primary object of this invention to provide an automotive air conditioning system having a starting load reducing device of which production does not require a high precision manufacturing process.

In the accompanying drawings:-

Figure 1 is a general refrigerant circuit used for an automotive air conditioning system.

Figure 2 is a longitudinal cross sectional view of a variable displacement swash plate type compressor equipped with a starting load reducing device, according to one embodiment of the present invention.

Figure 3 is a magnified transversal cross sectional view of the starting load reducing device shown in Figure 2.

Figure 4 is a cross sectional view taken along line IV-IV' in Figure 3.

Figure 5 is a cross sectional view taken along line IV-IV' in Figure 3, being especially for explaining the operation of the starting load reducing device shown in Figure 2.

With reference to Figure 2, Figure 3 and Figure 4, one embodiment of the present invention will be explained. With reference to Figure 2, Compressor 3 includes a closed housing assembly formed by a cylindrical compressor housing 50, front end plate 60 and a rear end plate in the form of cylinder head 30. Cylinder block 70 and crank chamber 80 are located in compressor housing 50. Front end plate 60 is attached to one end surface of compressor housing 50, and cylinder head 30 is disposed on the opposite end surface of compressor housing 50 and is fixedly mounted on one end surface of cylinder block 70 through a valve plate 90. Opening is formed in the central portion of front end plate 60 to receive a drive shaft 100.

Drive shaft 100 is rotatably supported in front end plate 60 through bearing 110. An inner end portion of drive shaft extends into central bore formed in the central portion of cylinder block 70, and is also rotatably supported therein by a bearing 120. A rotor 130, disposed in the interior of crank chamber 80, is connected to drive shaft 100 to be rotatable therewith, and engages an inclined plate 140 through a hinge mechanism 150. Wobble plate 160 is disposed on the opposite side surface of inclined plate 140 and bears against inclined plate through a bearing 170.

Hinge mechanism 150 comprises pin portion 135, formed on the inner end surface of rotor 130, and tab portion 145, having longitudinal hole 145h, formed on one end surface of inclined plate 140. The angle of inclination of inclined plate 140 with respect to drive shaft 100 can be adjusted by hinge mechanism 150.

A plurality of equiangularly spaced cylinder 180 are formed in cylinder block 70, and a piston 190 is reciprocatingly disposed within each cylinder 180. Each piston 190 is connected to wobble plate 160 through a connecting rod 200, i.e., one end of each connecting rod 200 is connected to wobble plate 160 with a ball joint and the other end of each connecting rod is connected to one of pistons 190 with a ball joint. A guide bar 210 extends within crank chamber of compressor housing 50. The lower end portion of wobble plate 160 engages guide bar 210 to enable wobble plate 160 to reciprocate along the guide bar while preventing rotational motion.

Pistons 190 are thus reciprocated in cylinders 180 by a drive mechanism formed of drive shaft 100, rotor 130, inclined plate 140, wobble plate 160 and connecting rods 200. Drive shaft 100 and rotor 130 are rotated and inclined plate 140, wobble plate 160 and connecting rods 200 function as a coupling mechanism to convert the rotational motion of the rotor into reciprocating motion of the pistons 190.

Cylinder head 30 is provided with a suction chamber 30b and a discharge chamber 30c, which communicate with cylinder 180 through suction hole 90a and discharge hole 90b, respectively, formed through valve plate 90. Furthermore, on cylinder head 30, there are formed an inlet port 30a and an outlet port (not shown in Figure 2) which place suction chamber 30b and discharge chamber 30c in fluid communication with an external refrigerant circuit respectively.

The capacity of the compressor is controlled by control valve 101 which regulates the pressure in the crank chamber 80. In Figure 2, Z is the inclination angle of the inclined plate 140 with respect to a plane perpendicular to the drive shaft 100. The control valve 101 functions so as to increase the inclination angle Z of the inclined plate 140 with respect to a plane perpendicular to the axis of drive shaft when the thermal load of the refrigerant circuit is high, i.e., when the refrigerant temperature at the outlet of the evaporator is high, and on the contrary, to decrease the inclination angle Z when the thermal load is low, i.e., when the refrigerant temperature is low.

Now, with reference to Figure 2 through Figure 5, the starting load reducing device 1 will be explained in detail.

Inlet port 30a is formed within the cylinder head 30, having cylindrical vacancy 301 in its interior. The central axis of the inlet port 30a is approximately parallel with the central axis of the drive shaft 100. In the cylindrical vacancy 301 is disposed a piston 10, being capable of reciprocating motion in the axial direction.

Cap like member 11 is disposed in the proximity of rear portion of the inlet port 30a (right side in Figure 4). The cap like member 11 has a plurality of rectangular projections 11b, and is fixedly disposed within the inlet port 30a by having the rectangular projections 11b fixed on the interior wall near rear portion of the inlet port 30a by a snap ring 14. The outer diameter of the lateral wall 11a of the cap like member 11 is designed to be slightly less than the inner diameter of the piston 10, so the lateral wall 11a of the cap like member 11 is accepted within the rear part of the piston 10 (right side in Figure 4). Furthermore, there are formed a plurality of holes 11c in the lateral wall 11a.

In the interior of the piston 10, is accommodated a shape memory alloy-made tension coil spring 12, one end of which is hooked on top portion 11d of the cap like member 11(right in Figure 4), and the other end of which is set on the other end surface of the piston 10. Here the tension coil spring 12 is suspended between the piston 10 and the cap like member 11 so that an attracting force Fs is always exerting on both. Further more, the transformation temperature Tc of the shape memory alloy used for the tension coil spring 12 is selected from between 10 Centi degrees Celsius to 30 Centi degrees Celsius, where 30 Centi degrees Celsius is representative temperature of refrigerant passing through the inlet port 30a corresponding to rest condition of the compressor, and 10 Centi degrees Celsius is the representative temperature corresponding to normal running condition of compressor. Because the tension coil spring 12 is made of shape memory alloy, the spring constant varies rather drastically upon its transformation temperature Tc. When the refrigerant temperature which can be regarded as the same temperature of the coil spring is higher than the transformation temperature Tc, the spring constant is large, and on the contrary, when the refrigerant temperature is less than the transformation temperature Tc the spring constant is small. In other words, if the refrigerant temperature decreases from a high value greater than the transformation temperature Tc, to a small value less than the transformation temperature Tc, then the shape memory alloy-made tension coil spring 12 changes its elastic property sharply upon the transformation temperature Tc, that is, changes from a elastically stiff spring to a elastically soft spring.

Thus the starting load reducing device 1 is composed of inlet port 30a, piston 10, cap like member 11, and tension coil spring 12.

Here, in a production line other than the main line for compressor, a semi-completed assembly can be made from piston 10, cap like member 11, and tension coil spring 12.

With reference to Figure 4 and Figure 5, the operation of the starting load reducing device 1 thus composed is explained as follows. When the thermal load of the air conditioning system is large, that is, when the refrigerant temperature at the outlet of evaporator is high, then the refrigerant temperature in the inlet port 30a is higher than the transformation temperature Tc. Accordingly the spring constant k of the tension coil spring is large, and also the attracting force Fs is large. With reference to Figure 2 and Figure 4, due to this attracting force exerted by the tension coil spring 12, the piston 10 is pulled to right direction in the figure, so that the right side end surface of the piston 10 is impinging upon the rectangular projections 11b. In this state, the lateral wall 11a of the cap like member 11 is fully accommodated within the rear part of the piston 10. Therefore, the communication between the suction chamber 30b of the compressor 3 and the outlet of evaporator is blocked substantially when the compressor is in a rest state.

When an operation of the compressor 3 is started by a control mechanism of the automotive air conditioning system from this state, since the communication between the suction chamber 30b of the compressor 3 and the outlet of the evaporator is blocked substantially, only the remaining refrigerant in the suction chamber 30b is sucked into the cylinder 180 to be compressed. So that, the pressure Ps in the suction chamber decreases down, to be lower than the refrigerant pressure Pe at the outlet of the evaporator. Then there occurs a pressure difference (Pe-Ps) accompanying a force Fp that act to push the piston 10 into the left direction in the Figure 4. As the suction of the residual refrigerant from the suction chamber into the cylinder 180 is kept on, the pressure difference(Pe-Ps) increases, and when a force Fp that act to move the piston 10 to the left direction in Figure 4 surmounts the attracting force Fs of the tension coil spring 12, i.e.,when Fp>Fs, then the piston 10 moves to the left direction in the Figure 4. Because of this displacement of the piston 10, the surface contact between the outer surface of the lateral wall 11a and the inner surface of piston 10 is removed as shown in Figure 5. As a result, the suction chamber 30b becomes capable of communicating with the outlet of the evaporator, so that the refrigerant starts to flow fully from the outlet of the evaporator into the suction chamber 30b.

When the suction chamber 30b becomes communicating with the outlet of the evaporator, and when the refrigerant starts to flow fully from the outlet of the evaporator into the suction chamber 30b, the pressure difference between the pressures in the outlet of the evaporator and the suction chamber vanishes, i.e.,Pe-Ps=0. However, since the compressor 3 is already in its normal running condition in this state, the refrigerant temperature flowing from the outlet of the evaporator into the suction chamber 30b becomes lower than the transformation temperature Tc of the coil spring 12. Accordingly the spring constant k of the tension coil spring 12 changes from a large value to small value, so that the attracting force Fs of the coil spring 12 becomes small. Therefore, the piston 10 is maintained at the position shown in Figure 5 by a force Fd which occurs due to the dynamic pressure of refrigerant flow from the outlet of the evaporator into the suction chamber 30b. In this way, in the normal running condition, full communication between the outlet of the evaporator and the suction chamber 30b is maintained to enable the refrigerant to flow normally from the outlet of the evaporator into the suction chamber 30b.

When the operation of the compressor 3 is stopped by the control mechanism of the automotive air conditioning system, the refrigerant temperature in the inlet port 30a rises. And when the refrigerant temperature becomes over transformation temperature Tc of the tension coil spring 12, the spring constant k of the tension coil spring 12 changes back to the original large value, and the attracting force Fs of the tension coil spring 12 also resumes its large value. This attracting force Fs of the tension coil spring 12 causes the piston 10 to move rightward, and the end surface of the piston 10 impinges on the rectangular projections 11b of the cap like member 11. And the lateral wall 11a of the cap like member 11 is accommodated again in the rear part of the piston 10(right in Figure 2 and Figure 4). As a result, the communication between the suction chamber 30b and the outlet of the evaporator is again blocked substantially.

Furthermore, if the compressor 3 is started by a control mechanism of the automotive air conditioning system when the thermal load of the air conditioning system is small, i.e., when the refrigerant temperature at the outlet of the evaporator is low, the pressure difference (Pe-Ps) between the outlet of the evaporator and the suction chamber 30b does not grow to a large value. However, since the refrigerant temperature in the suction port 30a is lower than the transformation temperature Tc, the spring constant k is small. Therefore the tension coil spring 12 is in a state in which the attracting force Fs is small before the compressor is started up. Therefore, the piston 10 moves leftward with ease, and the surface contact between the outer surface of the lateral wall 11a of the cap like member 11 and inner surface of the piston 10 is removed as shown in Figure 5. Thus, immediately after the compressor is started, the suction chamber 30b of the compressor 3 can communicate with the outlet of the evaporator, and the refrigerant can flow from the outlet of the evaporator into the suction chamber 30b. In other words, the function of restricting the amount of refrigerant sucked into the cylinder 180 does not operate immediately after the compressor 3 is started in this case. But in this circumstance, even though the function of restricting the amount of refrigerant sucked into the cylinder 180 does not operate, there is no problem because the thermal load for the air conditioning system is small. That is, since the thermal load for the air conditioning system is small from the outset, the inclination angle Z of the inclined plate 140 of the compressor 3 becomes minimum almost instantaneously after the compressor is started by the action of the control valve 101. So in this case of low thermal load, the compressor sucks and compresses little refrigerant initially. Therefore the torque shock does not occur in this case.

As described above, according to the starting load reducing device of the present invention, when the compressor is started under high thermal load condition for the air conditioning system, the starting load reducing device acts so as to throttle the amount of refrigerant which is compressed immediately after the compressor is started. As a consequence of this function, large amount of energy is not consumed in the starting operation of the compressor, so that the phenomenon that the rotational frequency of the drive shaft of the engine falls down immediately after the starting of the compressor(what is called a torque shock), can be eliminated. Hence an uncomfortable drive feeling which will be given to the car driver otherwise is also eliminated.

Moreover, since the device attains the above purpose by controlling the communication between the outlet of the evaporator and the suction chamber which are both low pressure, it does not require a high precision manufacturing process that must make both of low friction and tight sealing of the sliding portion of communication control mechanism compatible simultaneously.

Furthermore, according to the starting load reducing device of the present invention, it is possible to assemble a semi-completed product composed of piston 10, cap like member 11, and tension coil spring 12 in a production line other than the main production line for the compressor 3, so that a freedom of production control increases.

Though in the above embodiment of the present invention the suction port 30a formed within the cylinder head 30 is used as a casing for the starting load reducing device 1, the casing may be provided separately and disposed anywhere between the outlet of the evaporator and the suction chamber 30b of the compressor 3.

Claims

1. In a refrigerant circuit having a compressor, a condenser, an evaporator and starting load reducing means connected to each other in series, the improvement comprising:

said starting load reducing means disposed between an outlet of said evaporator and a suction chamber of said compressor, that operates to control the communication therebetween responsive to the temperature of refrigerant gas coming from the evaporator, wherein said starting load reducing means operate to restrict the communication responsive to high refrigerant temperature corresponding to the rest state of the compressor and operates to allow full communication responsive to low refrigerant temperature corresponding to normal running state of compressor.

2. The refrigerant circuit of claim 1 wherein said starting load reducing means comprises a cylindrical piston-throttle mechanism including a cylindrical piston which can slide in the direction of refrigerant flow, cap like member which has auxiliary holes and receives said piston, and a coil spring which is pulling said piston and said cap like member together.

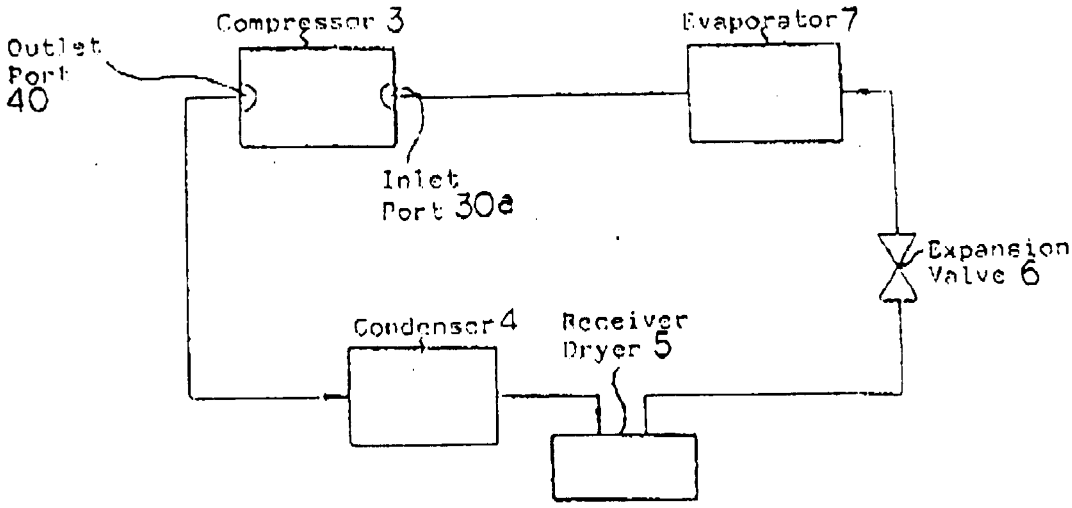
3. The refrigerant circuit of claim 1 and claim 2, wherein said starting load reducing means has said coil spring made of shape memory alloy of which transformation temperature lies between the refrigerant temperature corresponding to stationary operating state of compressor and the refrigerant temperature corresponding to the rest state of compressor.

4. The refrigerant circuit of claim 1 and claim 2 and claim 3, wherein said starting load reducing means is accommodated within the inlet port of compressor.

5. The refrigerant circuit of claim 1 and claim 2 and claim 3, wherein said starting load reducing means is accommodated within the outlet port of evaporator.

6. The refrigerant circuit of claim 1 and claim 2 and claim 3, wherein said starting load reducing means is accommodated within an separate casing which is disposed between outlet of evaporator and inlet port of compressor.

Fig. 1


Compressor 3
Evaporator 7
Outlet Port 40
Inlet Port 30a
Expansion Valve 6
Condenser 4
Receiver Dryer 5

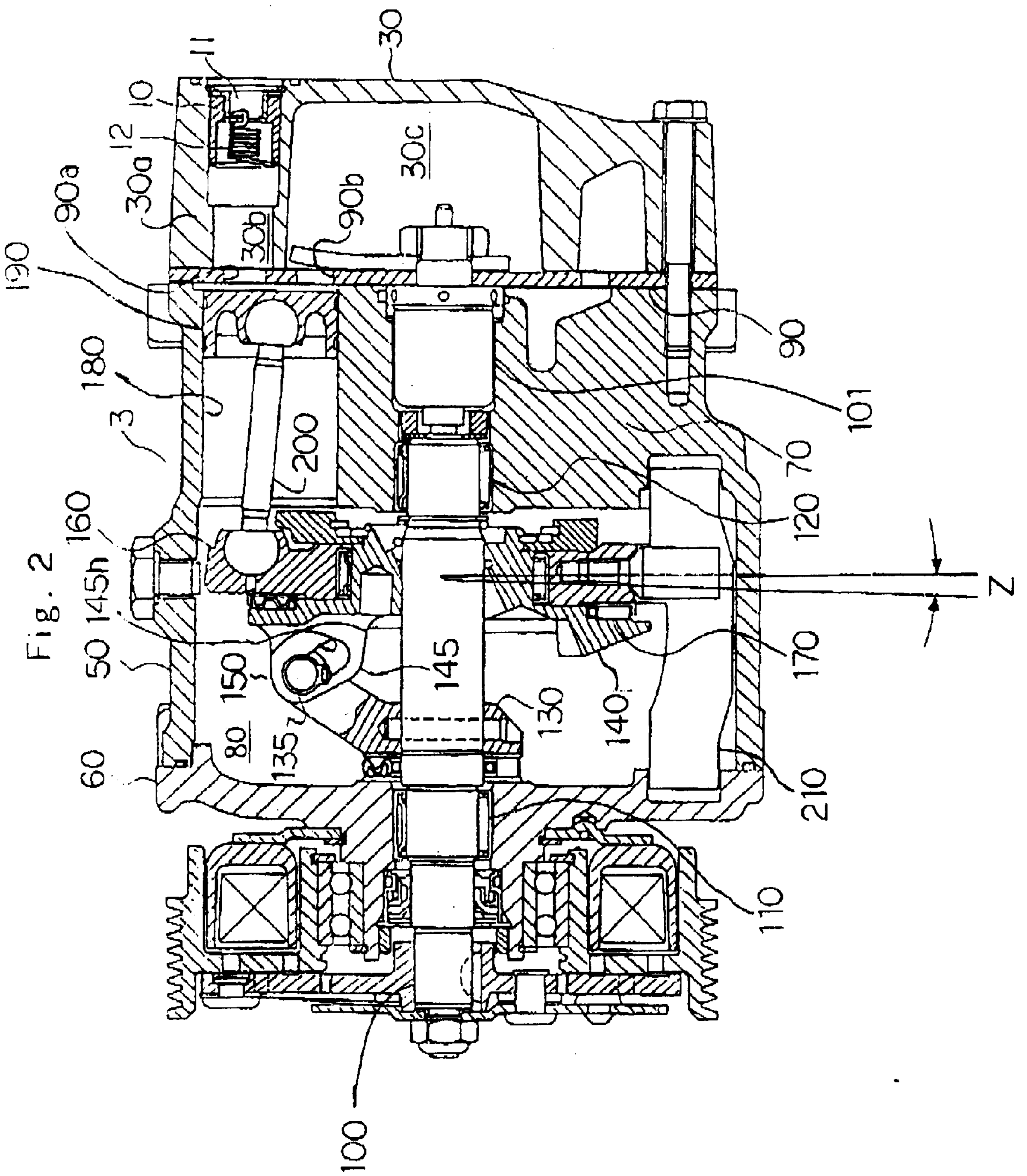

30
11
10
12
90a
30a
30c
90b
30b
190
180
90
101
3
200
70
120
160
145h
Z
50
150
145
170
140
130
80
135
60
210
110
100

Fig. 2

Fig. 3

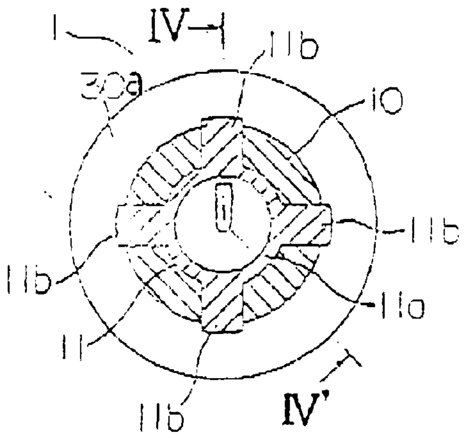


Fig. 4

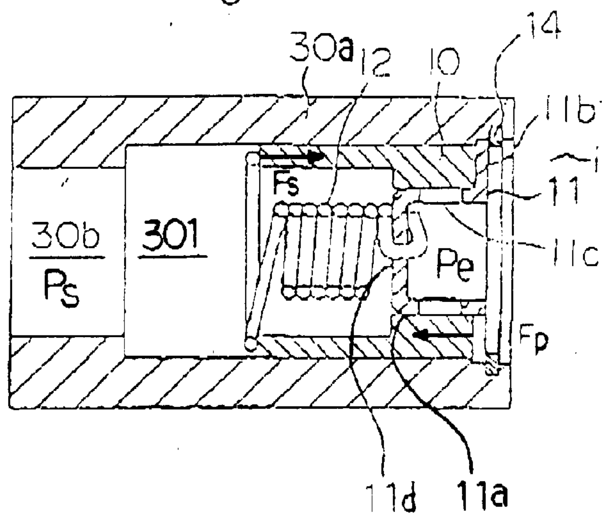
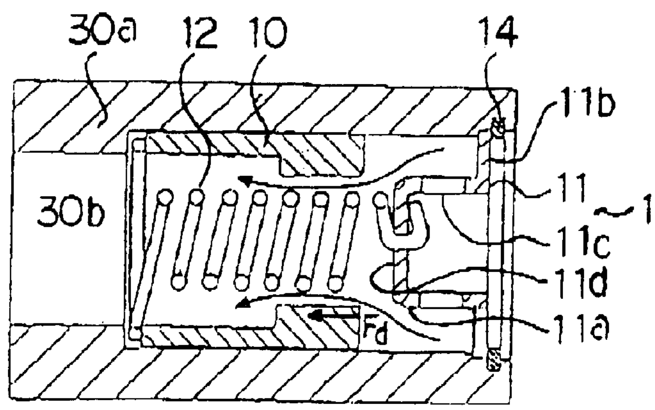


Fig. 5



European Patent Office

EUROPEAN SEARCH REPORT

Application Number
EP 97 30 1697

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int.Cl.6)
A,D	US 4 905 477 A (KAZUHIKO TAKAI) 6 March 1990 * column 3, line 65 - column 4, line 60; figures 1-3 * ---	1	F04B49/22 F25B41/04 F04B49/03
A	US 4 330 999 A (NAKAYAMA SHOZO) 25 May 1982 * column 7, line 12 - line 41; figure 7 * ---	1	
A	US 3 785 554 A (PROCTOR R) 15 January 1974 * column 3, line 63 - column 4, line 46; figure 1 * ---	1	
A	US 2 366 188 A (GIBSON) * page 2, left-hand column, line 50 - page 3, left-hand column, line 9; figure 1 * -----	1,2	
			TECHNICAL FIELDS SEARCHED (Int.Cl.6)
			F04B F25B

The present search report has been drawn up for all claims

Place of search	Date of completion of the search	Examiner
THE HAGUE	1 July 1997	Bertrand, G

CATEGORY OF CITED DOCUMENTS

X : particularly relevant if taken alone
Y : particularly relevant if combined with another document of the same category
A : technological background
O : non-written disclosure
P : intermediate document

T : theory or principle underlying the invention
E : earlier patent document, but published on, or after the filing date
D : document cited in the application
L : document cited for other reasons

& : member of the same patent family, corresponding document

EPO FORM 1503 03.82 (P04C01)