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

Vorrichtung zur Reduzierung der Startlast für Kältemittelverdichter

Dispositif de réduction de la charge de démarrage pour compresseur de réfrigérant

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Description

[0001] The present invention relates to an automotive air conditioning system generally, and more particularly, to a starting load reducing device for a refrigerant compressor.

[0002] 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.

[0003] In response to this requirement, US Patent No. 4905477 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 portion 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 portion compatible simultaneously.

[0004] 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.

[0005] According to the present invention there is provided a refrigerant circuit having a compressor, a condenser, an evaporator and starting load reducing means connected to each other in series, wherein:

said starting load reducing means is 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, characterised in that:

said starting load reducing means operates 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.

[0006] In the accompanying drawings:-

[0007] Figure 1 is a general refrigerant circuit used for an automotive air conditioning system.

[0008] 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.

[0009] Figure 3 is a magnified transversal cross sectional view of the starting load reducing device shown in Figure 2.

[0010] Figure 4 is a cross sectional view taken along line IV-IV' in Figure 3.

[0011] 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.

[0012] 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.

[0013] 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.

[0014] 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.

[0015] 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.

[0016] 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.

[0017] 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.

[0018] 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.

[0019] Now, with reference to Figure 2 through Figure 5, the starting load reducing device 1 will be explained in detail.

[0020] 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.

[0021] 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.

[0022] 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 lid 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 F_s is always exerting on both. Furthermore, the transformation temperature T_c 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 T_c . When the refrigerant temperature which can be regarded as the same temperature of the coil spring is higher than the transformation temperature T_c , the spring constant is large, and on the contrary, when the refrigerant temperature is less than the transformation temperature T_c the spring constant is small. In other words, if the refrigerant temperature decreases from a high value greater than the transformation temperature T_c , to a small value less than the transformation temperature T_c , then the shape memory alloy-made tension coil spring 12 changes its elastic property sharply upon the transformation temperature T_c , that is, changes from an elastically stiff spring to an elastically soft spring.

[0023] Thus the starting load reducing device 1 is composed of inlet port 30a, piston 10, cap like member 11, and tension coil spring 12.

[0024] 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.

[0025] 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 T_c . Accordingly the spring constant k of the tension coil spring is large, and also the attracting force F_s 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.

[0026] 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 P_s in the suction chamber decreases down, to be lower than the refrigerant pressure P_e at the outlet of the evaporator. Then there occurs a pressure difference ($P_e - P_s$) accompanying a force F_p 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 ($P_e - P_s$) increases, and when a force F_p that act to move the piston 10 to the left direction in Figure 4 surmounts the attracting force F_s of the tension coil spring 12, i.e., when $F_p > F_s$, 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.

[0027] 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., $P_e - P_s = 0$. 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 T_c 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 F_s of the coil spring 12 becomes small. Therefore, the piston 10 is maintained at the position shown in Figure 5 by a force F_d 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.

[0028] 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 T_c of the tension coil spring 12, the spring constant k of the ten-

sion coil spring 12 changes back to the original large value, and the attracting force F_s of the tension coil spring 12 also resumes its large value. This attracting force F_s 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.

[0029] 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 ($P_e - P_s$) 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 T_c , the spring constant k is small. Therefore the tension coil spring 12 is in a state in which the attracting force F_s 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

[0030] 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.

[0031] 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.

[0032] 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.

[0033] 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. A refrigerant circuit having a compressor (3), a condenser (4), an evaporator (7) and starting load reducing means (1) connected to each other in series, wherein:

said starting load reducing means is disposed between an outlet of said evaporator and a suction chamber (30b) of said compressor, that operates to control the communication therebetween responsive to the temperature of refrigerant gas coming from the evaporator, characterised in that:

said starting load reducing means operates to restrict the communication responsive to high refrigerant temperature corresponding to the rest state of the compressor (3) 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 (1) comprises a cylindrical piston-throttle mechanism including a cylindrical piston (10) which can slide in the direction of refrigerant flow, a cap like member (11) which has auxiliary holes and receives said piston, and a coil spring (12) which is pulling said piston and said cap like member together.

3. The refrigerant circuit of claim 1 or claim 2, wherein said starting load reducing means (1) has said coil spring (12) made of shape memory alloy of which transformation temperature lies between the refrigerant temperature corresponding to stationary operating state of compressor (3) and the refrigerant temperature corresponding to the rest state of compressor.

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

5. The refrigerant circuit of claim 1, 2 or 3, wherein said starting load reducing means (1) is accommodated within the outlet port of evaporator (7).

6. The refrigerant circuit of claim 1, 2 or 3, wherein said starting load reducing means (1) is accommodated within a separate casing which is disposed between outlet of evaporator (7) and inlet port of compressor (3).

25 Patentansprüche

1. Kältemittelkreislauf mit einem Kompressor (3), einem Kondensator (4), einem Verdampfer (7) und einem Startlastverringermittel (1), die miteinander in Reihe verbunden sind, worin:

das Startlastverringermittel zwischen einem Auslaß des Verdampfers und einer Ansaugkammer (30b) des Kompressors vorgesehen ist, das zum Steuern der Verbindung dazwischen als Reaktion auf die Temperatur des von dem Verdampfer kommenden Kühlmittelgases steuert, dadurch gekennzeichnet,

daß das Startlastverringermittel zum Beschränken der Verbindung als Reaktion auf hohe Kältemitteltemperatur entsprechend dem Ruhezustand des Kompressors (3) tätig ist und zum Erlauben der vollen Verbindung als Reaktion auf niedrige Kältemitteltemperatur entsprechend dem normalen Laufzustand des Kompressors tätig ist.

2. Kältemittelkreislauf nach Anspruch 1, bei dem das Startlastverringermittel (1) einen zylindrischen Kolben-Drosselmechanismus, der einen zylindrischen Kolben (10) enthält, der in der Richtung des Kältemittelflusses gleiten kann, ein kappenartiges Teil (11), das Hilfslöcher aufweist und den Kolben aufnimmt, und eine Schraubenfeder (12), die den Kolben und das kappenartige Teil zusammenzieht, aufweist.

3. Kältemittelkreislauf nach Anspruch 1 oder Anspruch 2, bei dem das Startlastverringermittel (1) die Schraubenfeder (12) aufweist, die aus einer

Formgedächtnislegierung hergestellt ist, deren Transformationstemperatur zwischen der Kältemitteltemperatur entsprechend dem stationären Betriebszustand des Kompressors (3) und der Kältemitteltemperatur entsprechend dem Ruhezustand des Kompressors liegt.

4. Kältemittelkreislauf nach Anspruch 1, 2 oder 3, bei dem das Startlastverringermittel (1) innerhalb der Einlaßöffnung des Kompressors aufgenommen ist. 10
5. Kältemittelkreislauf nach Anspruch 1, 2 oder 3, bei dem das Startlastverringermittel (1) innerhalb der Auslaßöffnung des Verdampfers (7) aufgenommen ist. 15
6. Kältemittelkreislauf nach Anspruch 1, 2 oder 3, bei dem das Startlastverringermittel (1) innerhalb eines getrennten Gehäuses aufgenommen ist, das zwischen dem Auslaß des Verdampfers (7) und der Einlaßöffnung des Kompressors (3) vorgesehen ist. 20

Revendications

1. Circuit de réfrigérant comportant un compresseur (3), un condenseur (4), un évaporateur (7) et un dispositif de réduction de la charge de démarrage (1) connectés les uns aux autres en série, dans lequel : le dispositif de réduction de la charge de démarrage est disposé entre une sortie de l'évaporateur et une chambre d'aspiration (30b) du compresseur pour servir à commander la communication entre les deux en réponse à la température du gaz réfrigérant arrivant de l'évaporateur, caractérisé en ce que le dispositif de réduction de la charge de démarrage fonctionne pour limiter la communication en réponse à une température de réfrigérant élevée correspondant à l'état de repos du compresseur (3), et fonctionne pour permettre une communication complète en réponse à une basse température du réfrigérant correspondant à l'état de fonctionnement normal du compresseur. 30 35 40 45
2. Circuit de réfrigérant selon la revendication 1, dans lequel le dispositif de réduction de la charge de démarrage (1) comprend un mécanisme d'étranglement à piston cylindrique comprenant un piston cylindrique (10) pouvant glisser dans la direction de l'écoulement du réfrigérant, un élément en forme de capuchon (11) percé de trous auxiliaires et recevant le piston, et un ressort hélicoïdal (12) tirant le piston et l'élément en forme de capuchon l'un vers l'autre. 50 55

3. Circuit de réfrigérant selon la revendication 1 ou la revendication 2, dans lequel le ressort hélicoïdal (12) du dispositif de réduction de la charge de démarrage (1) est réalisé dans un alliage à mémoire de forme dont la température de transformation se situe entre la température du réfrigérant correspondant à un état de fonctionnement stable du compresseur (3), et la température du réfrigérant correspondant à l'état de repos du compresseur. 5
4. Circuit de réfrigérant selon la revendication 1, 2 ou 3, dans lequel le dispositif de réduction de la charge de démarrage (1) est logé dans le port d'entrée du compresseur. 15
5. Circuit de réfrigérant selon la revendication 1, 2 ou 3, dans lequel le dispositif de réduction de la charge de démarrage (1) est logé dans le port de sortie de l'évaporateur (7). 20 25
6. Circuit de réfrigérant selon la revendication 1, 2 ou 3, dans lequel le dispositif de réduction de la charge de démarrage (1) est logé dans un boîtier séparé qui est disposé entre la sortie de l'évaporateur (7) et le port d'entrée du compresseur (3). 30

Fig. 1

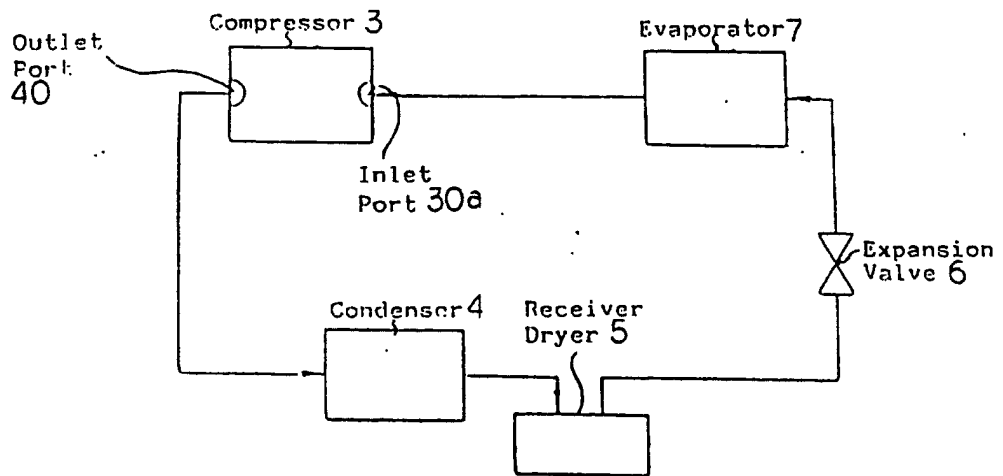


Fig. 3

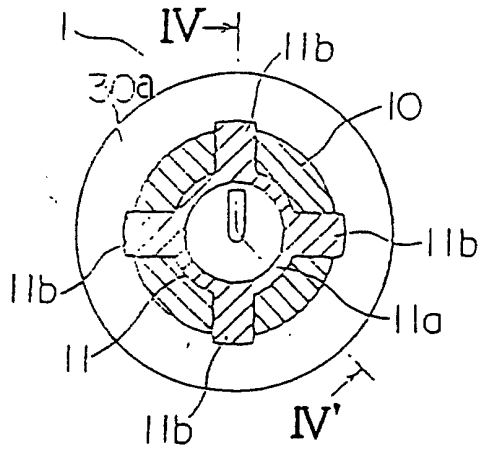


Fig. 4

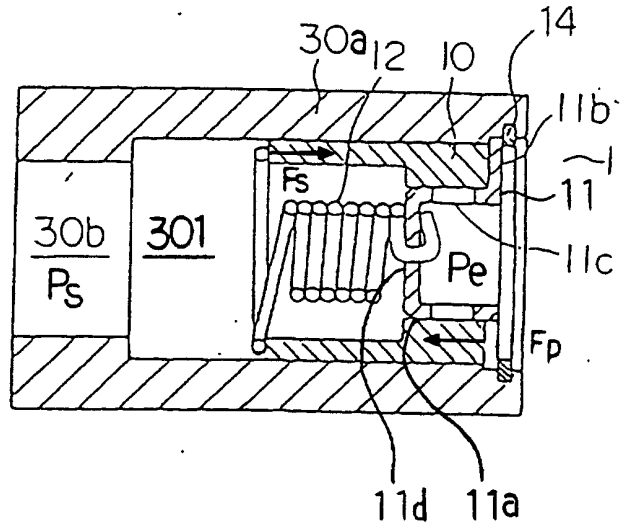


Fig. 5

