ABSTRACT OF THE DISCLOSURE

A tandem type vane compressor of the present invention has primary and secondary vanes and primary and secondary vane grooves. A primary backpressure chamber is defined between the bottom surface of each primary vane and the corresponding primary vane groove. A secondary backpressure chamber is defined between the bottom surface of each secondary vane and the corresponding secondary vane groove. The primary and secondary back pressure chambers are connected to a discharge chamber by first and second backpressure supplying mechanism in the compression process of first and second compression mechanisms, respectively. The first and the second backpressure supplying mechanisms include first and second rotation paths, which are formed in the drive shaft, and firs and second intermittent mechanisms, which selectively allow and interrupt communication of the first and second rotation paths with the discharge chamber and with each of the primary and secondary backpressure chambers according to the phase in the rotation direction of the drive shaft, respectively.

CLAIMS

1. A tandem type vane compressor comprising:

a housing that has a suction chamber, a discharge chamber, and a plurality of compression chambers, wherein the housing rotationally supports a drive shaft; and

a plurality of compression mechanisms including a first compression mechanism and a second compression mechanism, which are coupled to each other in tandem in the housing, wherein

the first and second compression mechanisms each have at least one of the compression chambers,

each compression mechanism is driven by rotation of the drive shaft to perform a suction process, in which each compression mechanism draws low-pressure refrigerant gas into the respective compression chamber from the suction chamber, a compression process, in which each compression mechanism compresses the refrigerant gas in the respective compression chamber, and a discharge process, in which each compression mechanism discharges the high-pressure refrigerant gas in the respective compression chamber to the discharge chamber,

the first compression mechanism includes a first cylinder chamber, which is formed in the housing, a first rotor, which is provided in the first cylinder chamber to rotate when the drive shaft rotates, the first rotor having a plurality of radially extending primary vane grooves, and a plurality of primary vanes, each of which is located in one of the primary vane grooves and is capable of projecting and retracting,

the compression chamber of the first compression mechanism is defined by an inner surface of the first cylinder chamber, an outer surface of the first rotor, and the primary vanes, wherein the compression chamber of the first compression mechanism is located at a position forward of the compression chamber of the second compression mechanism,

the second compression mechanism includes a second cylinder chamber, which is formed in the housing, a second rotor, which is provided in the second cylinder chamber to rotate when the drive shaft rotates, the second rotor having a plurality of radially extending secondary vane grooves, and a plurality of secondary vanes, each of which is located in one of the secondary vane grooves and is capable of projecting and retracting, and

the compression chamber of the second compression mechanism is defined by an inner surface of the second cylinder chamber, an outer surface of the second rotor, and the secondary vanes, the tandem type vane compressor being **characterized in that**

a bottom surface of each primary vane and the corresponding primary vane groove define a primary backpressure chamber,

a first backpressure supplying mechanism is provided, which, during the compression process of the first compression mechanism, allows each primary back pressure chamber to communicate with the discharge chamber,

the first backpressure supplying mechanism includes a first rotation path, which is formed in the drive shaft or in a rotator that rotates in synchronization with the drive shaft, and a first intermittent mechanism, which selectively allows and interrupts communication of the first rotation path with the discharge chamber and with each primary backpressure chamber according to the phase in the rotation direction of the drive shaft,

a bottom surface of each secondary vane and the corresponding secondary vane groove define a secondary backpressure chamber,

a second backpressure supplying mechanism is provided, which, during the compression process of the second compression mechanism, allows each secondary back pressure chamber to communicate with the discharge chamber, and

the second backpressure supplying mechanism includes a second rotation path, which is formed in the drive shaft or in the rotator, and a second intermittent mechanism, which selectively allows and interrupts communication of the second rotation path with the discharge chamber and with each secondary backpressure chamber according to the phase in the rotation direction of the drive shaft.

2. The tandem type vane compressor according to claim 1, wherein, while the first intermittent mechanism interrupts communication of the first rotation path with the discharge chamber and with the primary backpressure chambers, the second intermittent mechanism interrupts communication of the second rotation path with the discharge chamber and with the secondary back pressure chambers.

3. The tandem type vane compressor according to claim 1 or 2, wherein the housing includes:

a shell having a suction inlet and a discharge outlet, which are connected to the outside;

a first side plate, which is accommodated in the shell and defines, with the shell, the suction chamber such that the suction chamber communicates with the suction inlet;

a second side plate, which is accommodated in the shell and partitions the first compression mechanism and the second compression mechanism from each other;

a third side plate, which is accommodated in the shell and defines, with the shell, the discharge chamber such that the discharge chamber communicates with the discharge outlet;

a first cylinder block, which forms the first cylinder chamber while being held between the first side plate and the second side plate; and

a second cylinder block, which is accommodated in the shell while being held between the second plate and the third side plate, thereby forming the second cylinder chamber.

4. The tandem type vane compressor according to claim 3, wherein

the housing or the drive shaft has a common passage, which extends in a longitudinal direction of the drive shaft to communicate with the discharge chamber,

at least one of the first side plate and the second side plate has a first supplying passage, which connects the common passage with each primary backpressure chamber via the first intermittent mechanism, and

at least one of the second side plate and the third side plate has a second supplying passage, which connects the common passage with each secondary backpressure chamber via the second intermittent mechanism.

5. The tandem type vane compressor according to claim 1 or 2, wherein the drive shaft is one of a first drive shaft for driving the first compression mechanism and a second drive shaft for driving the second compression mechanism, the second drive shaft being fitted to the first drive shaft.

6. The tandem type vane compressor according to claim 3, wherein

the shell includes a front housing member, which has the suction inlet and defines the suction chamber with the first side plate, and a rear housing member, which has the discharge outlet and defines the discharge chamber with the third side plate,

the first cylinder block and the second cylinder are common components,

the first rotor and the second rotor are common components, and

the primary vanes and secondary vanes are common components.

7. A tandem type vane compressor comprising:

a housing that has a suction chamber, a discharge chamber, and a plurality of compression chambers, wherein the housing rotationally supports a drive shaft; and

a plurality of compression mechanisms including a first compression mechanism and a second compression mechanism, which are coupled to each other in tandem in the housing, wherein

the first and second compression mechanisms each have at least one of the compression chambers,

each compression mechanism is driven by rotation of the drive shaft to perform a suction process, in which each compression mechanism draws low-pressure refrigerant gas into the respective compression chamber from the suction chamber, a compression process, in which each compression mechanism compresses the refrigerant gas in the respective compression chamber, and a discharge process, in which each compression mechanism discharges the high-pressure refrigerant gas in the respective compression chamber to the discharge chamber,

the first compression mechanism includes a first cylinder chamber, which is formed in the housing, a first rotor, which is provided in the first cylinder chamber to rotate when the drive shaft rotates, the first rotor having a plurality of radially extending primary vane grooves, and a plurality of primary vanes, each of which is located in one of the primary vane grooves and is capable of projecting and retracting,

the compression chamber of the first compression mechanism is defined by an inner surface of the first cylinder chamber, an outer surface of the first rotor, and the primary vanes, wherein the compression chamber of the first compression mechanism is located at a position forward of the compression chamber of the second compression mechanism, the second compression mechanism includes a second cylinder chamber, which is formed in the housing, a second rotor, which is provided in the second cylinder chamber to rotate when the drive shaft rotates, wherein the second rotor has a plurality of radially extending secondary vane grooves and a plurality of secondary vanes, each of which is located in one of the secondary vane grooves and is capable of projecting and retracting, and

the compression chamber of the second compression mechanism is defined by an inner surface of the second cylinder chamber, an outer surface of the second rotor, and the secondary vanes, the tandem type vane compressor being **characterized in that**

a bottom surface of each primary vane and the corresponding primary vane groove define a primary backpressure chamber,

a first backpressure supplying mechanism is provided, which, during the compression process of the first compression mechanism, allows each primary back pressure chamber to communicate with the discharge chamber,

a bottom surface of each secondary vane and the corresponding secondary vane groove define a secondary backpressure chamber,

a second backpressure supplying mechanism is provided, which, during the compression process of the second compression mechanism, allows each secondary back pressure chamber to communicate with the discharge chamber, and

the first backpressure supplying mechanism or the second back pressure supplying mechanism includes a rotation path, which is formed in the drive shaft or in a rotator that rotates in synchronization with the drive shaft, and an intermittent mechanism, which selectively allows and interrupts communication of the rotation path with the discharge chamber and with each primary backpressure chamber or each secondary back pressure chamber according to the phase in the rotation direction of the drive shaft.

Dated this 27 day of March 2013.

Monica Batra

(Monica Batra Nagpal)
REG. No.: IN/PA-1298
of De Penning & De Penning
Agent For The Applicants

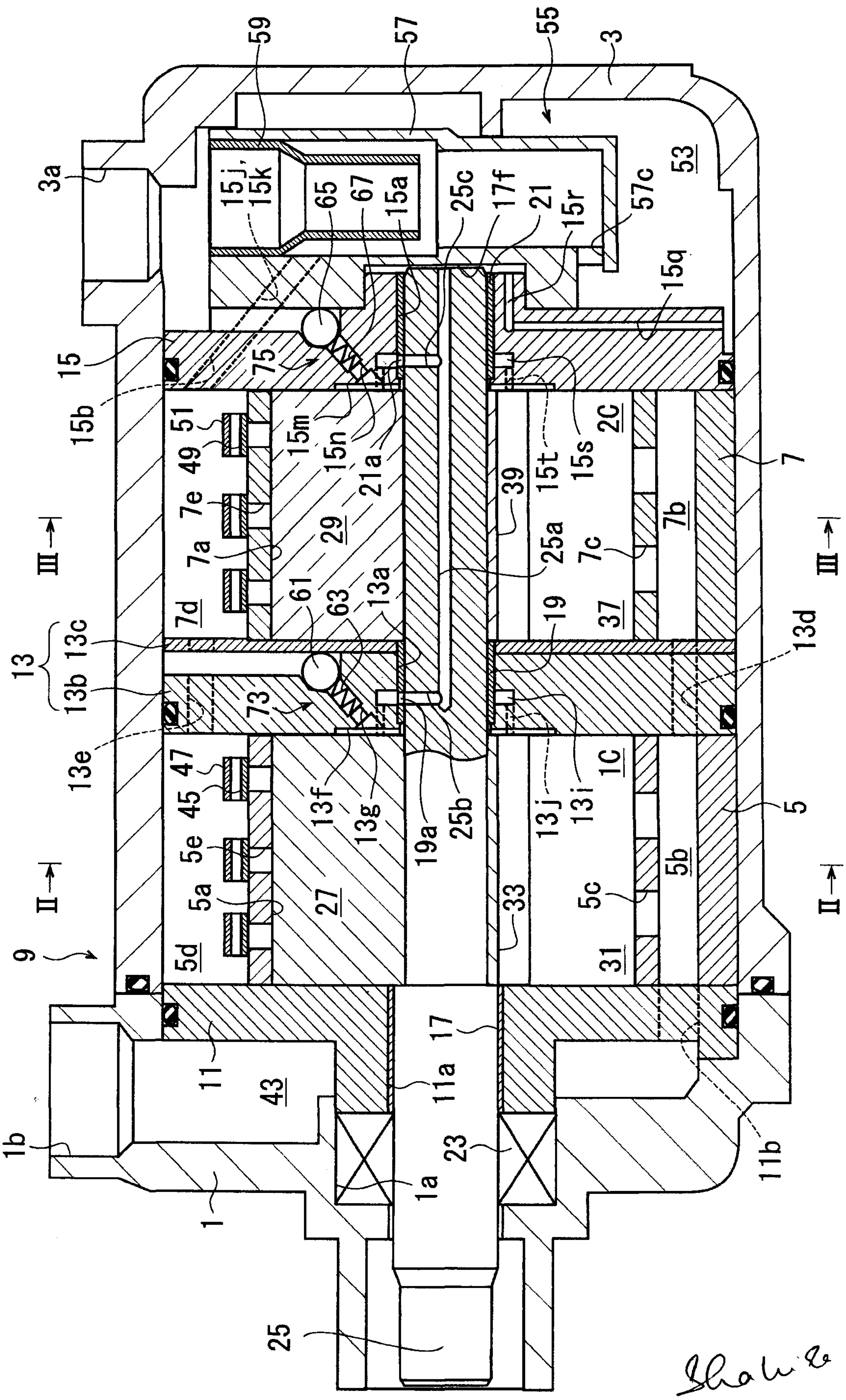
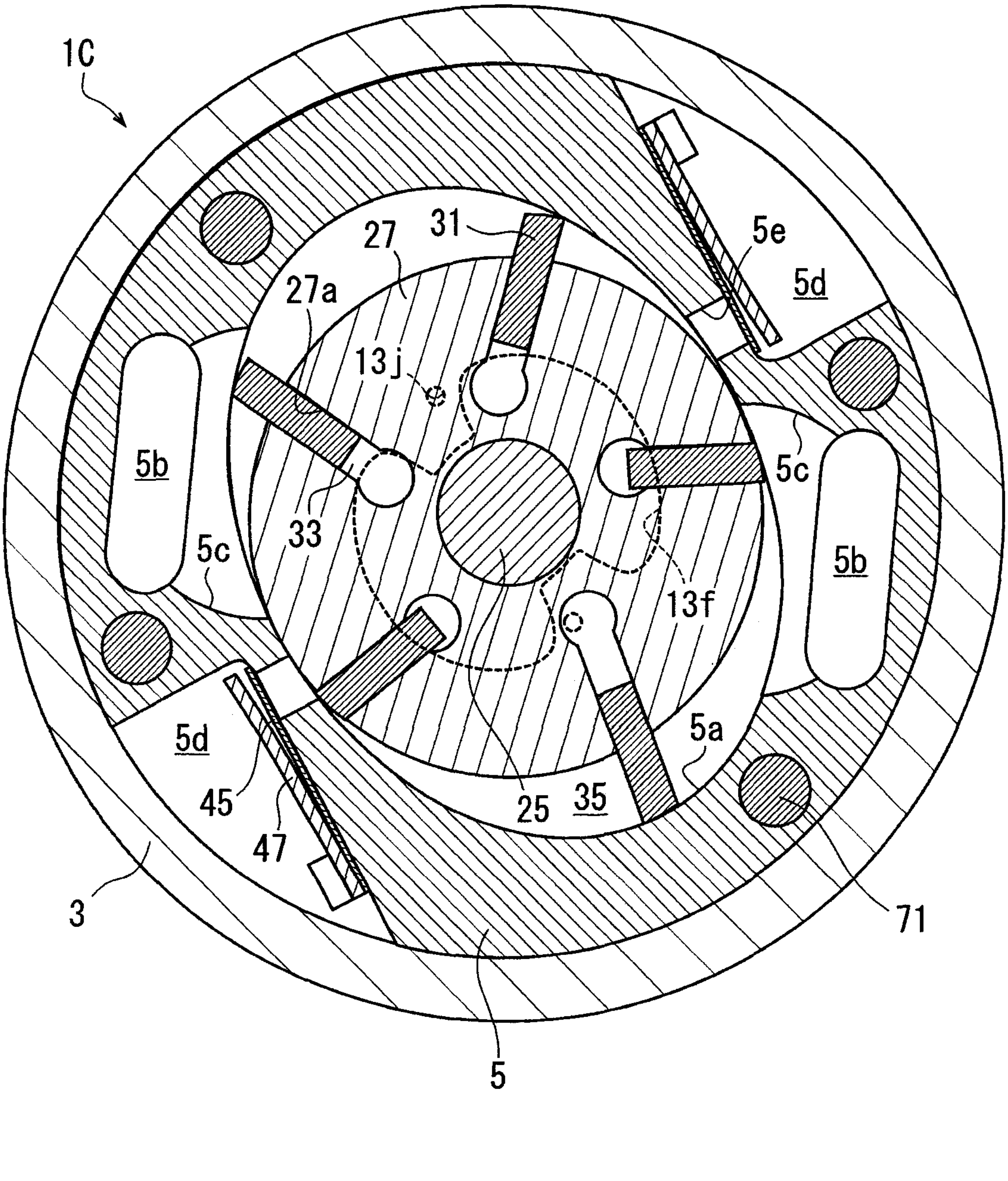


Fig.1

Shakira

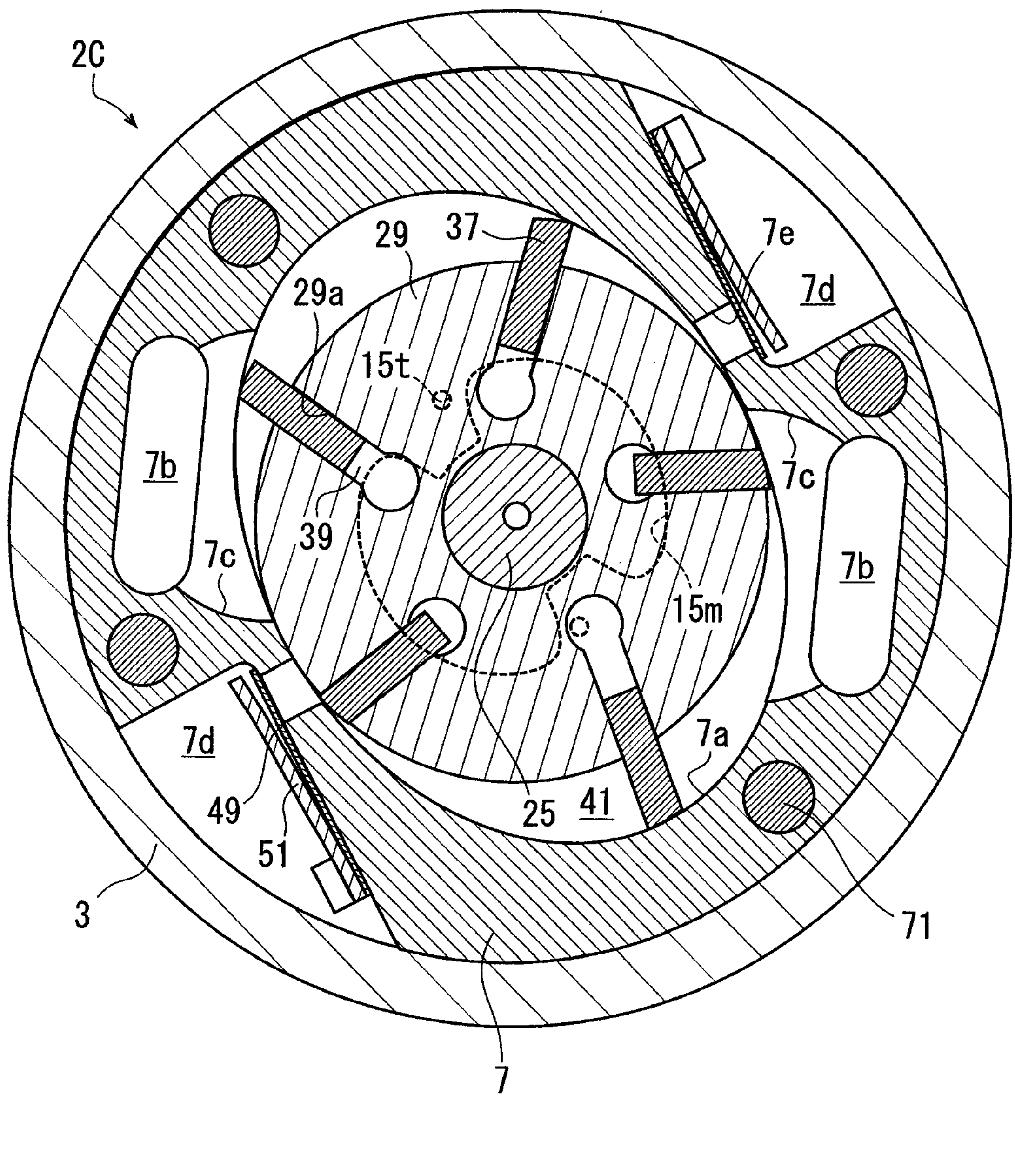
(Shakira N)
REG. No.: IN/PA-972
Of De Penning & De Penning
Agent for the Applicant

Fig.2



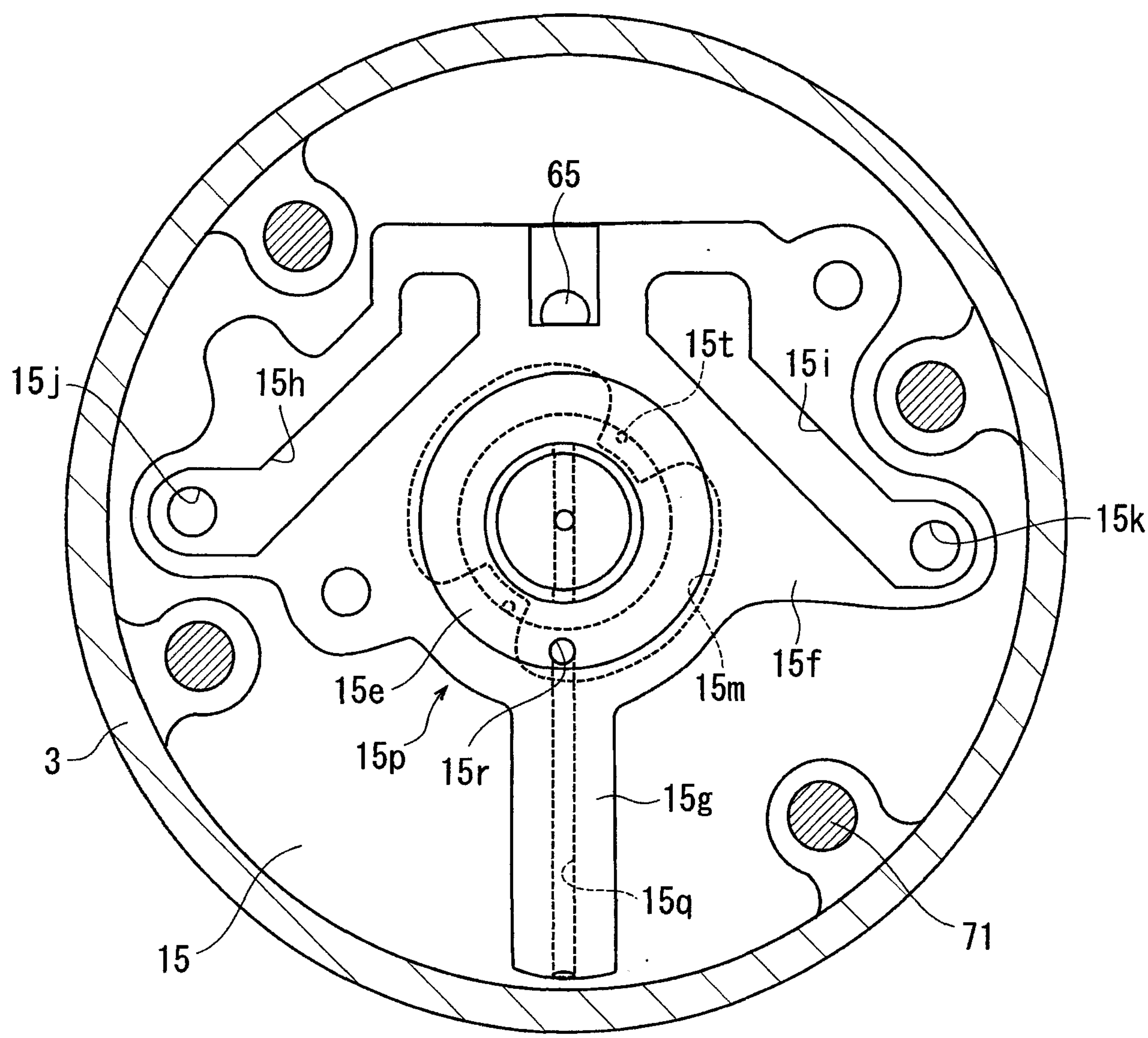
(Shakira N)
REG. No.: IN/PA-972
Of De Penning & De Penning
Agent for the Applicant

Fig.3



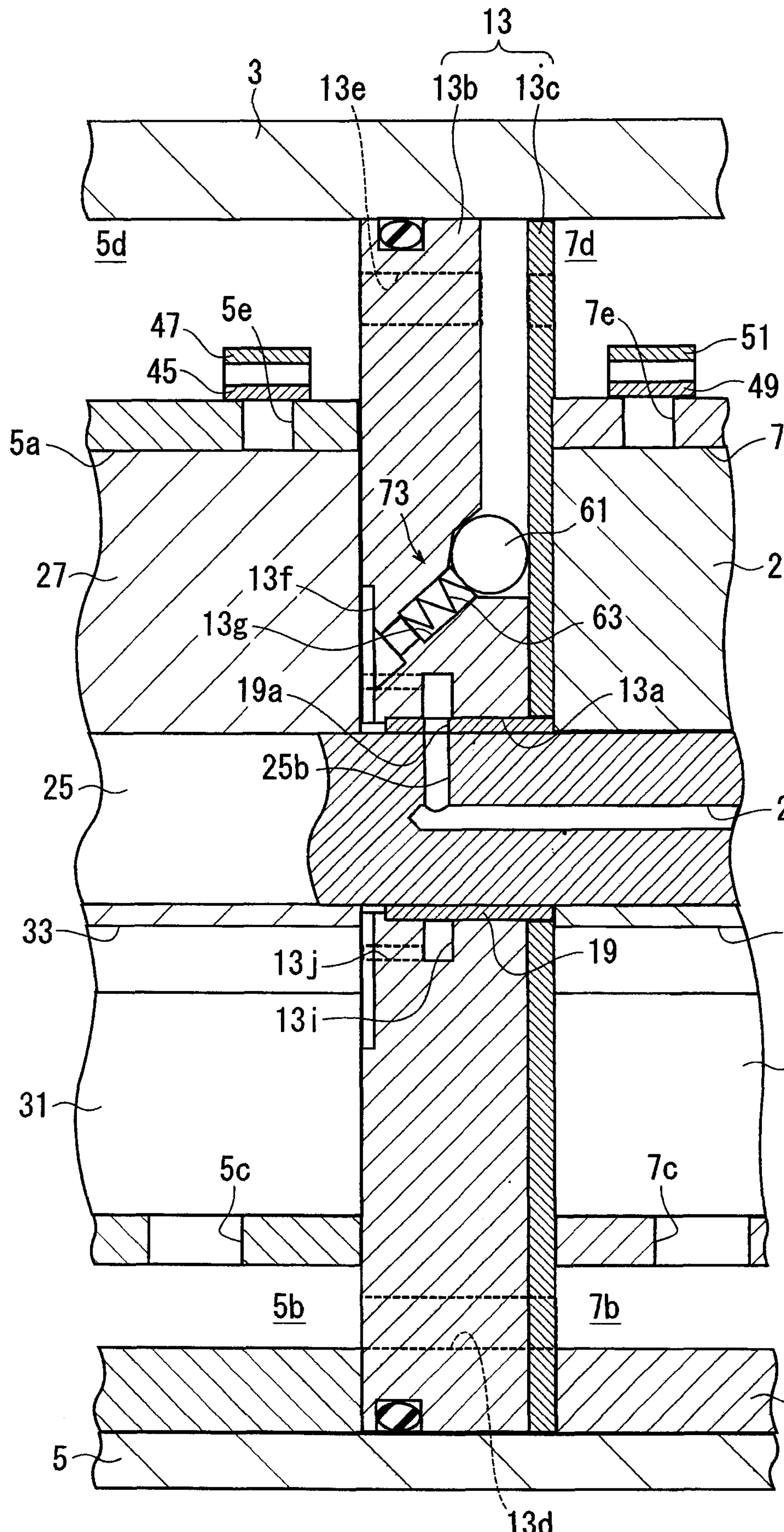
(Shakira N)
REG. No.: IN/PA-972
Of De Penning & De Penning
Agent for the Applicant

Fig.4



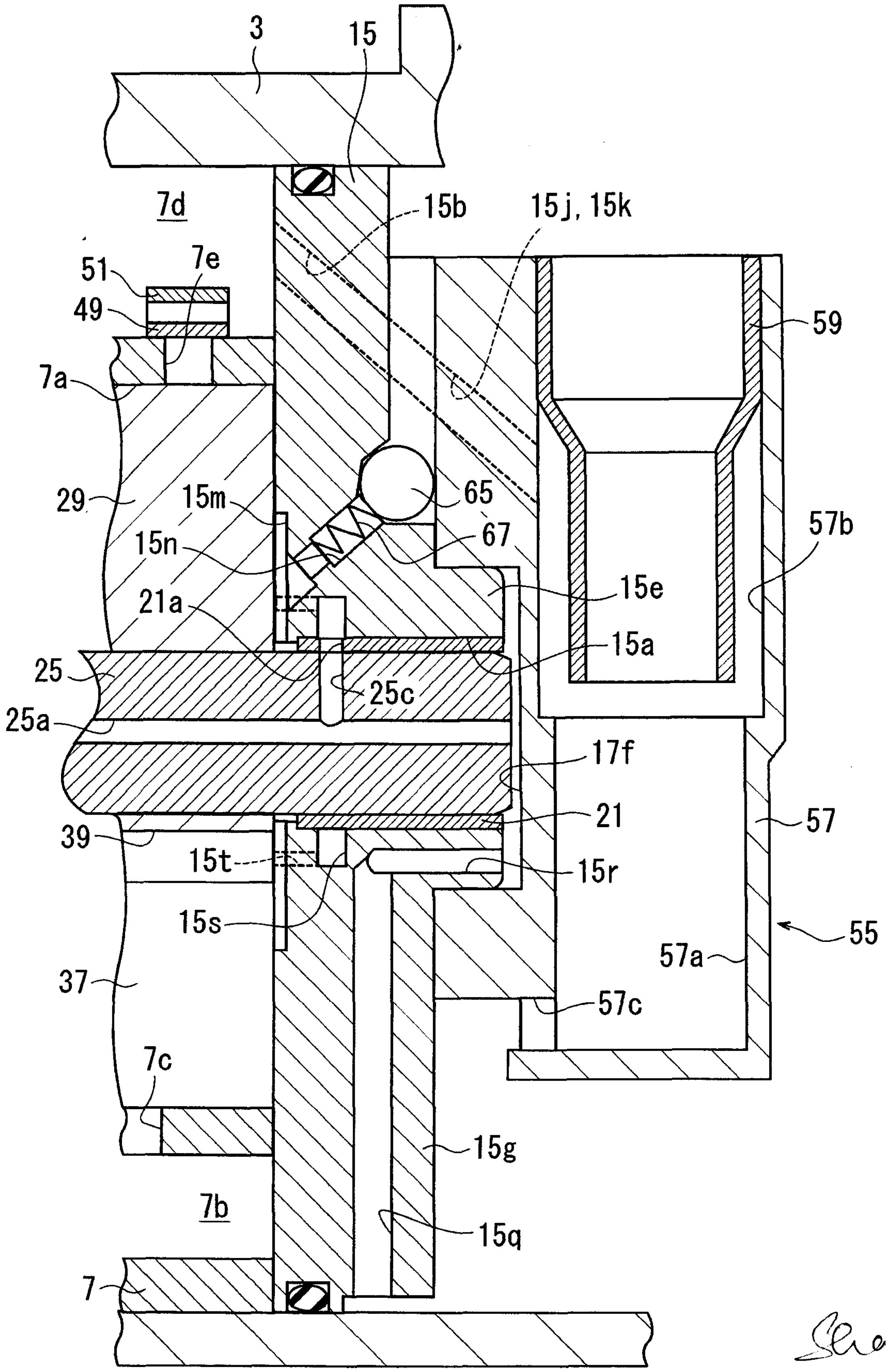
(Shakira N)
REG. No.: IN/PA-972
Of De Penning & De Penning
Agent for the Applicant

Fig.5



(Shakira N)
REG. No.: IN/PA-972
Of De Penning & De Penning
Agent for the Applicant

Fig.6



(Shakira N)
REG. No.: IN/PA-972
Of De Penning & De Penning
Agent for the Applicant

Fig.7

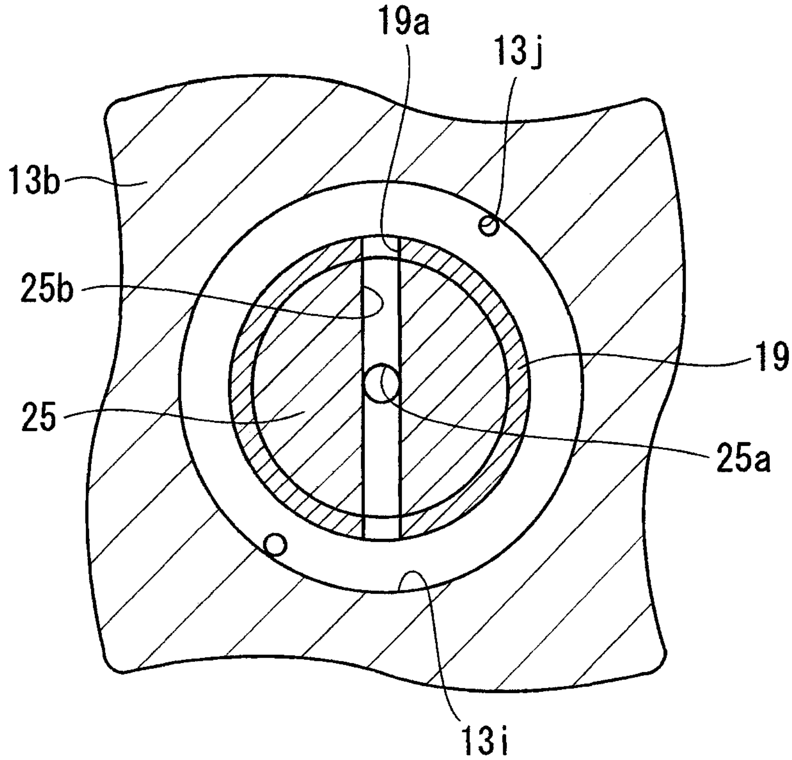
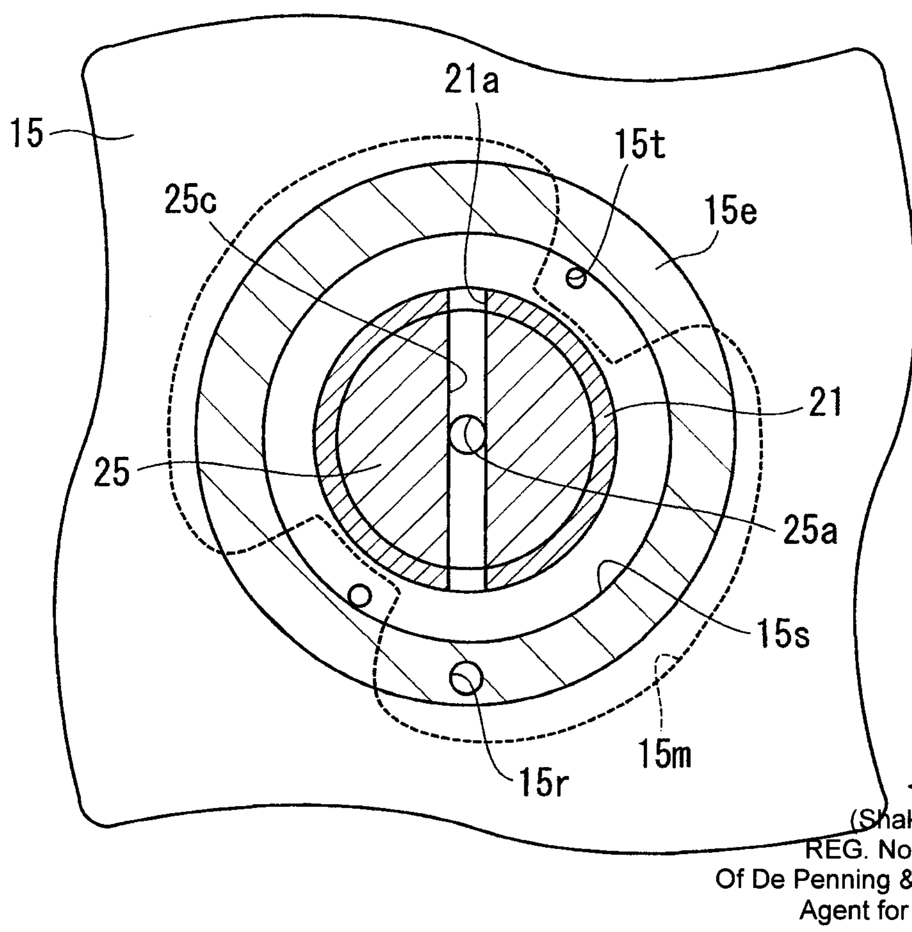


Fig.8



(Shakira N)
REG. No.: IN/PA-972
Of De Penning & De Penning
Agent for the Applicant

Fig.9

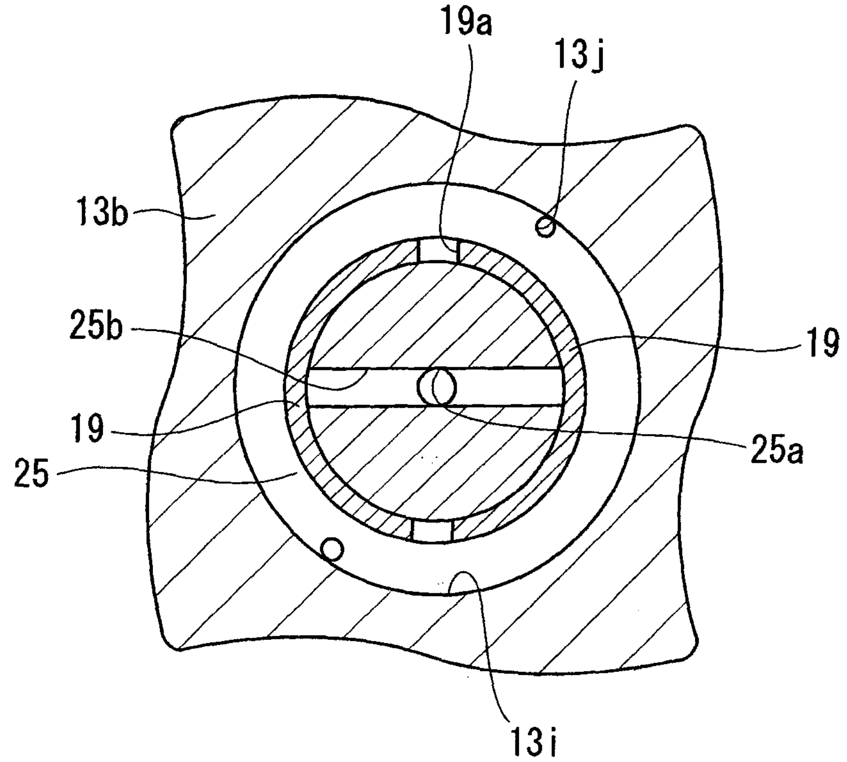
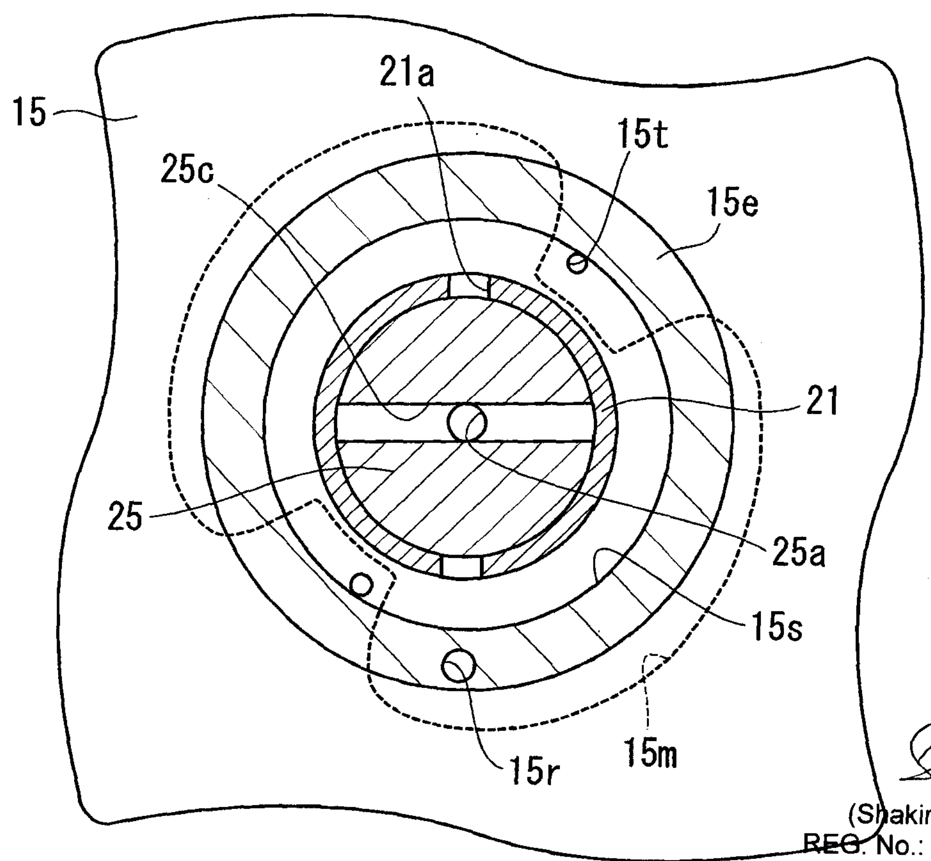


Fig.10



Shakira

(Shakira N)
~~REG.~~ No.: IN/PA-972
Of De Penning & De Penning
Agent for the Applicant

Fig.11

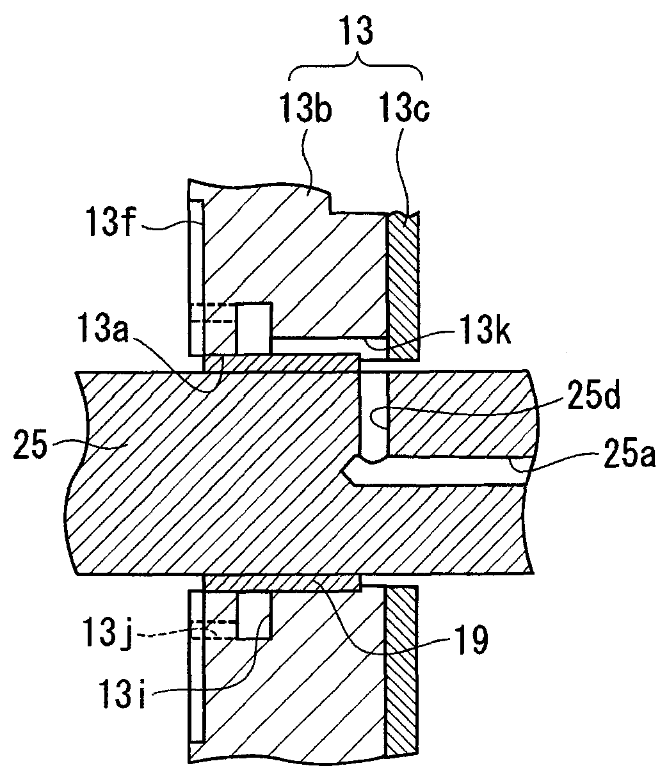
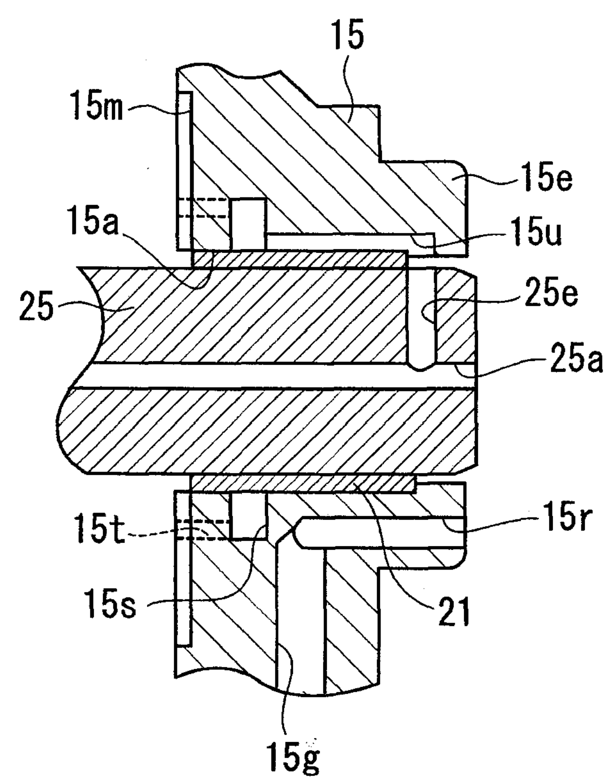


Fig.12



(Shakira N)
REG. No.: IN/PA-972
Of De Penning & De Penning
Agent for the Applicant

Fig.13

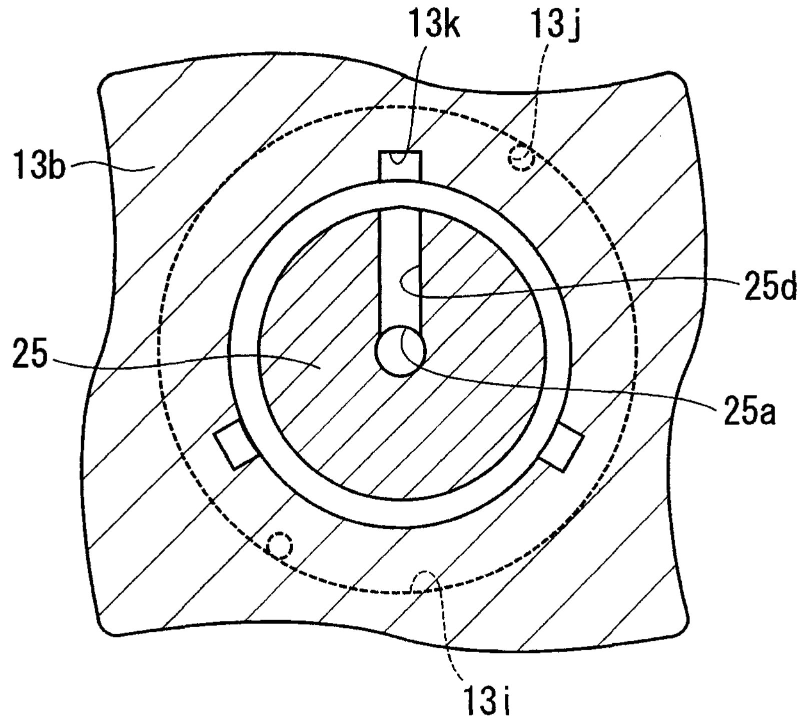
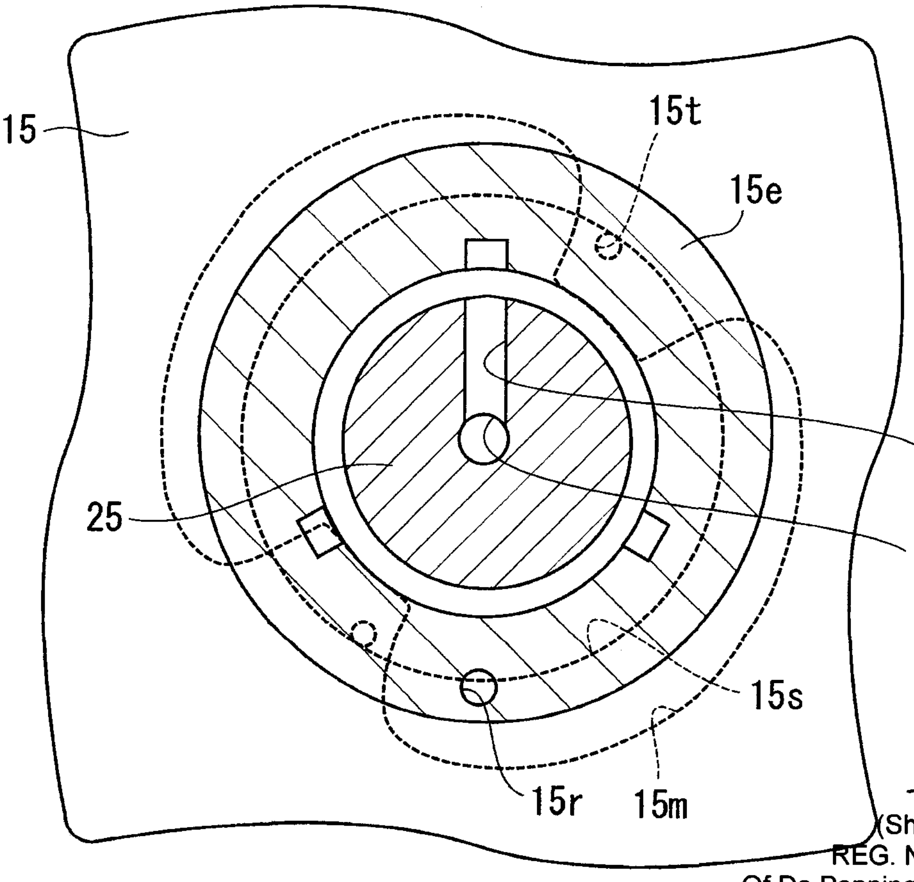


Fig.14



(Shakira N)
REG. No.: IN/PA-972
Of De Penning & De Penning
Agent for the Applicant

Fig.15

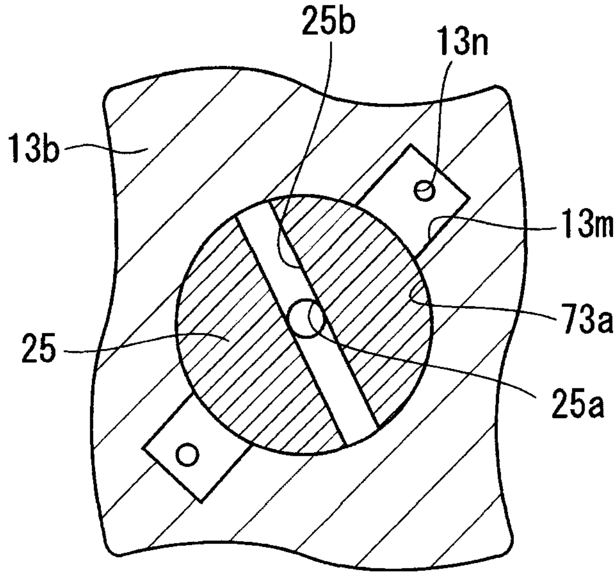
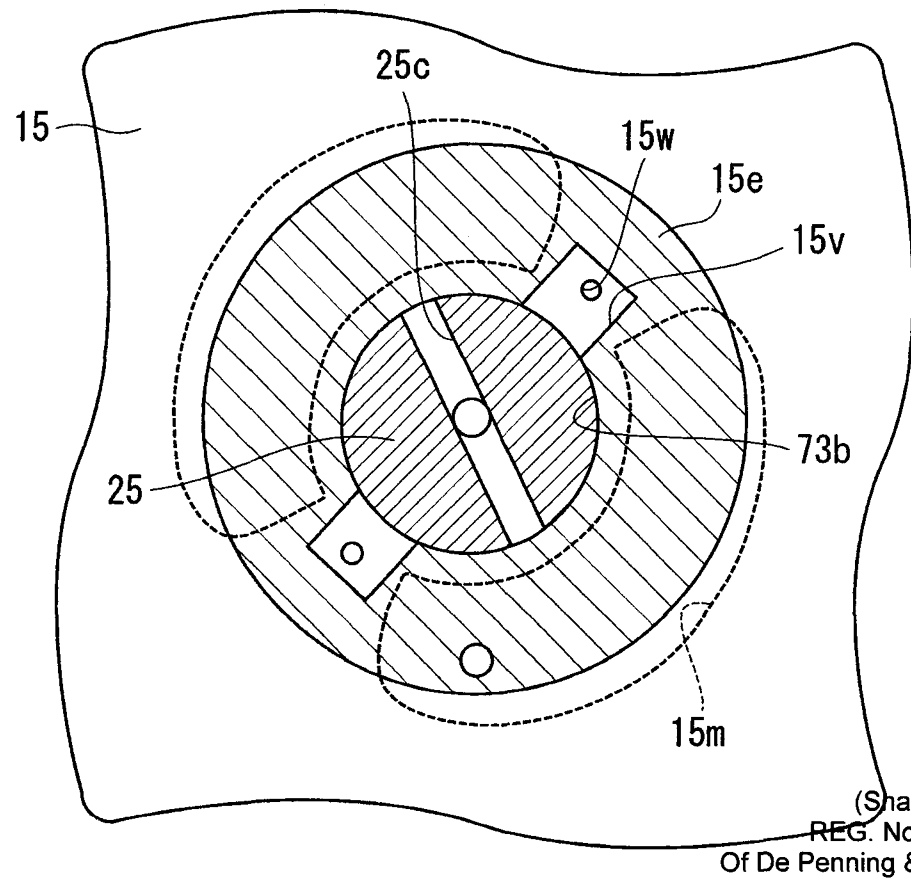
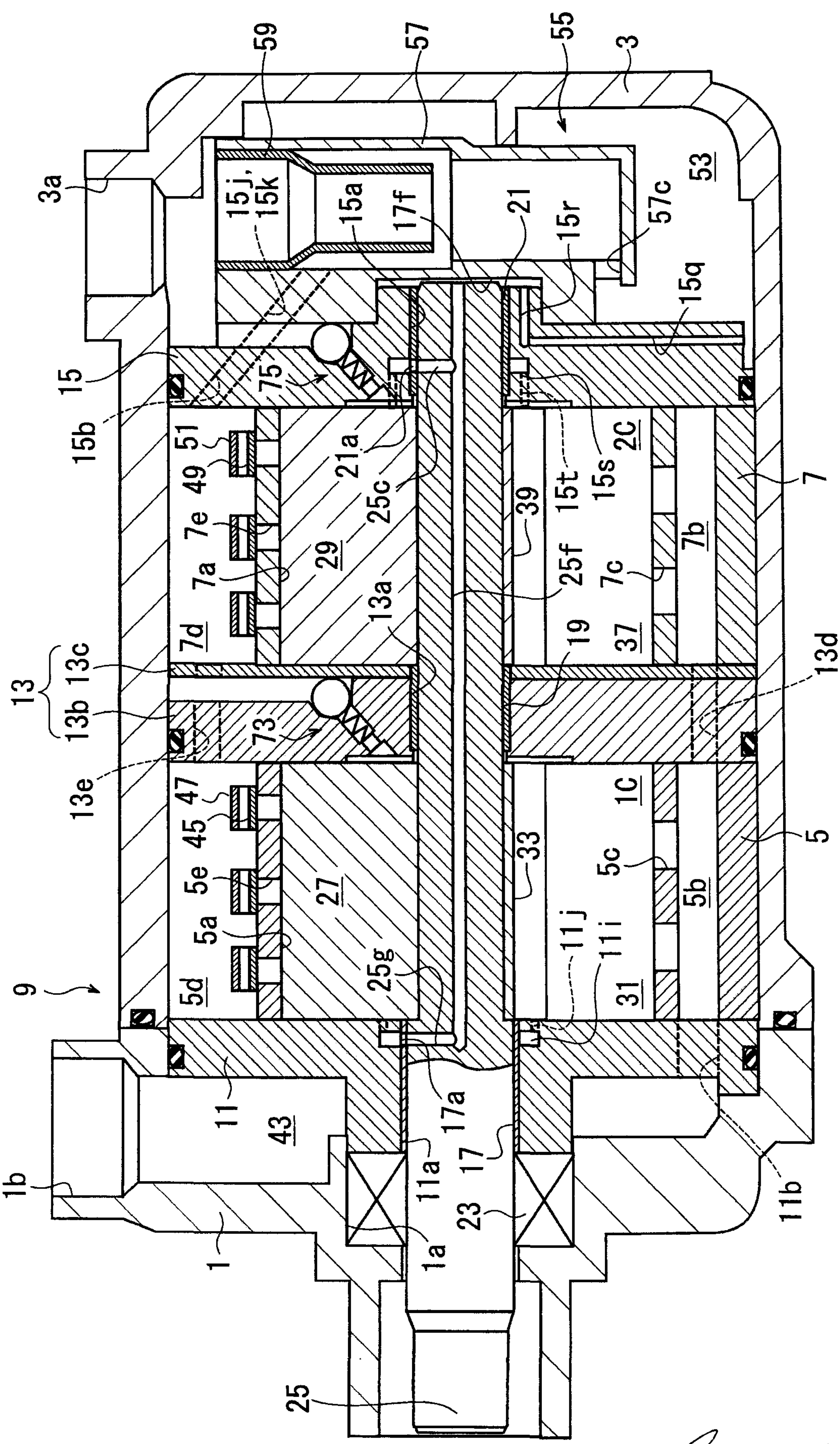


Fig.16



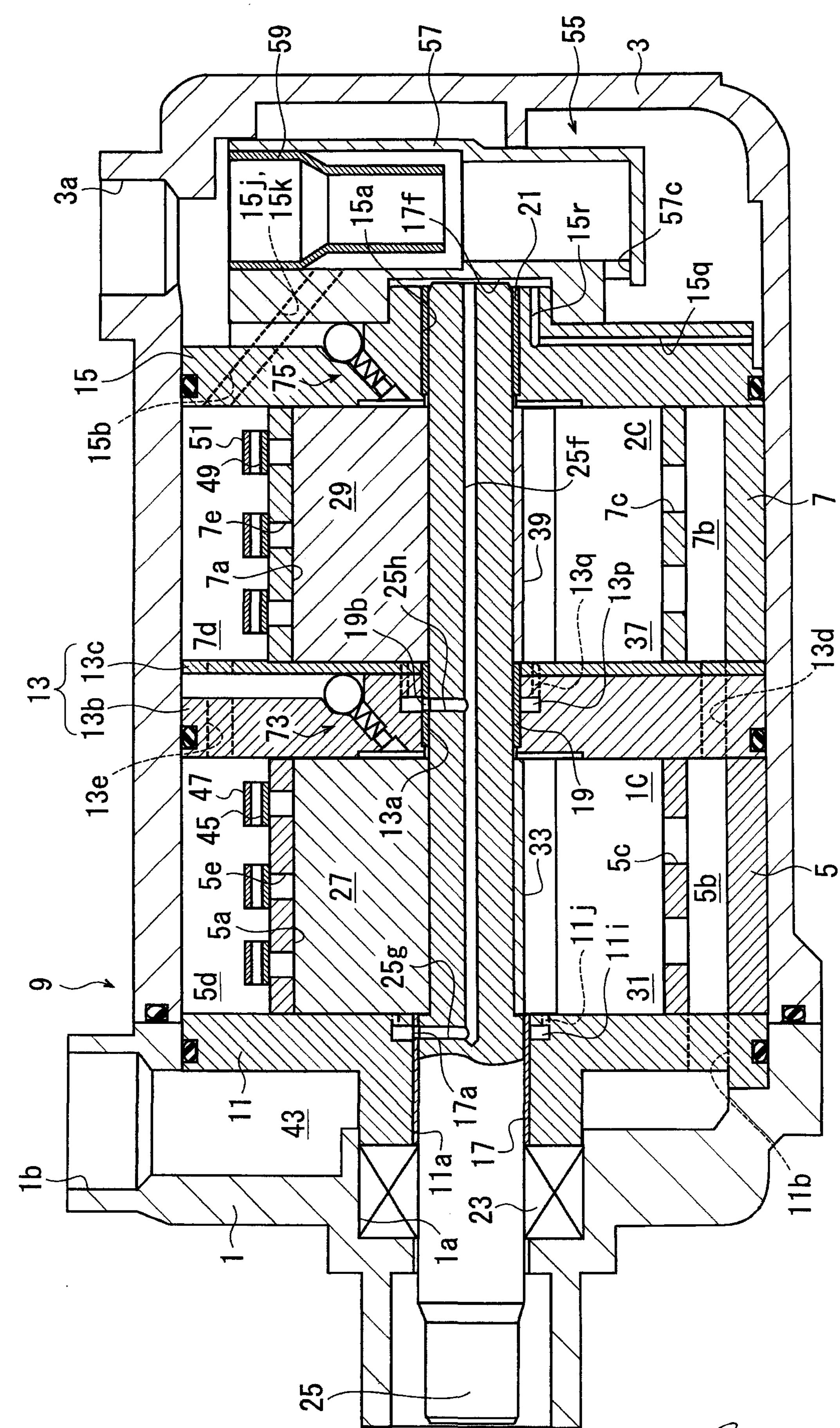
Shakira
(Shakira N)
REG. No.: IN/PA-972
Of De Penning & De Penning
Agent for the Applicant

Fig.17



(Shakira N)
REG. No.: IN/PA-972
Of De Penning & De Penning
Agent for the Applicant

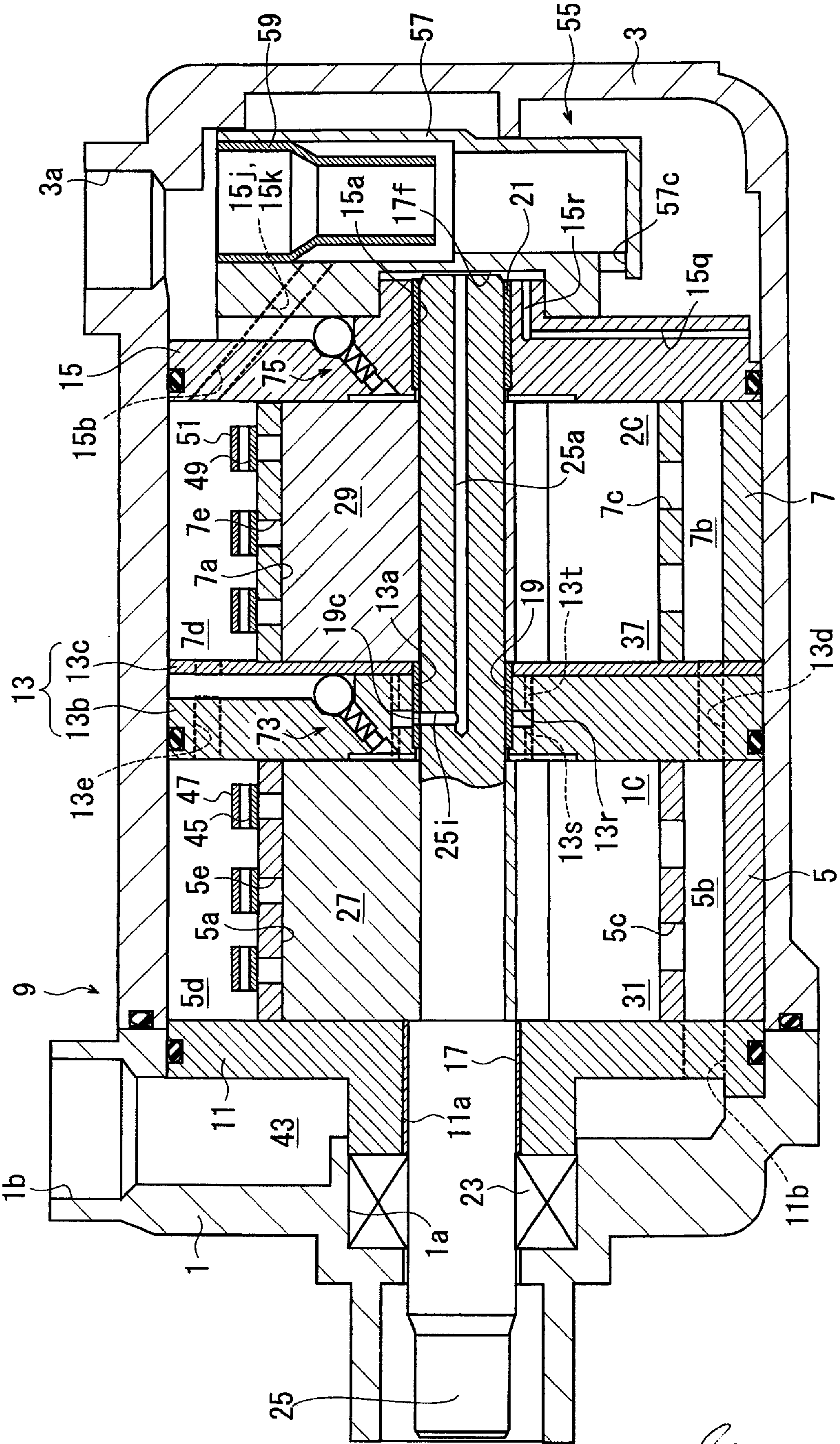
Fig.18



Shakira N

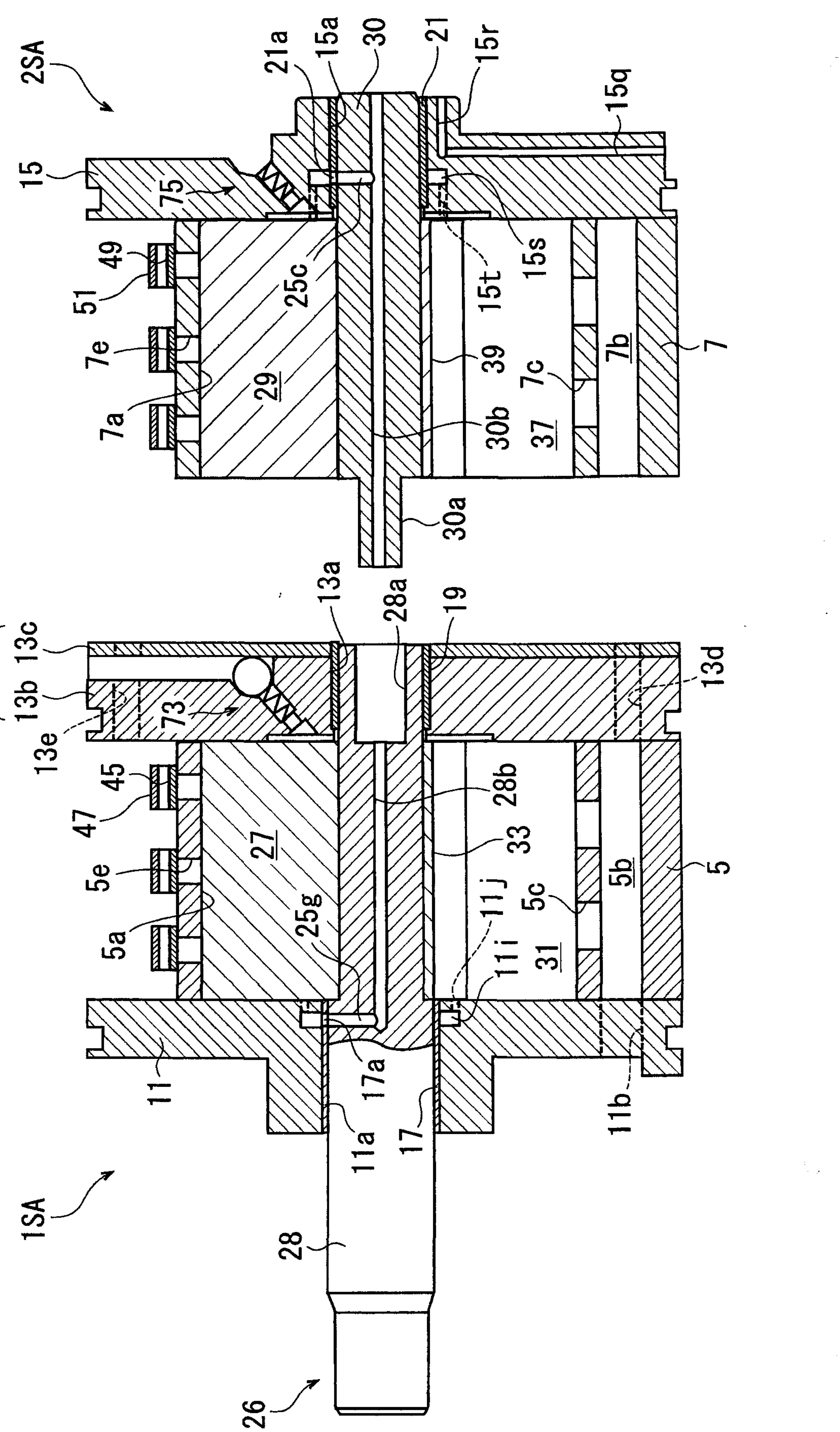
(Shakira N)
REG. No.: IN/PA-972
Of De Penning & De Penning
Agent for the Applicant

Fig.19



(Shakira N)
REG. No.: IN/PA-972
Of De Penning & De Penning
Agent for the Applicant

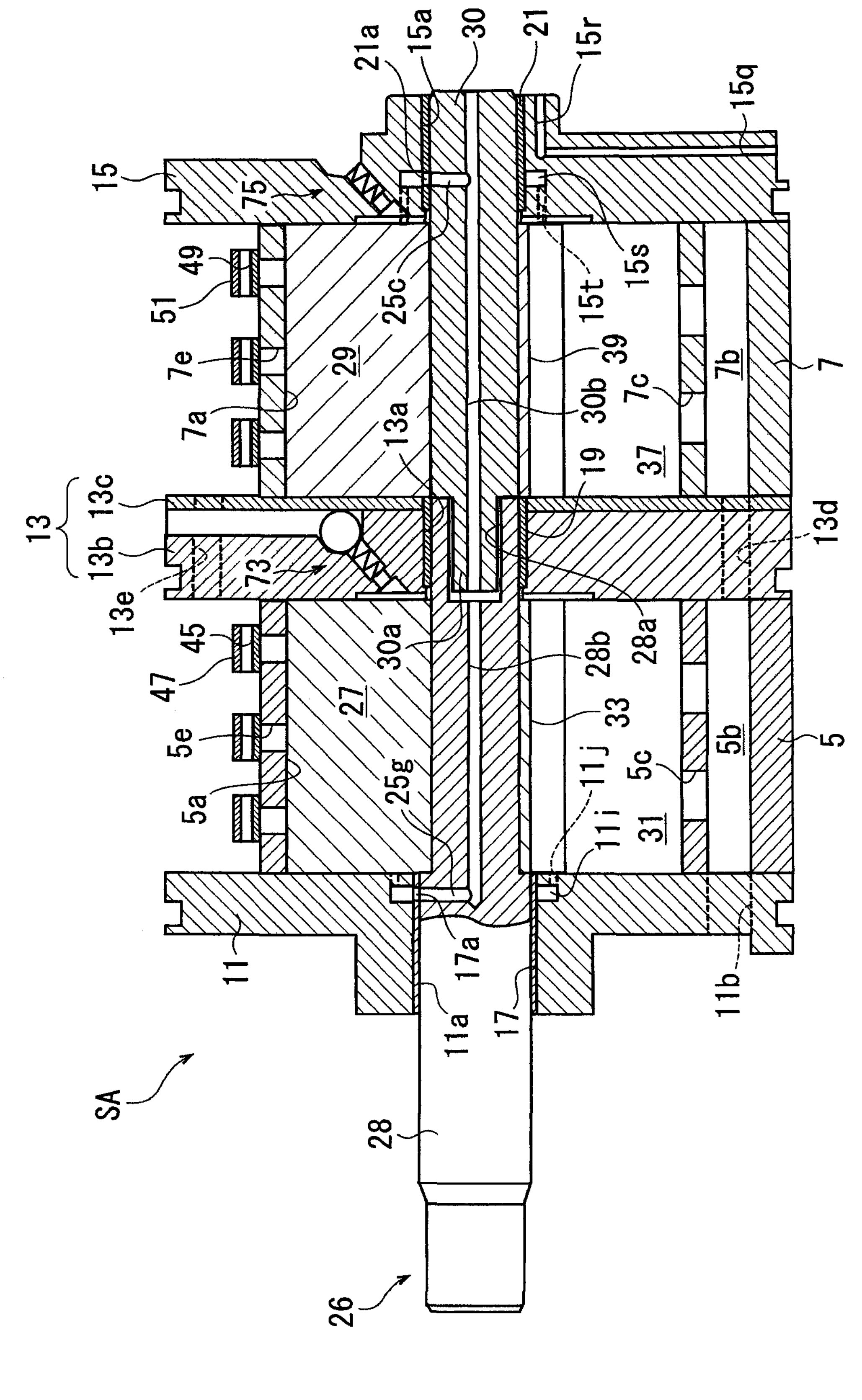
Fig.20



Shakira

(Shakira N)
REG. No.: IN/PA-972
Of De Penning & De Penning
Agent for the Applicant

Fig.21



(Shakira N)
REG. No.: IN/PA-972
Of De Penning & De Penning
Agent for the Applicant

Fig.22

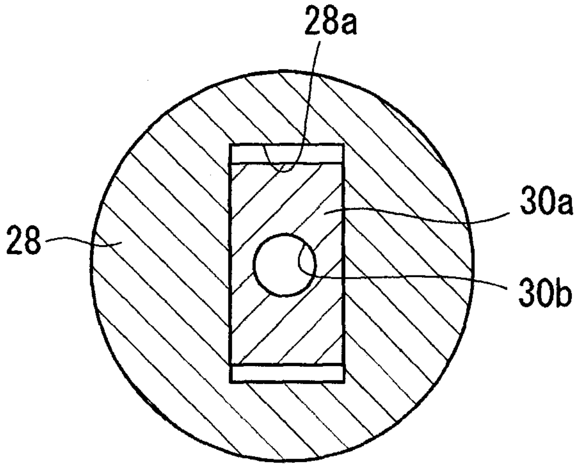
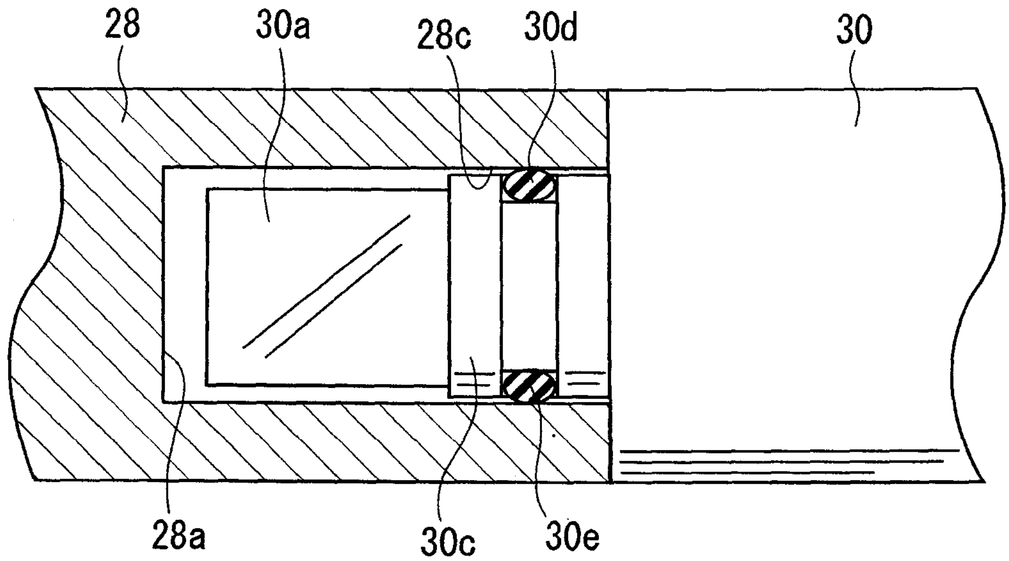


Fig.23



(Shakira N)
REG. No.: IN/PA-972
Of De Penning & De Penning
Agent for the Applicant

TANDEM TYPE VANE COMPRESSOR

BACKGROUND OF THE INVENTION

The present invention relates to a tandem type vane compressor.

Japanese Laid-Open Patent Publication No. 59-90086, Japanese Laid-Open Patent Publication No. 58-144687, Japanese Laid-Open Utility Model publication No. 3-102086, Japanese Laid-Open Utility Model Publication No. 60-39793, and Japanese Laid-Open Utility Model Publication No. 3-118294 disclose conventional tandem type vane compressors. These tandem type vane compressors have a suction chamber, a discharge chamber, a compression chamber in a housing, and a rotationally-supported drive shaft. Further, in the housing, a plurality of compression mechanisms are coupled in tandem to perform a suction process, in which the compression chamber draws a low-pressure refrigerant gas from the suction chamber, a compression process, in which the refrigerant gas is compressed in the compression chamber, and a discharge process, in which the high-pressure refrigerant gas in the compression chamber is discharged to the discharge chamber.

Each compression mechanism includes a first compression mechanism and a second compression mechanism. The first compression mechanism includes a first cylinder chamber formed in the housing, and a first rotor arranged in the first cylinder chamber to be rotational by the drive shaft. A plurality of radially extending primary vane grooves is formed in the first rotor. Further, the first compression mechanism includes primary vanes, which are arranged to be capable of projecting and retracting in respective primary vane grooves and form a primary compression chamber with the inner surface of the first cylinder chamber and the outer surface of the first rotor. The primary compression chamber is positioned at the front side.

Similar to the first compression mechanism, the second compression mechanism includes a second cylinder chamber formed in the housing, a second rotor, which is arranged in the second cylinder chamber and rotated by the drive shaft. A plurality of radially-extending secondary vane grooves is formed similarly in the second rotor. Further, the second compression mechanism similarly includes secondary vanes, which are arranged to be capable of projecting and retracting in respective secondary vane grooves and form a secondary compression chamber with the inner surface of the second cylinder chamber and the outer surface of the second rotor. The secondary compression chamber is positioned at the rear side.

In a case where these tandem type vane compressors are used in an air conditioning device of a vehicle and the like, the drive shaft is rotated and driven via an electromagnetic clutch, for example. Due to this, the first and second compression mechanisms are operated. That is, the first and second rotors rotate, and the primary and secondary compression chambers perform the suction process, the compression process, and the discharge process. Due to this, the refrigerant gas is drawn into the primary and secondary compression chambers from the suction chamber, is compressed in the primary and secondary compression chambers, and is discharged to the discharge chamber. The high-pressure refrigerant gas discharged to the discharge chamber is supplied to a refrigeration circuit of the air conditioning device.

Accordingly, in these tandem type vane compressors, since the primary and secondary compression chambers each perform the suction process, the compression process, and the discharge process, the discharge amount per rotation of the drive shaft can be increased.

Further, in order to increase the discharge amount, when a single-cylinder type vane compressor, which has a single cylinder chamber and a rotor, is simply axially elongated, it is expected to cause each vane to be easily tilted with respect to the front-rear direction. This may cause refrigerant gas to be easily leaked from a compression chamber, and deteriorate the sliding characteristics of each vane. In this respect, in the tandem type vane compressor, it is expected that each of primary and secondary vanes is unlikely to be tilted with respect to the front-rear direction, leakage of the refrigerant gas from the primary and secondary compression chambers is reduced, and the sliding characteristics of each of the primary and secondary vanes is increased. Due to this, an excellent mechanical efficiency is expected to be exhibited in the tandem type vane compressors.

Further, since the tandem type vane compressors may have the same shell diameter as the single-cylinder type vane compressors, they are relatively compact; that is, it is relatively easy to mount such compressors in a crowded engine compartment.

However, the conventional tandem type vane compressors as above are not configured to supply high-pressure lubricant oil to a primary backpressure chamber formed between the bottom surface of each primary vane and each primary vane groove and to a secondary backpressure chamber formed between the bottom surface of each secondary vane and each secondary vane groove. Thus, in the tandem type vane compressors, while the first and second compression mechanisms respectively perform the compression process and the discharge process, each of the primary and secondary vanes is not pressed against the inner surfaces of the first and second cylinder chambers. Therefore, the refrigerant gas may leak from the primary and secondary compression chambers. Due to this, it is difficult to reliably exhibit the high mechanical efficiency in the tandem type vane compressors.

Further, in the tandem type vane compressors, when the drive shaft is not rotated by the electromagnetic clutch, the refrigerant gas within the discharge chamber and the lubricant oil contained in the refrigerant gas flow back to the compression chambers, so that the drive shaft may be rotated reversely. In this case, since an evaporator is heated by the high-temperature refrigerant gas flowing back to the suction side of the refrigerant circuit, the temperature of air flowing into the passenger compartment upon restarting the rotation of the drive shaft is increased. This causes a decrease in refrigerating efficiency. Further, a decrease in durability is caused by the occurrence of liquid compression when the compressor is started again. Noise also is generated upon the reverse rotation. Due to this, if an on-off valve is provided in the tandem type vane compressor, or a check valve is provided in the discharge chamber or the suction chamber, spaces for the on-off valve and the check valve need to be provided therein. This is likely to increase the size of the tandem type vane compressor.

An objective of the present invention is to provide a tandem type vane compressor that increases the discharge amount per rotation of a drive shaft, is reliably compact and efficient, has good mounting characteristics, and reduces backflow of refrigerant gas and the like and reverse rotation of the drive shaft.

SUMMARY OF THE INVENTION

To achieve the foregoing objective and in accordance with one aspect of the present invention, a tandem type vane compressor is provided that includes a housing and a plurality of compression mechanisms. The housing has a suction chamber, a discharge chamber, and a plurality of compression chambers. The housing rotationally supports a drive shaft. The compression mechanisms include a first compression mechanism and a second compression mechanism, which are coupled to each other in tandem in the housing. The first and second compression mechanisms each have at least one of the compression chambers. Each compression mechanism is driven by rotation of the drive shaft to perform a suction process, in which each compression mechanism draws low-pressure refrigerant gas into the respective compression chamber from the suction chamber, a compression process, in which each compression mechanism compresses the refrigerant gas in the respective compression chamber, and a discharge process, in which each compression mechanism discharges the high-pressure refrigerant gas in the respective compression chamber to the discharge chamber. The first compression mechanism includes a first cylinder chamber, which is formed in the housing, a first rotor, which is provided in the first cylinder chamber to rotate when the drive shaft rotates, and a plurality of primary vanes. The first rotor has a plurality of radially extending primary vane grooves. Each primary vane is located in one of the primary vane grooves and is capable of projecting and retracting. The compression chamber of the first compression mechanism is defined by an inner surface of the first cylinder chamber, an outer surface of the first rotor, and the primary vanes. The compression chamber of the first compression mechanism is located at a position forward of the compression chamber of the second compression mechanism. The second compression mechanism includes a second cylinder chamber, which is formed in the housing, a second rotor, which is provided in the second cylinder chamber to rotate when the drive shaft rotates, and a plurality of secondary vanes. The second rotor has a plurality of radially extending secondary vane grooves. Each secondary vane is located in one of the secondary vane grooves and is capable of projecting and retracting. The compression chamber of the second compression mechanism is defined by an inner surface of the second cylinder chamber, an outer surface of the second rotor, and the secondary vanes. A bottom surface of each primary vane and the corresponding primary vane groove define a primary backpressure chamber. A first backpressure supplying mechanism is provided, which, during the compression process of the first compression mechanism, allows each primary back pressure chamber to communicate with the discharge chamber. The first backpressure supplying mechanism includes a first rotation path, which is formed in the drive shaft or in a rotator that rotates in synchronization with the drive shaft, and a first intermittent mechanism, which selectively allows and interrupts communication of the first rotation path with the discharge chamber and with each primary backpressure chamber according to the phase in the rotation direction of the drive shaft. A bottom surface of each secondary vane and the corresponding secondary vane groove define a secondary backpressure chamber. A second backpressure supplying mechanism is provided, which, during the compression process of the second compression mechanism, allows each secondary back pressure chamber to communicate with the discharge chamber. The second backpressure supplying mechanism includes a second rotation path, which is formed in the drive shaft or in the rotator, and a second intermittent mechanism, which selectively allows and interrupts communication of the second rotation path with the discharge chamber and with each secondary backpressure chamber according to the phase in the rotation direction of the drive shaft.

In the tandem type vane compressor of the present invention, each primary backpressure chamber is connected to the discharge chamber by the first backpressure supplying mechanism in a compression process of the first compression mechanism, and each secondary backpressure chamber is connected to the discharge chamber by the second backpressure supplying mechanism in a compression process of the second compression mechanism. Due to this, high-pressure lubricant oil can be supplied to the primary backpressure chambers and the secondary backpressure chambers. Due to this, in the tandem type vane compressor, each of primary and secondary vanes is suitably pressed against the inner surfaces of first and second cylinder chambers while the first and second compression mechanisms respectively perform the compression process and the discharge process. As a result, leakage of the refrigerant gas from the primary and secondary compression chambers is reduced. Due to this, the tandem type vane compressor has high mechanical efficiency. Further, in the tandem type vane compressor, each primary backpressure chamber and each secondary backpressure chamber do not need to be connected to the discharge chamber separately. This reduces the production costs.

Further, in the tandem type vane compressor, the first backpressure supplying mechanism includes a first rotation path and a first intermittent mechanism, and the first intermittent mechanism allows and interrupts communication of the first rotation path with the discharge chamber and the primary backpressure chamber depending on the phase of the drive shaft in the rotation direction.

In other words, when the drive shaft is at a first phase in the rotation direction, the first rotation path allows communication between the discharge chamber and each primary backpressure chamber, and the high-pressure lubricant oil can be supplied to each primary backpressure chamber. Due to this, each primary vane is suitably pressed against the inner surface of the first cylinder chamber while the first compression mechanism performs the compression process and the discharge process. Thus, leakage of the refrigerant gas from the primary compression chamber is reduced.

Further, when the drive shaft is at a second phase in the rotation direction, the first rotation path interrupts communication between the discharge chamber and each primary backpressure chamber. Due to this, during the compression process, the high-pressure lubricant oil is supplied intermittently to each primary backpressure chamber, and each primary vane is intermittently pressed against the inner surface of the first cylinder chamber. Thus, each primary vane is lubricated within the primary vane grooves, thereby preventing chatter, and the leakage of the refrigerant gas from the primary compression chamber is prevented, leading to an increased efficiency.

If the rotation of the drive shaft is stopped and communication between the discharge chamber and each primary backpressure chamber is interrupted, the backflow of the refrigerant gas and the like and the reverse rotation of the drive shaft are not caused. Even if the rotation of the drive shaft is stopped in a state in which the discharge chamber and each primary backpressure chamber are connected with each other, the phase of the drive shaft is displaced due to the backflow of the refrigerant gas and the like and the reverse rotation of the drive shaft even at the slightest degree. Accordingly, the communication between the discharge chamber and each primary backpressure chamber immediately becomes interrupted, the backflow of the refrigerant gas and the like and the reverse rotation of the drive shaft are no longer caused. Due to this, the tandem type vane compressor can reliably and quickly prevent the backflow of the refrigerant gas and the like and the reverse rotation of the drive shaft.

In the tandem type vane compressor, the first rotation path is formed on the drive shaft or a rotator. Due to this, spaces for an on-off valve and a check valve do not need to be provided inside as has been conventionally necessary, and the tandem type vane compressor is not increased in size. Further, since the first rotation path can easily be formed on the drive shaft or the rotator, a burden of developing a large number of different varieties depending on vehicles and the like is also omitted.

The same applies to the second compression mechanism.

Thus, with the tandem type vane compressor, the discharge amount per rotation of the drive shaft is increased, and the compactness and efficiency are favorable. Further, with to the tandem type vane compressor, the backflow of the refrigerant gas and the like and the reverse rotation of the drive shaft are reduced.

In the tandem type vane compressor of the invention, the compression mechanisms may include other compression mechanisms other than the first compression mechanism and the second compression mechanism.

The rotator may be integrated with the drive shaft, or may be a separate component from the drive shaft and fixed to the drive shaft to rotate in synchrony with the drive shaft.

Other aspects and advantages of the present invention will become apparent from the following description, taken in conjunction with the accompanying drawings, illustrating by way of example the principles of the invention.

BRIEF DESCRIPTION OF THE DRAWINGS

The invention, together with objects and advantages thereof, may best be understood by reference to the following description of the presently preferred embodiments together with the accompanying drawings in which:

Fig. 1 is a vertical cross-sectional view of a tandem type vane compressor of a first embodiment;

Fig. 2 is cross-section view taken along line II-II of Fig. 1, illustrating the tandem type vane compressor of the first embodiment;

Fig. 3 is cross-section view taken along line III-III of Fig. 1, illustrating the tandem type vane compressor of the first embodiment;

Fig. 4 is a partially cross-sectional rear view illustrating the tandem type vane compressor of the first embodiment, with the separator is dismounted;

Fig. 5 is a partially enlarged vertical cross-sectional view of the tandem type vane compressor of the first embodiment;

Fig. 6 is a partially enlarged vertical cross-sectional view of the tandem type vane compressor of the first embodiment;

Fig. 7 is a partially enlarged horizontal cross-sectional view of the tandem type vane compressor of the first embodiment;

Fig. 8 is a partially enlarged horizontal cross-sectional view of the tandem type vane compressor of the first embodiment;

Fig. 9 is a partially enlarged horizontal cross-sectional view of the tandem type vane compressor of the first embodiment;

Fig. 10 is a partially enlarged horizontal cross-sectional view of the tandem type vane compressor of the first embodiment;

Fig. 11 is a partially enlarged vertical cross-sectional view of a tandem type vane compressor of a second embodiment;

Fig. 12 is a partially enlarged vertical cross-sectional view of the tandem type vane compressor of the second embodiment;

Fig. 13 is a partially enlarged horizontal cross-sectional view of the tandem type vane compressor of the second embodiment;

Fig. 14 is a partially enlarged horizontal cross-sectional view of the tandem type vane compressor of the second embodiment;

Fig. 15 is a partially enlarged horizontal cross-sectional view of a tandem type vane compressor of a third embodiment;

Fig. 16 is a partially enlarged horizontal cross-sectional view of the tandem type vane compressor of the third embodiment;

Fig. 17 is a vertical cross-sectional view of a tandem type vane compressor of a fourth embodiment;

Fig. 18 is a vertical cross-sectional view of a tandem type vane compressor of a fifth embodiment;

Fig. 19 is a vertical cross-sectional view of a tandem type vane compressor of a sixth embodiment;

Fig. 20 is a disassembled cross-sectional view of a sub assembly in a tandem type vane compressor of a seventh embodiment;

Fig. 21 is a partially enlarged cross-sectional view of the sub assembly in the tandem type vane compressor of the seventh embodiment;

Fig. 22 is a horizontal cross-sectional view of the tandem type vane compressor of the seventh embodiment with its primary portion being enlarged; and

Fig. 23 is a partially enlarged vertical cross-sectional view of a tandem type vane compressor of a modification of the seventh embodiment.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

Hereinbelow, first to seventh embodiments of the present invention will be described with reference to the drawings.

(First Embodiment)

As shown in Fig. 1, a tandem type vane compressor of the first embodiment has a first side plate 11, a first cylinder block 5, a second side plate 13, a second cylinder block 7, and a third side plate 15, which are fixed in a state of being housed in a front housing member 1 and a rear housing member 3, which are coupled to each other. The front housing member 1 and the rear housing member 3 form a shell 9. The diameter of the shell 9 is the same as that of a single-cylinder type vane compressor. Further, the first and second cylinder blocks 5, 7 have an identical outer shape.

As shown in Fig. 2 also, a first cylinder chamber 5a of elliptical shape in a direction orthogonal to the axis is formed in the first cylinder block 5. As shown in Fig. 3 also, a second cylinder chamber 7a having the identical shape as the first cylinder chamber 5a is formed in the second cylinder block 7. The first and second cylinder blocks 5, 7 are fixed such that the first and second cylinder chambers 5a, 7a have an identical phase.

As shown in Fig. 1, the first cylinder block 5 is housed in the shell 9 while being sandwiched by the first side plate 11 and the second side plate 13. The second side plate 13 is formed of a second side plate main body 13b positioned on the front side and a second side plate cover 13c positioned on the rear side of the second side plate main body 13b. Front and rear ends of the first cylinder chamber 5a are respectively closed by the first side plate 11 and the second side plate main body 13b.

Further, the second cylinder block 7 is housed in the shell 9 while being sandwiched by the second side plate cover 13c and the third side plate 15. Front and rear ends of the second cylinder chamber 7a are respectively closed by the second side plate cover 13c and the third side plate 15. The shell 9, the first and second cylinder blocks 5, 7, and the first to third side plates 11, 13, 15 correspond to a housing.

Axial holes 11a, 13a, 15a are formed respectively to extend through the first to third side plates 11, 13, 15, and slide bearings 17, 19, 21 are press-fitted into the respective axial holes 11a, 13a, 15a. An axial hole 1a is formed to extend through the front housing member 1 also, and a shaft sealing device 23 is press-fitted into the axial hole 1a. A drive shaft 25 is rotationally retained by the shaft sealing device 23 and the slide bearings 17, 19, 21. An electromagnetic clutch or pulley (neither is shown) is fixed to the distal end of the drive shaft 25 exposed from the front housing member 1. Driving force of an engine or a motor of a vehicle is configured to be transmitted to the electromagnetic clutch or pulley.

Further, first and second rotors 27, 29 having a circular cross section are press-fitted about the drive shaft 25. The first rotor 27 is arranged within the first cylinder chamber 5a, and the second rotor 29 is arranged within the second cylinder chamber 7a.

As shown in Fig. 2, five radially extending primary vane grooves 27a are formed on the outer circumferential surface of the first rotor 27, and a primary vane 31 is housed in each primary vane groove 27a to be able to project and retract. A space between the bottom surface of each primary vane 31 and the corresponding primary vane groove 27a is a primary backpressure chamber 33. Five primary compression chambers 35 are each formed by two adjacent primary vanes 31, 31, the outer circumferential surface of the first rotor 27, the inner circumferential surface of the first cylinder block 5, the back surface of the first side plate 11, and the front surface of the second side plate main body 13b.

Further, as shown in Fig. 3, five radially extending secondary vane grooves 29a are formed on an outer circumferential surface of the second rotor 29 also, and a secondary vane 37 is housed in each secondary vane groove 29a to be able to project and retract. A space between the bottom surface of each secondary vane 37 and a corresponding secondary vane groove 29a is a secondary backpressure chamber 39. Five secondary compression chambers 41 are each formed by two adjacent secondary vanes 37, 37, the outer circumferential surface of the second rotor 29, the inner circumferential surface of the second cylinder block 7, the back surface of the second side plate cover 13c, and the front surface of the third side plate 15.

The first rotor 27 and the second rotor 29 are identical components. Further, the primary vanes 31 and the secondary vanes 37 are identical components. These are employed in the single-cylinder type vane compressor.

As shown in Fig. 1, a suction chamber 43 is formed between the front housing member 1 and the first side plate 11. A suction inlet 1b for connecting the suction chamber 43 to the outside is opened upward in the front housing member 1. Two suction holes 11b for communicating with the suction chamber 43 are formed to extend through the first side plate 11, and each of the suction holes 11b communicates with a corresponding suction space 5b of the first cylinder block 5. As shown in Fig. 2, the respective suction spaces 5b are configured to communicate with the primary compression chambers 35 in a suction process through suction ports 5c.

Further, two discharge spaces 5d are formed between the first cylinder block 5 and the rear housing member 3. The compression chambers 35 in a discharge process and the respective discharge spaces 5d are connected through discharge ports 5e. A discharge valve 45, which closes the discharge port 5e, and a retainer 47, which restricts the lifting amount of the discharge valve 45, are provided in each discharge space 5d. Components such as the drive shaft 25, the first cylinder block 5, the first rotor 27, the respective primary vanes 31, the discharge valves 45, and the retainers 47 configure a first compression mechanism 1C.

As shown in Fig. 1, two suction inlets 13d for communicating with the respective suction spaces 5b of the first cylinder block 5 are formed to extend through the second side plate 13, and the suction inlets 13d communicate with suction spaces 7b of the second cylinder block 7, respectively. As shown in Fig. 3, the respective suction spaces 7b are configured to communicate with the secondary compression chambers 41 in the suction process through suction ports 7c.

Further, as shown in Fig. 1, two discharge holes 13e for communicating with the respective discharge spaces 5d are formed to extend through the second side plate 13. Further, two discharge spaces 7d are formed between the second cylinder block 7 and the rear housing member 3. The discharge holes 13e communicate with the discharge spaces 7d, respectively. As shown in Fig. 3, the compression chambers 41 in the discharge process and the respective discharge spaces 7d communicate with each other through discharge ports 7e. A discharge valve 49, which closes the discharge port 7e, and a retainer 51, which restricts the lifting amount of the discharge valve 49, are provided in each discharge space 7d. A second compression mechanism 2C is configured by components such as the drive shaft 25, the second cylinder block 7, the second rotor 29, the respective secondary vanes 37, the discharge valve 49, and the retainer 51.

As shown in Fig. 1, two discharge holes 15b for communicating with the respective discharge spaces 7d are formed to extend through the third side plate 15. Further, a discharge chamber 53 is formed between the third side plate 15 and the rear housing member 3. In the discharge chamber 53, a centrifugal-type separator 55 is fixed by being sandwiched by the third side plate 15 and the rear housing member 3. The separator 55 is configured by an end frame 57 and a cylindrical member 59 fixed in the end frame 57 and extending in the up-down direction.

As shown in Fig. 4, at the center of the outer surface of the third side plate 15, a bulging section 15p bulging rearward with a certain thickness is formed. The bulging section 15p is configured by a boss section 15e bulging rearward around the drive shaft 25 and the slide bearing 21, a step 15f extending in leftward and rightward directions with less thickness than the boss section 15e, and a pendent section 15g extending downward with the same thickness as the step 15f. Two discharge grooves 15h, 15i are formed in the step 15f to extend leftward and rightward from the vicinity of the upper center toward each lower outer side. Discharge holes 15j, 15k for communicating with the respective discharge holes 15b are formed to extend through lower ends of the discharge grooves 15h, 15i.

As shown in Figs. 1 and 6, an oil separation chamber 57a extending in the up-down direction in a columnar shape is formed in the end frame 57. The cylindrical member 59 is press-fitted into the upper end of the oil separation chamber 57a. Due to this, a part of the oil separation chamber 57a functions as a guiding surface 57b that swirls the refrigerant gas around an outer circumferential surface of the cylindrical member 59. The discharge holes 15j, 15k open to the space between the cylindrical member 59 and the guiding surface 57b. Further, a communication hole 57c is formed at the lower end of the end frame 57, which allows communication of the bottom surface of the oil separation chamber 57a with the discharge chamber 53. Further, a supply chamber 17f is formed in the end frame 57, which houses the boss section 15e of the third side plate 15 together with the drive shaft 25 and the slide bearing 21.

As shown in Figs. 1, 2, and 5, a pair of oil drain grooves 13f of sectoral shape is formed at a front surface of the second side plate main body 13b. The respective drain grooves 13f are configured to communicate with primary backpressure chambers 33 in the suction process and the like by rotation of the first rotor 27. Further, as shown in Figs. 1 and 5, a valve chamber 13g for communicating between the discharge holes 13e and the respective drain grooves 13f is formed to extend through the second side plate main body 13b, and a ball-shaped valve body 61 is housed in the valve chamber 13g. The valve body 61 is urged in a direction that opens the valve chamber 13g by a spring 63 housed in the valve chamber 13g. The valve body 61 is prevented from escaping by the second side plate cover 13c. The drain grooves 13f, the valve chamber 13g, the valve body 61, and the spring 63 configure a first chatter preventing valve 73, which prevents chattering of the first compression mechanism 1C.

As shown in Figs. 1, 3, 4, and 6, a pair of oil drain grooves 15m of sectoral shape is formed at a front surface of the third side plate 15 also. The respective drain grooves 15m are configured to communicate with secondary backpressure chambers 39 in the suction process and the like by a rotation of the second rotor 29. Further, as shown in Figs. 1 and 6, a valve chamber 15n for communicating between the discharge chamber 53 and the respective oil drain grooves 15m is formed to extend through the third side plate 15, and a ball-shaped valve body 65 is housed in the valve chamber 15n also. The valve body 65 is urged in a direction that opens the valve chamber 15n by a spring 67 housed in the valve chamber 15n. The valve body 65 is prevented from escaping by the end frame 57 of the separator 55. The drain grooves 15m, the valve chamber 15n, the valve body 65, and the spring 67 configure a second chatter preventing valve 75, which prevents chattering of the second compression mechanism 2C.

The first rotor 27 and the second rotor 29 are fixed to the drive shaft 25 such that the primary and secondary vane grooves 27a, 29a, the drain grooves 13f, 15m, and the first and second chatter preventing valves 73, 75 have the same phase.

As shown in Figs. 1, 4, and 6, a first passage 15q, which extends upward from the lower end, is formed to extend through the pendent section 15g of the third side plate 15.

The lower end of the first passage 15q communicates with the discharge chamber 53. The upper end of the first passage 15q communicates with a second passage 15r, which extends horizontally to the supply chamber 17f of the end frame 57.

As shown in Figs. 1, 5, and 6, an axial hole 25a extending in the axial direction (the longitudinal direction of the drive shaft 25) from the rear end, a primary radial hole 25b extending in a radial direction from the front end of the axial hole 25a to the circumferential surface, and a secondary radial hole 25c extending from the midst of the axial hole 25a to the circumferential surface in the radial direction are formed in the drive shaft 25. As shown in Figs. 7 to 10, the primary and secondary radial holes 25b, 25c open at the positions of the diameter of the drive shaft 25. The axial hole 25a and the primary radial hole 25b form the first rotation path extending from the supply chamber 17f to the inner surface of the slide bearing 19. The axial hole 25a and the secondary radial hole 25c form a second rotation path extending from the supply chamber 17f to the inner surface of the slide bearing 21. The first passage 15q, the second passage 15r, the supply chamber 17f, and a part of the axial hole 25a up to the branching position into the secondary radial hole 25c form a common passage.

Further, as shown in Figs. 5, 7, and 9, two primary through holes 19a are formed to extend through the slide bearing 19 in the radial direction. Communication of the primary through holes 19a with the primary radial hole 25b is allowed as shown in Fig. 7 and interrupted as shown in Fig. 9 according to the phase of the drive shaft 25 in the rotation direction. Further, as shown in Figs. 5, 7, and 9, a primary annular groove 13i, which surrounds the axial hole 13a, is formed in the second side plate main body 13b. The primary through holes 19a and the primary annular groove 13i are connected with each other. Further, two primary back pressure holes 13j for communicating with the primary annular groove 13i extend forward in the axial direction in the second side plate main body 13b. Each of the primary back pressure holes 13j is configured to communicate with the primary backpressure chambers 33 in the compression process and the discharge process by the rotation of the first rotor 27. The primary radial hole 25b, the primary through holes 19a, the primary annular groove 13i and the primary back pressure holes 13j form a first supplying passage. The primary through holes 19a, the primary annular groove 13i and the primary back pressure holes 13j form a first intermittent mechanism.

Further, as shown in Figs. 6, 8 and 10, two secondary through holes 21a are formed to extend through the slide bearing 21 in the radial direction. The secondary through holes 21a are connected to and disconnected from the secondary radial hole 25c as shown in Figs. 8 and 10 according to the phase of the drive shaft 25 in the rotation direction. Further, as shown in Figs. 6, 8, and 10, a secondary annular groove 15s, which surrounds the axial hole 15a, is formed in the third side plate 15. Further, two secondary back pressure holes 15t for communicating with the secondary annular groove 15s extend forward in the axial direction in the third side plate 15. Each of the secondary back pressure holes 15t is configured to communicate with the secondary backpressure chambers 39 in the compression process and the discharge process by the rotation of the second rotor 29. The secondary radial hole 25c, the secondary through holes 21a, the secondary annular groove 15s, and the secondary back pressure holes 15t form a second supplying passage. The secondary through holes 21a, the secondary annular groove 15s, and the secondary back pressure holes 15t form a second intermittent mechanism.

The primary radial hole 25b and the secondary radial hole 25c are parallel to each other and have an identical inner diameter, with which the primary through holes 19a and the secondary through holes 21a have an identical inner diameter. While the first intermittent mechanism interrupts communication of the first rotation path with the discharge chamber 53 and the primary backpressure chambers 33, the second intermittent mechanism interrupts communication of the second rotation path with the discharge chamber 53 and the secondary backpressure chambers 39. Notably, according to requirements in machining and the like, the primary annular groove 13i and the secondary annular groove 15s have an identical inner diameter, and in addition, the primary back pressure holes 13j and the secondary back pressure holes 15t have an identical inner diameter.

Further, as shown in Fig. 1, a discharge outlet 3a for connecting the upper end of the discharge chamber 53 to the outside is formed in the rear housing member 3. The discharge outlet 3a is positioned above the cylindrical member 59.

The drive shaft 25, the slide bearings 17, 19, 21, the first side plate 11, the first cylinder block 5, the respective primary vanes 31, the discharge valves 45, the retainers 47, the second side plate 13, the first chatter preventing valve 73, the second cylinder block 7, the respective secondary vanes 37, the discharge valve 49, the retainer 51, the third side plate 15, the second chatter preventing valve 75, and the separator 55 are assembled as a sub assembly SA.

An O-ring is attached to the sub assembly SA, and the assembled product is inserted into the rear housing member 3. Next, an O-ring is attached to the rear housing member 3, and the front housing member 1 is covered thereon. Further, a plurality of bolts 71 shown in Figs. 2 and 3 is tightened. Accordingly, the tandem type vane compressor of the first embodiment is assembled.

Although not shown, in this tandem type vane compressor, the discharge outlet 3a is connected to a condenser by a pipe, the condenser is connected to an expansion valve by a pipe, the expansion valve is connected to the evaporator by a pipe, and the evaporator is connected to the suction inlet 1b by a pipe. The tandem type vane compressor, the condenser, the expansion valve, the evaporator, and the pipes form a refrigeration circuit. The refrigeration circuit is part of an air conditioning device for a vehicle.

In the tandem type vane compressor, when the drive shaft 25 is driven by an engine and the like, the first and second compression mechanisms 1C, 2C respectively repeat the suction process, the compression process, and the discharge process.

That is, the first and second rotors 27, 29 rotate in synchrony with the drive shaft 25, and a volume change is generated in the primary and secondary compression chambers 35, 41. Due to this, the refrigerant gas from the evaporator is drawn into the suction chamber 43 through the suction inlet 1b. The refrigerant gas in the suction chamber 43 is drawn into the primary compression chambers 35 via the suction holes 11b, the suction spaces 5b, and the suction ports 5c. Further, the refrigerant gas in the suction spaces 5b is drawn into the secondary compression chambers 41 via the suction inlets 13d, the suction spaces 7b, and the suction ports 7c.

Then, the refrigerant gas that has been compressed in the primary compression chambers 35 is discharged to the discharge spaces 5d via the discharge ports 5e. The high-pressure refrigerant gas in the discharge spaces 5d reaches the discharge spaces 7d via the discharge holes 13e. Further, the refrigerant gas that has been compressed in the secondary compression chambers 41 is discharged to the discharge spaces 7d via the discharge ports 7e. The high-pressure refrigerant gas in the discharge spaces 7d is discharged toward the guiding surface 57b of the separator 50 via the discharge holes 15b, 15j, 15k. Due to this, the refrigerant gas is circulated on the guiding surface 57b, and thereby the lubricant oil is centrifugally separated. Then, the refrigerant gas from which the lubricant oil has been separated is discharged toward the condenser from the discharge outlet 3a. Accordingly, in the tandem type vane compressor, a discharge amount per rotation of the drive shaft 25 is doubled compared to the single-cylinder type vane compressor.

Further, in the tandem type vane compressor, since respective ones of the primary and secondary vanes 31, 37 have a short axial length as employed in the single-cylinder type vane compressor, they are unlikely to be tilted with respect to the front-rear direction. Due to this, leakage of the refrigerant gas from the primary and secondary compression chambers 35, 41 is small, and the sliding characteristics of the respective ones of the primary and secondary vanes 31, 37 are excellent.

The separated lubricant oil is stored in the discharge chamber 53 from the oil separation chamber 57a via the communication hole 57c. The lubricant oil in the discharge chamber 53 is supplied to the axial hole 25a via the first passage 15q, the second passage 15r, and the supply chamber 17f, due to the discharge chamber 53 being under a high pressure.

Further, as shown in Fig. 7, when the primary radial holes 25b communicate with the primary through holes 19a by the phase of the drive shaft 25 in the rotation direction, the high-pressure lubricant oil in the axial hole 25a and the primary radial holes 25b is supplied to the respective primary backpressure chambers 33 shown in Fig. 2 via the respective primary through holes 19a, the primary annular groove 13i, and the primary back pressure holes 13j. Especially, in the tandem type vane compressor, since the single primary annular groove 13i and the respective primary backpressure chambers 33 are connected through the respective primary back pressure holes 13j, the high-pressure lubricant oil existing in the discharge chamber 53 can easily be supplied evenly to the respective primary backpressure chambers 33.

Further, as shown in Fig. 9, when communications of the primary radial holes 25b with the primary through holes 19a are interrupted by the phase of the drive shaft 25, the high-pressure lubricant oil in the axial hole 25a and the primary radial holes 25b is not supplied to the respective primary backpressure chambers 33 via the respective primary through holes 19a, the primary annular groove 13i, and the primary back pressure holes 13j.

Due to this, during the compression process, the high-pressure lubricant oil is intermittently supplied to the respective primary backpressure chambers 33. Due to this, the respective primary vanes 31 are intermittently pressed against the inner surface of the first cylinder chamber 5a. Due to this, the respective primary vanes 31 are lubricated within the primary vane grooves 27a, preventing chatter, and the leakage of the refrigerant gas from the primary compression chambers 35 is prevented, improving the efficiency.

Further, by intermittently supplying the lubricant oil to the primary backpressure chambers 33, a backpressure of the respective primary vanes 31 can be adjusted by adjusting a supply amount of the backpressure. Due to this, pressing force on the respective primary vanes 31 can be reduced, and thereby the power during operation can be reduced.

On the other hand, as shown in Fig. 8, when the secondary radial holes 25c communicate with the secondary through holes 21a by the phase of the drive shaft 25 in the rotation direction, the high-pressure lubricant oil within the axial hole 25a and the secondary radial holes 25c is supplied to the respective secondary backpressure chambers 39 shown in Fig. 3 via the respective secondary through holes 21a, the secondary annular groove 15s, and the respective secondary back pressure holes 15t. Especially, in the tandem type vane compressor, since the single secondary annular groove 15s and the respective secondary backpressure chambers 39 are communicated through the respective secondary back pressure holes 15t, the high-pressure lubricant oil existing in the discharge chamber 53 tends to be supplied evenly to the respective secondary backpressure chambers 39.

Further, as shown in Fig. 10, when communication of the secondary radial holes 25c with the secondary through holes 21a is interrupted by the phase of the drive shaft 25 in the rotation direction, the high-pressure lubricant oil within the axial hole 25a and the secondary radial holes 25c is not supplied to the respective secondary backpressure chambers 39 via the respective secondary through holes 21a, the secondary annular groove 15s, and the respective secondary back pressure holes 15t.

Due to this, during the compression process, the high-pressure lubricant oil is intermittently supplied to the respective secondary backpressure chambers 39. Due to this, the respective secondary vanes 37 are intermittently pressed against the inner surface of the second cylinder chamber 7a. Due to this, the respective secondary vanes 37 are lubricated within the secondary vane grooves 29a, whereby chatter is prevented, and the leakage of the refrigerant gas from the secondary compression chambers 41 is prevented, whereby the efficiency is improved.

Due to this, high mechanical efficiency is reliably exhibited in the tandem type vane compressor. Further, in the tandem type vane compressor, each primary backpressure chamber 33 and each secondary backpressure chamber 39 do not need to be connected to the discharge chamber 53 separately. This reduces production costs.

Further, in the tandem type vane compressor, by intermittently supplying the lubricant oil to the secondary backpressure chambers 39, a supply amount of the backpressure can be adjusted, and thereby a backpressure of the respective secondary vanes 37 can be adjusted. Due to this, pressing force on the respective secondary vanes 37 is reduced, and the power during operation is reduced.

With the rotation of the drive shaft 25 being stopped, in a state in which communication between the primary radial holes 25b and the primary through holes 19a is interrupted and also communication between the secondary radial holes 25c and the secondary through holes 21a are interrupted, backflow of the refrigerant gas and the like and reverse rotation of the drive shaft 25 are not caused. As shown in Figs. 7 and 8, even with the rotation of the drive shaft 25 being stopped in a state in which the primary radial holes 25b communicate with the primary through holes 19a and the secondary radial holes 25c communicate with the secondary through holes 21a, if the backflow of the refrigerant gas and the like and the reverse rotation of the drive shaft 25 are caused even at the slightest degree, the state shown in Figs. 9 and 10 is assumed. Due to this, the phase of the drive shaft 25 is displaced, whereby immediately communication of the primary radial holes 25b with the primary through holes 19a and also communication of the secondary radial holes 25c with the secondary through holes 21a are interrupted, and the backflow of the refrigerant gas and the like and the reverse rotation of the drive shaft 25 are not caused. Due to this, the tandem type vane compressor can reliably and quickly prevent the backflow of the refrigerant gas and the like and the reverse rotation of the drive shaft 25.

In the tandem type vane compressor, the axial hole 25a and the primary and secondary radial holes 25b, 25c are formed in the drive shaft 25. Due to this, spaces for an on-off valve and a check valve do not need to be provided within the tandem type vane compressor as has conventionally been necessary, and the size of the tandem type vane compressor is not increased. Further, since the axial hole 25a and the primary and secondary radial holes 25b, 25c can easily be formed on the drive shaft 25, the burden of developing a large number of different varieties depending on vehicles and the like can also be omitted.

Notably, the lubricant oil supplied to the respective primary backpressure chambers 33 contributes for lubricating sliding between the primary vanes 31 and the primary vane grooves 27a, sliding between the first rotor 27 and each of the first side plate 11 and the second side plate main body 13b, sliding between the slide bearings 17, 19 and the drive shaft 25, and the like. Further, the lubricant oil supplied to the respective secondary backpressure chambers 39 contributes for lubricating sliding between the secondary vanes 37 and the secondary vane grooves 29a, sliding between the second rotor 29 and each of the second side plate cover 13c and the third side plate 15, sliding between the slide bearings 19, 21 and the drive shaft 25, and the like.

Thus, with the above described tandem type vane compressor, the discharge amount per rotation of the drive shaft 25 is increased, and the efficiency and size characteristics (compactness) are excellent. Further, according to the tandem type vane compressor, the backflow of the refrigerant gas and the like and the reverse rotation of the drive shaft can be reduced. Further, the above described tandem type vane compressors have the same shell diameter as the single-cylinder type vane compressors, thus the mounting characteristics are excellent.

Further, in the tandem type vane compressor, a common component is used respectively for the first cylinder block 5 and the second cylinder block 7, for the first rotor 27 and the second rotor 29, and for the primary vanes 31 and the secondary vanes 37. The component commonality reduces production costs.

(Second Embodiment)

As shown in Fig. 11, a tandem type vane compressor of the second embodiment has a slide bearing 19 press-fitted into an axial hole 13a of a second side plate 13 from the front side. As shown in Fig. 13, an axial hole 25a communicates with a primary radial hole 25d extending in a radial direction from the front end of the axial hole 25a to the circumferential surface. The axial hole 25a and the primary radial hole 25d form a first rotation path.

Further, three primary axial grooves 13k for communicating with a primary annular groove 13i are formed on a second side plate main body 13b. As shown in Fig. 11, the respective primary axial grooves 13k extend rearward in the axial direction, and are opened to axial holes 13a. The length from the front end surface of the second side plate 13 to the primary radial hole 25d is longer than the length from the rear end surface of the second side plate 13 to the primary radial hole 25d.

Further, as shown in Fig. 12, a slide bearing 21 is press-fitted into an axial hole 15a of a third side plate 15. As shown in Fig. 14, the axial hole 25a communicates with a secondary radial hole 25e extending from the middle of the axial hole 25a to the circumferential surface in the radial direction. The axial hole 25a and the secondary radial hole 25e form a second rotation path.

Further, three secondary axial grooves 15u for communicating with a secondary annular groove 15s are formed on the third side plate 15. As shown in Fig. 12, the respective secondary axial grooves 15u extend rearward in the axial direction, and are opened to the axial holes 15a. The length from the front end surface of the third side plate 15 to the secondary radial hole 25e is longer than the length from the rear end surface of the third side plate 15 to the secondary radial hole 25e. Other configurations are similar to the first embodiment.

In the tandem type vane compressor, since the length from the front end surface of the third side plate 15 to the secondary radial hole 25e is longer than the length from the rear end surface of the third side plate 15 to the secondary radial hole 25e, a long sealing length is provided on the inner surface of the slide bearing 21, so high-pressure lubricant oil is easily supplied to secondary backpressure chambers 39 at a required timing from a supply chamber 17f on the rear end surface side of the third side plate 15. Secondary vanes 37 are more reliably intermittently pressed against the inner surface of a second cylinder chamber 7a, and thereby efficiency is improved.

Further, since the length from the front end surface of the second side plate 13 to the primary radial hole 25d is longer than the length from the rear end surface of the second side plate 13 to the primary radial hole 25d, a long sealing length is provided on the inner surface of the slide bearing 19. Accordingly, the lubricant oil, which is supplied to the secondary backpressure chambers 39 at a required timing, tends to be supplied at a required timing even via the primary axial grooves 13k, the primary annular groove 13i, and primary back pressure holes 13j. Due to this, primary vanes 31 are more reliably intermittently pressed against the inner surface of a first cylinder chamber 5a, and thereby the efficiency is improved.

Further, by configuring the primary and secondary radial holes 25d, 25e as above, the weight is reduced by positioning the primary annular groove 13i and the secondary annular groove 15s relatively forward, and employing a boss section 15e with a small diameter, and advantages of the invention are thereby achieved. Other advantages are similar to the first embodiment.

(Third Embodiment)

As shown in Fig. 15, a vane compressor of the third embodiment does not employ slide bearings 17, 19, 21 as in the first and second embodiments. A second side plate 13 and a drive shaft 25 make contact with each other via a cylindrical sliding surface 73a. Further, two primary recessed portions 13m are formed on the sliding surface 73a of the second side plate 13. The primary recessed portions 13m are positioned with a distance from the rear end of the drive shaft 25 equal to a primary radial hole 25b. The primary recessed portions 13m open to the sliding surface 73a at the positions passing through an axial center of the drive shaft 25. Further, two primary back pressure holes 13n extending from the respective primary recessed portions 13m to the rear end surface of a first rotor 27 are formed in the second side plate 13.

Further, as shown in Fig. 16, a third side plate 15 and the drive shaft 25 make contact with each other via a cylindrical sliding surface 73b. Further, two secondary recessed portions 15v are formed on the sliding surface 73b of the third side plate 15. The secondary recessed portions 15v are positioned with a distance from the rear end of the drive shaft 25 equal to a secondary radial hole 25c. The secondary recessed portions 15v open to the sliding surface 73b at the positions passing through the axial center of the drive shaft 25. Further, two secondary back pressure holes 15w extending from the respective secondary recessed portions 15v to the rear end surface of a second rotor 29 are formed in the third side plate 15.

In the tandem type vane compressor, when the primary radial hole 25b communicates with the primary recessed portions 13m by the phase of the drive shaft 25, high-pressure lubricant oil in the axial hole 25a and the primary radial hole 25b is supplied to respective primary backpressure chambers 33 via the primary back pressure holes 13n. Further, when the communication of the primary radial hole 25b with the primary recessed portions 13m is interrupted by the phase of the drive shaft 25, the high-pressure lubricant oil in the axial hole 25a and the primary radial hole 25b is not supplied to the respective primary backpressure chambers 33 via the primary back pressure holes 13n.

On the other hand, when the secondary radial hole 25c communicates with the secondary recessed portions 15v by the phase of the drive shaft 25, the high-pressure lubricant oil in the axial hole 25a and the secondary radial hole 25c is supplied to respective secondary backpressure chambers 39 via the secondary back pressure holes 15w. Further, when the communication of the secondary radial hole 25c with the secondary recessed portions 15v is interrupted by the phase of the drive shaft 25, the high-pressure lubricant oil in the axial hole 25a and the secondary radial hole 25c is not supplied to the respective secondary backpressure chambers 39 via the secondary back pressure holes 15w.

Accordingly, by making the first to third side plates 11, 13, 15 respectively contact with the drive shaft 25, it is possible to reduce the number of components, thereby reducing the assembly steps and facilities. This further reduces the production cost.

(Fourth Embodiment)

In a tandem type vane compressor of the fourth embodiment, as shown in Fig. 17, two primary through holes 17a are formed to extend through a slide bearing 17 in a radial direction. Further, a primary annular groove 11i that surrounds an axial hole 11a is formed in a first side plate 11. The primary through holes 17a and the primary annular groove 11i are connected with each other. Further, in the first side plate 11, two primary back pressure holes 11j for communicating with the primary annular groove 11i extend toward the rear side in the axial direction. Each of the primary back pressure holes 11j is configured to be connected to primary backpressure chambers 33 in a compression process and a discharge process by a rotation of a first rotor 27.

Further, similar to the first embodiment, in a slide bearing 21 and a third side plate 15, secondary through holes 21a, a secondary annular groove 15s, and secondary back pressure holes 15t are formed.

In a drive shaft 25, an axial hole 25f is formed to extend in the axial direction (the longitudinal direction of the drive shaft 25) from the rear end toward the front side at a length longer than the first embodiment. The axial hole 25f also communicates with a supply chamber 17f. Further, in the drive shaft 25, a primary radial hole 25g is formed to extend from the distal end of the axial hole 25f in the radial direction. Further, similar to the first embodiment, a secondary radial hole 25c is formed in the drive shaft 25. A first passage 15q, a second passage 15r, the supply chamber 17f, and a part of the axial hole 25f up to the branching position into the secondary radial hole 25c form a common passage. The primary radial hole 25g, the primary through holes 17a, the primary annular groove 11i and the primary back pressure holes 11j form a first supplying passage. Further, the secondary radial hole 25c, secondary through holes 21a, a secondary annular groove 15s, and secondary back pressure holes 15t form a second supplying passage.

Other configurations are similar to the first embodiment. Due to this, same reference numerals as the first embodiment will be given to components similar to the first embodiment, and detailed descriptions thereof will be omitted.

communicating with the primary annular groove 11i extend toward the rear side in the axial direction. Each of the primary back pressure holes 11j is configured to be connected to primary backpressure chambers 33 in a compression process and a discharge process by a rotation of a first rotor 27.

Further, similar to the first embodiment, in a slide bearing 21 and a third side plate 15, secondary through holes 21a, a secondary annular groove 15s, and secondary back pressure holes 15t are formed.

In a drive shaft 25, an axial hole 25f is formed to extend in the axial direction (the longitudinal direction of the drive shaft 25) from the rear end toward the front side at a length longer than the first embodiment. The axial hole 25f also communicates with a supply chamber 17f. Further, in the drive shaft 25, a primary radial hole 25g is formed to extend from the distal end of the axial hole 25f in the radial direction. Further, similar to the first embodiment, a secondary radial hole 25c is formed in the drive shaft 25. A first passage 15q, a second passage 15r, the supply chamber 17f, and a part of the axial hole 25f up to the branching position into the secondary radial hole 25c form a common passage. The primary radial hole 25g, the primary through holes 17a, the primary annular groove 11i and the primary back pressure holes 11j form a first supplying passage. Further, the secondary radial hole 25c, secondary through holes 21a, a secondary annular groove 15s, and secondary back pressure holes 15t form a second supplying passage.

Other configurations are similar to the first embodiment. Due to this, same reference numerals as the first embodiment will be given to components similar to the first embodiment, and detailed descriptions thereof will be omitted.

In this tandem type vane compressor also, similar advantages as the first embodiment are achieved.

(Fifth Embodiment)

In a tandem type vane compressor of the fifth embodiment, as shown in Fig. 18, two secondary through holes 19b are formed to extend through a slide bearing 19 in a radial direction. Further, a secondary annular groove 13p that surrounds an axial hole 13a is formed in a second side plate 13. The secondary through holes 19b and the secondary annular groove 13p are connected with each other. Further, in the second side plate 13, two secondary back pressure holes 13q for communicating with the secondary annular groove 13p extends toward the rear side in the axial direction. Each of the secondary back pressure holes 13q are configured to communicate with secondary backpressure chambers 39 in a compression process and a discharge process by a rotation of a second rotor 29.

Further, similar to the fourth embodiment, two primary through holes 17a are formed to extend through a slide bearing 17 in the radial direction. Further, in a first side plate 11, a primary annular groove 11i and primary back pressure holes 11j are formed.

Further, similar to the fourth embodiment, an axial hole 25f is formed in the drive shaft 25. The axial hole 25f extends in the longitudinal direction of the drive shaft 25. Further, similar to the fourth embodiment, a primary radial hole 25g is formed in the drive shaft 25. Further, in the drive shaft 25, a secondary radial hole 25h is formed to extend in the radial direction from the middle of the axial hole 25f. A first passage 15q, a second passage 15r, a supply chamber 17f, and a part of the axial hole 25f up to the branching position into the secondary radial hole 25h form a common passage. The primary radial hole 25g, the primary through holes 17a, the primary annular groove 11i and the primary back pressure holes 11j form a first supplying passage. Further, the secondary radial hole 25h, secondary through holes 19b, a secondary annular groove 13p, and secondary back pressure holes 13q form a second supplying passage.

Other configurations are similar to the first and fourth embodiments. Due to this, same reference numerals as the first and fourth embodiments will be given to components similar to the first and fourth embodiments, and detailed descriptions thereof will be omitted.

In this tandem type vane compressor also, similar advantages as the first embodiment are achieved.

(Sixth Embodiment)

In a tandem type vane compressor of the sixth embodiment, as shown in Fig. 19, two through holes 19c are formed to extend through a slide bearing 19 in a radial direction. Further, an annular groove 13r that surrounds an axial hole 13a is formed in a second side plate 13. The through holes 19c and the annular groove 13r are connected with each other. Further, in the second side plate 13, two primary back pressure holes 13s for communicating with the annular groove 13r extend frontward in the axial direction, and two secondary back pressure holes 13t for communicating with the annular groove 13r extend toward the rear side in the axial direction. Each of the primary back pressure holes 13s is configured to communicate with primary backpressure chambers 33 in a compression process and a discharge process by a rotation of a first rotor 27. Each of the secondary back pressure holes 13t is configured to be connected to secondary backpressure chambers 39 in the compression process and the discharge process by a rotation of a second rotor 29.

Further, similar to the first embodiment, an axial hole 25a is formed in a drive shaft 25. The axial hole 25a extends in the longitudinal direction of the drive shaft 25. Further, in the drive shaft 25, a radial hole 25i is formed to extend from the front end of the axial hole 25a in the radial direction. A first passage 15q, a second passage 15r, a supply chamber 17f, and the axial hole 25a form a common passage. A radial hole 25i, the through holes 19c, the annular groove 13r, and the primary back pressure holes 13s form a first supplying passage. Further, the radial hole 25i, through holes 19c, the annular groove 13r, and secondary back pressure holes 13t form a second supplying passage.

Other configurations are similar to the first embodiment. Due to this, same reference numerals as the first embodiment will be given to components similar to the first embodiment, and detailed descriptions thereof will be omitted.

In this tandem type vane compressor also, similar advantages as the first embodiment are achieved. Note that, in the tandem type vane compressor, the radial hole 25i, the through holes 19c, and the annular groove 13r can be regarded as a common passage.

(Seventh Embodiment)

In a tandem type vane compressor of the seventh embodiment, as shown in Figs. 20 and 21, a drive shaft 26 is configured by a first drive shaft 28, which drives a first compression mechanism 1C, and a second drive shaft 30, which is fitted into the first drive shaft 28 and drives a second compression mechanism 2C.

As shown in Fig. 22, a housing chamber 28a having a rectangular cross section is formed at the rear end of the first drive shaft 28. On the other hand, at the distal end of the second drive shaft 30, a protruding section 30a having an identical width as the housing chamber 28a and a vertical length slightly smaller than the housing chamber 28a is formed. Further, in the drive shaft 28, an axial hole 28b extending in the axial direction (the longitudinal direction of the drive shaft 25) and a primary radial hole 25g extending from the distal end of the axial hole 28b in a radial direction are formed. Further, in the second drive shaft 30, an axial hole 30b extending in the axial direction (the longitudinal direction of the drive shaft 25) and a secondary radial hole 25c extending from the middle of the axial hole 30b in the radial direction are formed. The axial hole 28b and the axial hole 30b are connected with each other by fitting the protruding section 30a into the housing chamber 28a, and provide a similar configuration to the axial hole 25f in the fourth embodiment. Other configurations are similar to the fourth embodiment.

In assembling the tandem type vane compressor, firstly, as shown in Fig. 20, a first rotor 27 is press-fitted to the first drive shaft 28, then is inserted in a first cylinder block 5, and a first side plate 11 and finally a second side plate 13 are assembled, providing a first sub assembly 1SA. Further, a second rotor 29 is press-fitted to the second drive shaft 30, then is inserted in a second cylinder block 7, and finally a third side plate 15 is assembled, providing a second sub assembly 2SA. Then, by fitting the protruding section 30a into the housing chamber 28a, the second sub assembly 2SA is fitted into the first sub assembly 1SA. Accordingly, a sub assembly SA ready to be inserted into a shell 9 is obtained as shown in Fig. 21.

An O-ring is attached to the sub assembly SA, which is inserted into a rear housing member 3. Next, an O-ring is attached to the rear housing member 3, and a front housing member 1 is covered thereon. Then, a plurality of bolts 71 is tightened. Accordingly, the tandem type vane compressor is assembled, similar to the fourth embodiment.

In the tandem type vane compressor, since the second side plate 13 does not need to be divided in the radial direction upon assembly, the first rotor 27 and the second rotor 29 can suitably slide with respect to the second side plate 13. Due to this, excellent assembly property and durability are reliably exhibited in the tandem type vane compressor. Other advantages are similar to the fourth embodiment. The tandem type vane compressors of the other embodiments can be similarly assembled as above.

Further, in the tandem type vane compressor of the seventh embodiment, as shown in Fig. 23, a columnar-shaped housing chamber 28c may be provided at the rear end of the housing chamber 28a of the first drive shaft 28, and a columnar-shaped shaft portion 30c may be provided at the rear side of the protruding section 30a of the second drive shaft 30. The shaft portion 30c has an O-ring groove 30d. In this case, after mounting an O-ring 30e in the O-ring groove 30d, by fitting the protruding section 30a and the shaft portion 30c to the housing chambers 28a, 28c, the first drive shaft 28 and the second drive shaft 30 can be easily ensured to be coaxially arranged.

In the above, the invention has been described with reference to the first to seventh embodiments, the invention is not limited to the above first to seventh embodiments, and can be adapted by being suitably modified within the scope of the invention.

For example, a third compression mechanism may be provided in addition to the first compression mechanism 1C and the second compression mechanism 2C.

Further, although the first compression mechanism 1C and the second compression mechanism 2C are operated under the same phase in the first to seventh embodiments, the first compression mechanism 1C and the second compression mechanism 2C may be operated under different phases depending on a purpose, such as reducing a discharge pulsation.

Further, the refrigerant gas compressed by the first compression mechanism 1C may be drawn into the second compression mechanism 2C and further compressed by the second compression mechanism 2C, providing multiple level compression.

Further, the common passages may be formed in the housing. Further, the first and second rotation paths may be formed in the rotator that rotates in synchrony with the drive shaft, instead of the drive shaft.

Therefore, the present examples and embodiments are to be considered as illustrative and not restrictive and the invention is not to be limited to the details given herein, but may be modified within the scope and equivalence of the appended claims.