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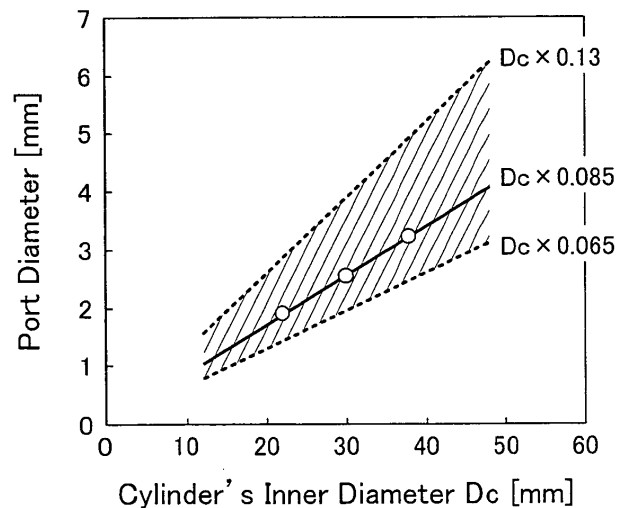
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(54) **EXPANDER**

(57) An inflow port and an outflow port are formed at a cylinder of an expander. The inner diameter of the cyl-

inder is D_c , and the diameter D_i of the inflow port is set to satisfy a relationship $0.065 \times D_c \leq D_i \leq 0.13 \times D_c$.

FIG.8



Description

TECHNICAL FIELD

[0001] The present invention relates to expanders which recover power of a fluid depressurized in a cylinder.

BACKGROUND ART

[0002] Expanders which recover power of a fluid have been known, and are applied in refrigeration devices such as air conditioners.

[0003] Patent Document 1 discloses a positive displacement expander of this type. The positive displacement expander is connected to a refrigerant circuit in which a refrigerant circulates to implement a refrigeration cycle, and is configured to perform an expansion process of the refrigeration cycle. The expander forms a so-called rotary fluid device, and includes a cylinder and a piston which rotates along an inner periphery surface of the cylinder. In the expander, a fluid chamber (expansion chamber) is formed between the cylinder and the piston.

[0004] In the expander, a refrigerant flows into the expansion chamber through an inflow port. When the refrigerant is depressurized in the expansion chamber, the power of the refrigerant is recovered as rotational power of the piston. The refrigerant depressurized in the expansion chamber flows out into the refrigerant circuit through an outflow port. As mentioned above, in the positive displacement expander of this type, the kinetic energy of the refrigerant is recovered as rotational power of the piston and a rotational shaft. The rotational power is used as power for driving a compressor, or a power source of a generator.

CITATION LIST

PATENT DOCUMENT

[0005]

PATENT DOCUMENT 1: Japanese Patent Publication No. H08-338356

SUMMARY OF THE INVENTION

TECHNICAL PROBLEM

[0006] However, the conventional rotary expander has a problem that the inflow port and the outflow port temporarily communicate with each other via the expansion chamber when the expansion chamber starts to communicate with the inflow port, that is, a blow-by phenomenon.

[0007] Specifically, as shown, for example, in FIG. 9, a piston (102) of a conventional rotary expander (100) rotates in a cylinder (101) as sequentially shown in FIG. 9(A), (B), (C), (D), (A), Here, a fluid having flowed into an expansion chamber (105) in the state of FIG. 9(A)

through an inflow port (103) is depressurized because the capacity of the expansion chamber (105) is increased as the piston (102) rotates as sequentially shown in FIG. 9(B) and (C). The power of the refrigerant is recovered at this time. The piston (102) further rotates, and when the piston reaches the rotational position as shown in FIG. 9(D) (the location of the top dead center of the cylinder that is closest to a bush (106)), the inflow port (103) and the outflow port (104) may communicate with each other via the expansion chamber (105). As a result, the above-mentioned blow-by phenomenon may temporarily occur in the state of FIG. 9(D), in which the power of refrigerant having flowed through the inflow port (103) is not recovered and the refrigerant flows out through the outflow port (104), thereby decreasing the power recovery efficiency of the expander.

[0008] The present invention was made in view of the above problems, and it is an objective of the invention to avoid a blow-by phenomenon of an expander and thereby increase the efficiency of the expander.

SOLUTION TO THE PROBLEM

[0009] The first aspect of the present invention is intended for an expander having a cylinder (71) at which an inflow port (81) and an outflow port (82) are formed, a piston (75) accommodated in the cylinder (71) so as to be eccentric to a rotational shaft (40), and a blade (76) which divides a fluid chamber (72) formed between the piston (75) and the cylinder (71) into a high pressure side and a low pressure side of a fluid, the expander recovering power of a fluid depressurized in the fluid chamber (72). In this expander, an inner diameter of the cylinder (71) is D_c [mm], and a diameter D_i [mm] of the inflow port (81) satisfies a relationship $0.065 \times D_c \leq D_i \leq 0.13 \times D_c$.

[0010] According to the first aspect of the present invention, a high pressure fluid flows into the fluid chamber (72) of the cylinder (71) through the inflow port (81). This fluid is depressurized in the fluid chamber (72) to a low pressure fluid. Here, the piston (75) is rotated by power of the fluid, and the power of the fluid is recovered as rotational power of the piston (75) and the rotational shaft (40). The low pressure fluid depressurized in the fluid chamber (72) flows out of the expander through the outflow port (82).

[0011] Here, $D_i \leq 0.13D_c$ stands in the present invention, where the diameter of the inflow port (81) is D_i , and the inner diameter of the cylinder (71) is D_c . In other words, the inflow port diameter D_i of the present invention is set to $0.13D_c$ or less. Thus, the inflow port diameter D_i is not too large according to the present invention, and it is thus possible to prevent the inflow port (81) and the outflow port (82) from communicating with each other via the fluid chamber (72) in the state of the conventional case shown in FIG. 9(D). As a result, according to the expander of the present invention, it is possible to avoid the occurrence of a co-called blow-by phenomenon, and possible to avoid a reduction in efficiency of the expander.

[0012] Further, $0.065D_c \leq D_i$ stands in the present invention. In other words, the inflow port diameter D_i of the present invention is set to $0.065D_c$ or more. Thus, the inflow port diameter D_i is not too small according to the present invention, and it is thus possible to avoid an increase in pressure loss in the inflow port (81). As a result, according to the expander of the present invention, it is also possible to avoid a reduction in power recovery efficiency of the expander with an increase in pressure loss. That is, according to the expander of the present invention, it is possible to increase the efficiency of the expander to a maximum, because the blow-by phenomenon can be avoided and the pressure loss in the inflow port (81) can be reduced.

[0013] The second aspect of the present invention is that, in the expander of the first aspect of the present invention, a density of a fluid in the inflow port (81) is ρ_i [kg/m³], and a density of a fluid in the outflow port (82) is ρ_o [kg/m³]; and a diameter D_o [mm] of the outflow port (82) satisfies a relationship $D_o = D_i \times (\rho_i/\rho_o)^2$

[0014] According to the second aspect of the present invention, a density of a fluid in the inflow port (81) is ρ_i , and a density of a fluid in the outflow port (82) is ρ_o ; and the diameter D_o of the outflow port (82) satisfies $D_o = D_i \times (\rho_i/\rho_o)^2$. Here, if the inflow port diameter D_i and the outflow port diameter D_o are set to have the same diameter, the diameter of the outflow port (82) may be too small and the pressure loss in the outflow port (82) may be slightly increased, because a fluid whose density is lower than the density of a fluid flowing in the inflow port (81) flows in the outflow port (82). In view of this, according to the present invention, a ratio of the density ρ_i of a fluid before depressurization in the fluid chamber (72) to the density ρ_o of the fluid after depressurization in the fluid chamber (72) (i.e., density ratio ρ_i/ρ_o) is considered, and the outflow port diameter D_o is decided by multiplying the inflow port diameter D_i by the square of the density ratio (ρ_i/ρ_o).

[0015] The third aspect of the present invention is intended for an expander having a cylinder (71) at which an inflow port (81) and an outflow port (82) are formed, a piston (75) accommodated in the cylinder (71) so as to be eccentric to a rotational shaft (40), and a blade (76) which divides a fluid chamber (72) formed between the piston (75) and the cylinder (71) into a high pressure side and a low pressure side of a fluid, the expander recovering power of a fluid depressurized in the fluid chamber (72). In this expander, an inner diameter of the cylinder (71) is D_c [mm], and a diameter D_o [mm] of the outflow port (82) satisfies a relationship $0.065 \times D_c \leq D_o \leq 0.13 \times D_c$.

[0016] The third aspect of the present invention is intended for the same expander as in the first aspect of the present invention. Here, $D_o \leq 0.13D_c$ stands in the present invention, where the diameter of the outflow port (82) is D_o , and the inner diameter of the cylinder (71) is D_c . In other words, the outflow port diameter of D_o of the present invention is set to $0.13D_c$ or less. Thus, the out-

flow port diameter D_o is not too large according to the present invention. Thus, it is possible to avoid the blow-by phenomenon which occurs in the conventional case shown in FIG. 9(D), and possible to avoid a reduction in efficiency of the expander.

[0017] Further, $0.065D_c \leq D_o$ stands in the present invention. In other words, the outflow port diameter D_o of the present invention is set to $0.065D_c$ or more. Thus, the outflow port diameter D_o is not too small according to the present invention, and it is thus possible to avoid an increase in pressure loss in the outflow port (82). As a result, according to the expander of the present invention, it is also possible to avoid a reduction in power recovery efficiency of the expander with an increase in pressure loss. That is, according to the expander of the present invention, it is possible to increase the efficiency of the expander to a maximum as in the first aspect of the present invention, because the blow-by phenomenon can be avoided and the pressure loss in the outflow port (82) can be reduced.

[0018] The fourth aspect of the present invention is that, in the expander of the third aspect of the present invention, a density of a fluid in the inflow port (81) is ρ_i [kg/m³], and a density of a fluid in the outflow port (82) is ρ_o [kg/m³]; and a diameter D_i [mm] of the inflow port (81) satisfies a relationship $D_i = D_o \times (\rho_o/\rho_i)^2$.

[0019] According to the fourth aspect of the present invention, a density of a fluid in the inflow port (81) is ρ_i , and a density of a fluid in the outflow port (82) is ρ_o ; and the diameter D_i of the inflow port (81) satisfies $D_i = D_o \times (\rho_o/\rho_i)^2$. Here, if the inflow port diameter D_i and the outflow port diameter D_o are set to have the same diameter, the diameter of the inflow port (81) may be larger than desired, because a fluid whose density is higher than the density of a fluid flowing in the outflow port (82) flows in the inflow port (81). In view of this, according to the present invention, a ratio of the density ρ_i of a fluid before depressurization in the fluid chamber (72) to the density ρ_o of a fluid after depressurization in the fluid chamber (72) (i.e., density ratio ρ_i/ρ_o) is considered, and the inflow port diameter D_i is decided by multiplying the outflow port diameter D_o by the square of the density ratio (ρ_o/ρ_i).

ADVANTAGES OF THE INVENTION

[0020] According to the first aspect of the present invention, the inner diameter of the cylinder (71) is D_c [mm], and the diameter D_i [mm] of the inflow port (81) satisfies a relationship $0.065 \times D_c \leq D_i \leq 0.13 \times D_c$. Therefore, according to the present invention, the inflow port diameter D_i is not too large, and thus, a so-called blow-by phenomenon can be avoided. Further, the inflow port diameter D_i is not too small, and thus, an increase in pressure loss in the inflow port (81) can be prevented. As a result, desired efficiency can be obtained according to the present invention, thereby making it possible to enhance the energy saving characteristics of a refrigeration

device or the like to which the expander is applied.

[0021] Similarly, according to the third aspect of the present invention, the diameter D_o [mm] of the outflow port (82) satisfies the relationship $0.065 \times D_c \leq D_o \leq 0.13 \times D_c$. Therefore, according to the present invention, the outflow port diameter D_o is not too large, and thus, a so-called blow-by phenomenon can be avoided. Further, the outflow port diameter D_o is not too small, and thus, an increase in pressure loss in the outflow port (82) can be prevented. As a result, desired efficiency can be obtained according to the present invention, thereby making it possible to enhance the energy saving characteristics of a refrigeration device or the like to which the expander is applied.

[0022] Further, according to the second and the fourth aspects of the present invention, a ratio of the density of a fluid in the inflow port (81) and the outflow port (82) is considered, thereby making it possible to obtain optimum diameters of the inflow port (81) and the outflow port (82) that can reduce pressure loss in the inflow port (81) and the outflow port (82).

BRIEF DESCRIPTION OF THE DRAWINGS

[0023]

[FIG. 1] FIG. 1 shows a piping diagram schematically illustrating a structure of an air conditioner according to an embodiment.

[FIG. 2] FIG. 2 shows a vertical cross-sectional view of a compressor-expander unit according to the embodiment.

[FIG. 3] FIG. 3 shows a horizontal cross-sectional view of an expansion mechanism according to the embodiment.

[FIG. 4] FIG. 4 shows an example of each design point of the expansion mechanism according to the embodiment.

[FIG. 5] FIG. 5 shows an example of each design dimension of the expansion mechanism according to the embodiment.

[FIG. 6] FIG. 6 shows horizontal cross-sectional views for explaining the movement of the expansion mechanism according to the embodiment. FIG. 6(A) shows the state when a rotational angle is 90° ; FIG. 6(B) shows the state when the rotational angle is 180° ; FIG. 6(C) shows the state when the rotational angle is 270° ; and FIG. 6(D) shows the state when the rotational angle is 360° (0°).

[FIG. 7] FIG. 7 is a graph showing a relationship between a port diameter and the efficiency of the expander, with respect to three types of expansion mechanisms having a different cylinder's inner diameter.

[FIG. 8] FIG. 8 is a graph showing a relationship between the cylinder's inner diameter and a port diameter, for achieving high efficiency of the expander.

[FIG. 9] FIG. 9 shows horizontal cross-sectional

views for explaining the movement of a conventional expansion mechanism. FIG. 6(A) shows the state when a rotational angle is 90° ; FIG. 6(B) shows the state when the rotational angle is 180° ; FIG. 6(C) shows the state when the rotational angle is 270° ; and FIG. 6(D) shows the state when the rotational angle is 360° (0°).

DESCRIPTION OF REFERENCE CHARACTERS

[0024]

60	expansion mechanism (expander)
71	cylinder
72	fluid chamber
73	high pressure chamber
74	low pressure chamber
75	piston
76	blade
81	inflow port
82	outflow port
D_c	cylinder's inner diameter
D_i	inflow port diameter
D_o	outflow port diameter
ρ_i	refrigerant density of an inflow side (high pressure side)
ρ_o	refrigerant density of an outflow side (low pressure side)

DESCRIPTION OF EMBODIMENTS

[0025] An embodiment of the present invention will be described in detail hereinafter, with reference to the drawings.

[0026] An expander according to the present invention is installed in a compressor-expander unit (30). The compressor-expander unit (30) is mounted in an air conditioner (10) which performs and switches between a cooling operation and a heating operation for a room air. The air conditioner (10) forms a refrigeration device having a refrigerant circuit (20) in which a refrigerant circulates to implement a refrigeration cycle.

[0027] As shown in FIG. 1, the air conditioner (10) includes an outdoor unit (11) placed outdoors and an indoor unit (13) placed indoors. An outdoor fan (12), an outdoor heat exchanger (23), a first four-way switching valve (21), a second four-way switching valve (22), and the compressor-expander unit (30) are accommodated in the outdoor unit (11). An indoor fan (14) and an indoor heat exchanger (24) are accommodated in the indoor unit (13). The outdoor unit (11) and the indoor unit (13) are connected to each other via a pair of connecting pipes (15, 16).

[0028] The air conditioner (10) includes a refrigerant circuit (20). The refrigerant circuit (20) is a closed circuit in which a refrigerant circulates to implement a refrigeration cycle. The refrigerant circuit (20) is filled with carbon dioxide (CO_2) as a refrigerant.

[0029] The outdoor heat exchanger (23) and the indoor heat exchanger (24) form an air heat exchanger which exchange heat between the refrigerant and air. That is, in the outdoor heat exchanger (23), heat is exchanged between an outdoor air blown by the outdoor fan (12) and the refrigerant. In the indoor heat exchanger (24), heat is exchanged between a room air blown by the indoor fan (14) and the refrigerant.

[0030] The compressor-expander unit (30) is configured by a compression mechanism (50), an expansion mechanism (60), a shaft (40) and a motor (45) which are accommodated in the casing (31). A suction pipe (32) and a discharge pipe (33) are connected to the compression mechanism (50). An inflow pipe (34) and an outflow pipe (35) are connected to the expansion mechanism (60). The compressor-expander unit (30) will be described in detail later.

[0031] Each of the first four-way switching valve (21) and the second four-way switching valve (22) has four ports. In the first four-way switching valve (21), the first port is connected to the discharge pipe (33) of the compressor-expander unit (30); the second port is connected to one end of the indoor heat exchanger (24); the third port is connected to one end of the outdoor heat exchanger (23); and the fourth port is connected to the suction pipe (32) of the compressor-expander unit (30). In the second four-way switching valve (22), the first port is connected to the outflow pipe (35) of the compressor-expander unit (30); the second port is connected to the other end of the outdoor heat exchanger (23); the third port is connected to the other end of the indoor heat exchanger (24); and the fourth port is connected to the inflow pipe (34) of the compressor-expander unit (30).

[0032] Each of the first four-way switching valve (21) and the second four-way switching valve (22) is configured to switch between the state in which the first port and the second port are connected to each other, and the third port and the fourth port are connected to each other (the state shown in solid line in FIG. 1), and the state in which the first port and the third port are connected to each other, and the second port and the fourth port are connected to each other (the state shown in broken line in FIG. 1).

[0033] As shown in FIG. 2, the compressor-expander unit (30) has a casing (31), which is a closed container having a circular cylindrical shape. The compression mechanism (50), the motor (45), and the expansion mechanism (60) are sequentially disposed in the casing (31), from one end to the other end of the longitudinal dimension of the casing (31).

[0034] The motor (45) includes a stator (46) and a rotor (47). The stator (46) is fixed on the inner wall of the casing (31). The rotor (47) is disposed at an inner side of the stator (46), and a shaft (40) goes through the rotor (47).

[0035] The shaft (40) forms a rotational shaft, and includes a main shaft portion (44), a small-diameter eccentric portion (43), and a large-diameter eccentric portion (41). One end of the shaft (40) is provided with the small-

diameter eccentric portion (43), and the other end of the shaft (40) is provided with the large-diameter eccentric portion (41). The small-diameter eccentric portion (43) has a diameter smaller than the diameter of the main shaft portion (44), and is eccentric to the axis of the main shaft portion (44) by a predetermined amount. The large-diameter eccentric portion (41) has a diameter larger than the diameter of the main shaft portion (44), and is eccentric to the axis of the main shaft portion (44) by a predetermined amount.

[0036] The compression mechanism (50) forms a so-called scroll type compressor. The compression mechanism (50) includes a fixed scroll (51) and a movable scroll (54). The fixed scroll (51) is formed of an end plate (52) on which a spiral-wall-like fixed side lap (53) is vertically provided. The end plate (52) of the fixed scroll (51) is fixed on the inner wall of the casing (31). On the other hand, the movable scroll (54) is formed of a plate-like end plate (55) on which a spiral-wall-like movable side lap (56) is vertically provided. The fixed scroll (51) and the movable scroll (54) are disposed face-to-face with each other. The fixed side lap (53) and the movable side lap (56) engage with each other, thereby forming a compression chamber (59). The movable scroll (54) includes a protrusion portion formed at a central portion of the upper surface of the end plate (55), and the small-diameter eccentric portion (43) of the shaft (40) is rotatably fitted into the protrusion portion. Further, the movable scroll (54) is supported by a frame (57) via an Oldham ring (58). The Oldham ring (58) is for controlling the rotation of the movable scroll (54) on its axis. The movable scroll (54) is configured to revolve at a predetermined turning radius, without rotating on its axis.

[0037] The suction pipe (32) is connected to an outer periphery side of the fixed scroll (51). The outflow end of the suction pipe (32) is open to the outermost periphery portion of the compression chamber (59). The discharge pipe (33) is connected to the shaft center of the fixed scroll (51). The inflow end of the discharge pipe (33) is open to a central portion of the compression chamber (59).

[0038] The expansion mechanism (60) is a so-called oscillating piston type fluid device as shown in FIG. 2 and FIG. 3, and forms an expander according to the present invention. The expansion mechanism (60) includes a cylinder (71), a piston (75) accommodated in the cylinder (71), a front head (61), and a rear head (62). In the expansion mechanism (60), the front head (61), the cylinder (71), and the rear head (62) are stacked in this order from one end to the other end of the axis of the shaft (40). The cylinder (71) has a circular cylindrical shape whose both ends are open. One end of the cylinder (71) is closed by the front head (61), and the other end is closed by the rear head (62). The main shaft portion (44) of the shaft (40) passes through the front head (61), the cylinder (71), and the rear head (62).

[0039] The piston (75) has a circular cylindrical shape whose outer diameter is smaller than the inner diameter

of the cylinder (71). The outer periphery surface of the piston (75) is slidable on the inner periphery surface of the cylinder (71); one end surface of the piston (75) is slidable on the front head (61); and the other end surface of the piston (75) is slidable on the rear head (62). A fluid chamber (72) is formed between the inner periphery surface of the cylinder (71) and the outer periphery surface of the piston (75).

[0040] Further, the expansion mechanism (60) is provided with a blade (76) and a pair of bushes (77). The blade (76) is integrally formed with the piston (75). The blade (76) has a plate-like shape which extends radially outward from the outer periphery surface of the piston (75). The fluid chamber (72) in the cylinder (71) is divided by the blade (76) into a high pressure chamber (73) in which the pressure of the refrigerant is high, and a low pressure chamber (74) in which the pressure of the refrigerant is low (see FIG. 3).

[0041] The pair of bushes (77) are disposed so as to fit in a bush groove in the cylinder (71). Each bush (77) has a flat surface and an arc-shaped outer surface, thereby forming a generally half-moon shape, and the flat surfaces face each other. The blade (76) is held slidable between the bushes (77). In other words, in the expansion mechanism (60), the bush (77) is rotatable with respect to the cylinder (71), and the blade (76) is movable back and forth with respect to the bush (77).

[0042] The cylinder (71) is provided with an inflow port (81) and an outflow port (82). The inflow port (81) forms an inflow opening for introducing a high pressure refrigerant into the fluid chamber (72). The outflow port (82) forms an outflow opening for discharging a low pressure refrigerant depressurized in the fluid chamber (72) to the outside of the fluid chamber (72).

[0043] Specifically, the inflow port (81) and the outflow port (82) pass through the cylinder (71) in a radial direction. The inflow port (81) and the outflow port (82) are formed in the cylinder (71) such that they are located close to the bush (77) and that they have the bush (77) interposed therebetween. The inside of each of the inflow port (81) and the outflow port (82) forms a channel having a circular cross section. Here, diameters of the inflow port (81) and the outflow port (82) are uniform from the inflow end to the outflow end.

[0044] The inflow port (81) is formed at a location in the cylinder (71) that is slightly shifted from the bush (77) in a clockwise direction. The outflow end of the inflow pipe (34) is connected to the inflow end of the inflow port (81). The outflow end of the inflow port (81) is open in the high pressure chamber (73) of the fluid chamber (72). The outflow port (82) is formed at a location in the cylinder (71) that is slightly shifted from the bush (77) in a counterclockwise direction. The inflow end of the outflow port (82) is open in the low pressure chamber (74) of the fluid chamber (72). The inflow end of the outflow pipe (35) is connected to the outflow end of the outflow port (82).

<Design Values of Expander>

[0045] Next, design values of an expansion mechanism (60) according to the present embodiment will be described. Here, FIG. 4 shows an example of each design point of the expansion mechanism (60) based on operational conditions of the air conditioner (10). FIG. 5 shows an example of each design dimension of the expansion mechanism (60).

[0046] As shown in FIG. 4, when a normal refrigeration cycle is performed in the refrigerant circuit (20) of the air conditioner (10), the pressure P_i of the refrigerant of the high pressure side (inflow side) of the expansion mechanism (60) is 11.5 [MPa], and the temperature T_i of the refrigerant of the high pressure side is 10°C. Further, the pressure P_o of the refrigerant of the low pressure side (outflow side) of the expansion mechanism (60) is 3.5 [Mpa], and the temperature T_o of the refrigerant of the high pressure side is 0°C. Moreover, the density ρ_i of the refrigerant of the inflow side (i.e., in the inflow port (81)) of the expansion mechanism (60) is 931.7 [kg/m³], and the density ρ_o of the refrigerant of the outflow side (i.e., in the outflow port (82)) of the expansion mechanism (60) is 713.3 [kg/m³].

[0047] FIG. 5 shows example design dimensions for each of three types (A, B, and C) of the expansion mechanism (60) having a different cylinder capacity V_{cc} . Here, in these expansion mechanisms (60), the diameter D_i of the inflow port (81) is set based on the inner diameter D_c of the cylinder (71). Specifically, in the respective expansion mechanisms (60), the relationship between the inner diameter D_c [mm] of the cylinder and the diameter D_i [mm] of the inflow port is set to satisfy $0.065 \times D_c \leq D_i \leq 0.13 \times D_c$. Further, the inflow port (81) of each expansion mechanism (60) has a diameter D_i which satisfies $D_i = 0.085 \times D_c$. On the other hand, in each expansion mechanism (60), the diameter of the outflow port (82) is set based on the diameter D_i of the inflow port (81) and the density ρ_i and the density ρ_o of the above-described refrigerant. Specifically, the outflow port (82) of each expansion mechanism (60) has a diameter D_o [mm] which satisfies $D_o = D_i \times (\rho_i/\rho_o)^2$.

[0048] In the present embodiment, the diameters of the inflow port (81) and the outflow port (82) are set as described above, thereby avoiding the blow-by phenomenon of the refrigerant from the inflow port (81) to the outflow port (82), and reducing the pressure loss of the refrigerant in the inflow port (81) and the outflow port (82). This will be described in detail later.

-Operational Behavior-

[0049] Next, a general operational behavior of the air conditioner (10) will be described with reference to FIG. 1. In the air conditioner (10), the operation is switched between a cooling operation and a heating operation.

[0050] During a cooling operation, the first four-way switching valve (21) and the second four-way switching

valve (22) are switched into the state as shown in broken line in FIG. 1. If the motor (45) of the compressor-expander unit (30) is energized in this state, the refrigerant circulates in the refrigerant circuit (20) to perform a vapor compression refrigerating cycle.

[0051] The high pressure refrigerant compressed in the compression mechanism (50) is discharged from the compressor-expander unit (30) through the discharge pipe (33). The high pressure refrigerant is transferred to the outdoor heat exchanger (23), and heat is dissipated to the outdoor air. The high pressure refrigerant whose heat is dissipated in the outdoor heat exchanger (23) flows into the expansion mechanism (60) of the compressor-expander unit (30) through the inflow pipe (34). The high pressure refrigerant is depressurized in the fluid chamber (72) of the expansion mechanism (60). Here, the internal energy of the refrigerant is converted to the rotational power of the shaft (40). The low pressure refrigerant depressurized in the fluid chamber (72) flows out of the compressor-expander unit (30) through the outflow pipe (35).

[0052] The low pressure refrigerant is transferred to the indoor heat exchanger (24). In the indoor heat exchanger (24), the low pressure refrigerant absorbs heat from a room air and evaporates, thereby cooling the room air. The low pressure refrigerant which flows out from the indoor heat exchanger (24) is sucked into the compression mechanism (50) of the compressor-expander unit (30) through the suction pipe (32), and is compressed again by the compression mechanism (50).

[0053] During a heating operation, the first four-way switching valve (21) and the second four-way switching valve (22) are switched into the state as shown in solid line in FIG. 1. If the motor (45) of the compressor-expander unit (30) is energized in this state, the refrigerant circulates in the refrigerant circuit (20) to perform a vapor compression refrigerating cycle.

[0054] The high pressure refrigerant compressed in the compression mechanism (50) is discharged from the compressor-expander unit (30) through the discharge pipe (33). The high pressure refrigerant is transferred to the indoor heat exchanger (24). In the indoor heat exchanger (24), heat of the high pressure refrigerant is dissipated into the room air, thereby heating the room air. The high pressure refrigerant whose heat is dissipated in the indoor heat exchanger (24) flows into the expansion mechanism (60) of the compressor-expander unit (30) through the inflow pipe (34). The high pressure refrigerant is depressurized in the fluid chamber (72) of the expansion mechanism (60). Here, the internal energy of the refrigerant is converted to the rotational power of the shaft (40). The low pressure refrigerant depressurized in the fluid chamber (72) flows out of the compressor-expander unit (30) through the outflow pipe (35).

[0055] The low pressure refrigerant is transferred to the outdoor heat exchanger (23), where the low pressure refrigerant absorbs heat from the outdoor air and evaporates. The low pressure refrigerant which flows out from

the outdoor heat exchanger (23) is sucked into the compression mechanism (50) of the compressor-expander unit (30) through the suction pipe (32), and is compressed again by the compression mechanism (50).

[0056] Next, a behavior of the expansion mechanism (60) will be described with reference to FIG. 6. In FIG. 6, drawings of the piston (75) whose rotational angle is different by 90° in a clockwise direction are sequentially shown in FIG. 6(A), (B), (C), and (D). In the expansion mechanism (60), a process in which a capacity of the high pressure chamber (73) increases (inflow process) and a process in which a capacity of the low pressure chamber (74) decreases (outflow process) are simultaneously performed as the shaft (40) rotates.

[0057] First, the inflow process in which the high pressure refrigerant flows into the fluid chamber (72) of the expansion mechanism (60) will be described. When the shaft (40) slightly rotates from the position shown in FIG. 6(D), and the contact point between the piston (75) and the cylinder (71) passes by the inflow port (81), the high pressure refrigerant starts to flow into the fluid chamber (72) through the inflow port (81). After that, as the shaft (40) rotates as sequentially shown in FIG. 6(A), (B) and (C), thereby increasing the capacity of the high pressure chamber (73), the refrigerant is sequentially taken into the fluid chamber (72) through the inflow port (81). Here, the high pressure refrigerant is depressurized, and at the same time, the piston (75) is rotated by the internal energy of the refrigerant. The inflow of the high pressure refrigerant into the high pressure chamber (73) continues until the rotational angle of the piston (75) reaches 360°.

[0058] Here, in the present embodiment, the inflow port (81) and the outflow port (82) in the state of FIG. 6(D) are separated by the piston (75). In other words, the diameter D_i of the inflow port (81) according to the present embodiment is set to satisfy $0.065 \times D_c \leq D_i \leq 0.13 \times D_c$. Therefore, communication between the inflow port (81) and the outflow port (82) via the fluid chamber (72) is avoided in the state of FIG. 6(D). With this structure, the refrigerant having flowed into the fluid chamber (72) through the inflow port (81) is prevented from flowing to the outside through the outflow port (82) without giving the internal energy to the piston (75) and the shaft (40).

[0059] Next, the outflow process in which the low pressure refrigerant flows out of the low pressure chamber (74) of the expansion mechanism (60) will be described. When the shaft (40) slightly rotates from the position shown in FIG. 6(D), and the contact point between the piston (75) and the cylinder (71) reaches the outflow port (82), the low pressure chamber (74) and the outflow port (82) communicate with each other. As a result, the low pressure refrigerant in the low pressure chamber (74) starts to flow out of the fluid chamber (72) through the outflow port (82). After that, as the shaft (40) rotates as sequentially shown in FIG. 6(A), (B) and (C), thereby reducing the capacity of the low pressure chamber (74), the refrigerant is sequentially discharged from the fluid chamber (72) to the outside through the outflow port (82).

The outflow of the low pressure refrigerant from the low pressure chamber (74) continues until the rotational angle of the piston (75) reaches 360°.

-Effects of Embodiment-

[0060] In the above embodiment, the diameter D_i of the inflow port (81) satisfies the relationship $0.065 \times D_c \leq D_i \leq 0.13 \times D_c$. With this structure, it is possible to increase the efficiency of the expansion mechanism (60) (power recovery efficiency) to a maximum in the present embodiment. This will be described with reference to FIG. 7 and FIG. 8.

[0061] First, FIG. 7 shows a relationship between an inflow port diameter D_i and the theoretical efficiency of an expander, with respect to three types of expansion mechanisms (A, B, C) having a different cylinder capacity (i.e., cylinder's inner diameter D_c). As shown in the drawing, in the case of the expansion mechanism A whose cylinder's inner diameter D_c is about 22 [mm], the efficiency of the expander is increased to its maximum when the inflow port diameter D_i is set to about 1.9 mm (the point "a" in FIG. 7). On the other hand, in the case where the inflow port diameter D_i of the expansion mechanism A is too large (e.g., about 2.9 [mm] or so), the efficiency of the expander of the expansion mechanism A decreases. This is because if the inflow port diameter D_i is too large, the inflow port (103) and the outflow port (104) communicate with each other as shown, for example, in FIG. 9(D), and a so-called blow-by phenomenon occurs. Also, in the case where the inflow port diameter D_i of the expansion mechanism A is too small (e.g., about 1.2 [mm] or so), the efficiency of the expander of the expansion mechanism A decreases. This is because if the inflow port diameter D_i is too small, the pressure loss of the refrigerant channel in the inflow port (81) increases, and so-called indicated efficiency decreases.

[0062] Similarly, in the case of the expansion mechanism B whose cylinder's inner diameter D_c is about 30 [mm], the efficiency of the expander is increased to its maximum when the inflow port diameter D_i is set to about 2.6 mm (the point "b" in FIG. 7). In the case of the expansion mechanism C whose cylinder's inner diameter D_c is about 38 [mm], the efficiency of the expander is increased to its maximum when the inflow port diameter D_i is set to about 3.2 [mm] (the point "c" in FIG. 7).

[0063] FIG. 8 is a graph showing a relationship between the cylinder's inner diameter D_c and the inflow port diameter D_i obtained by approximation, for achieving high efficiency of each of the above expansion mechanisms A, B and C. That is, high efficiency can be achieved by setting the inflow port diameter D_i to satisfy $0.085 \times D_c$ in relation to the cylinder's inner diameter D_c , as the relations in FIG. 8 indicate. Further, the hatched area in FIG. 8 indicates a range in which the theoretical efficiency of the expander shown in FIG. 7 is about 90 % or more (the line L or above in FIG. 7). In other words, relatively high efficiency of the expander can be achieved by setting

the inflow port diameter D_i to satisfy the relationship $0.065 \times D_c \leq D_i \leq 0.13 \times D_c$ in relation to the cylinder's inner diameter D_c .

[0064] In other words, according to the present embodiment, the diameter D_i [mm] of the inflow port (81) satisfies the relationship $0.065 \times D_c \leq D_i \leq 0.13 \times D_c$. Therefore, the inflow port diameter D_i is not too large, and thus, a so-called blow-by phenomenon can be prevented; and the inflow port diameter D_i is not too small, and thus, an increase in pressure loss in the inflow port (81) can be prevented. As a result, desired efficiency can be obtained in the expansion mechanism (60) according to the present embodiment, thereby making it possible to enhance the energy saving characteristics of the air conditioner (10) to which the expander is applied.

[0065] On the other hand, according to the present embodiment, the diameter D_o [mm] of the outflow port (82) is set to satisfy $D_o = D_i \times (\rho_i/\rho_o)^2$. Here, if the inflow port diameter D_i and the outflow port diameter D_o are set to have the same diameter, the outflow port diameter D_o may be too small and the pressure loss in the outflow port (82) may be slightly increased, because a fluid whose density is lower than the density of a fluid flowing in the inflow port (81) flows in the outflow port (82). In view of this, according to the present embodiment, a ratio of the density ρ_i of the refrigerant of the inflow side (high pressure side) to the density ρ_o of the refrigerant of the outflow side (low pressure side) (i.e., density ratio ρ_i/ρ_o) is considered, and the outflow port diameter D_o is decided by multiplying the inflow port diameter D_i by the square of the density ratio (ρ_i/ρ_o). With this structure, it is possible to avoid an increase in pressure loss in the outflow port (82), and possible to further enhance the efficiency of the expansion mechanism (60).

<<Other Embodiments>>

[0066] The following structures may be used in the above embodiment.

[0067] In the above embodiment, the diameter D_i [mm] of the inflow port (81) is set to satisfy the relationship $0.065 \times D_c \leq D_i \leq 0.13 \times D_c$. However, instead of this relationship, the diameter D_o of the outflow port (82) may be set to satisfy the relationship $0.065 \times D_c \leq D_o \leq 0.13 \times D_c$. In this case as well, it is possible to prevent the occurrence of a blow-by phenomenon because of the too large D_o , and possible to avoid an increase in pressure loss because of too small D_o . As a result, in this case as well, it is possible to achieve high efficiency of the expansion mechanism (60). Further, in this case, the diameter D_i of the inflow port (81) is set to satisfy the relationship $D_i = D_o \times (\rho_o/\rho_i)^2$. Here, if the inflow port diameter D_i and the outflow port diameter D_o are set to have the same diameter, the diameter of the inflow port (81) may be larger than desired, because a fluid whose density is higher than the density of a fluid flowing in the inflow port (81) flows in the outflow port (82). In view of this, a ratio of the density ρ_i of the refrigerant of the inflow side (high

pressure side) to the density ρ_o of the refrigerant of the outflow side (low pressure side) (i.e., density ratio ρ_i/ρ_o) may be considered, and the inflow port diameter D_i may be decided by multiplying the outflow port diameter D_o by the square of the density ratio (ρ_o/ρ_i).

[0068] The foregoing embodiments are merely preferred examples in nature, and are not intended to limit the scope, applications, and use of the invention.

INDUSTRIAL APPLICABILITY

[0069] As described above, the present invention is useful for an expander which recovers power of a fluid depressurized in a cylinder.

Claims

1. An expander having a cylinder at which an inflow port and an outflow port are formed, a piston accommodated in the cylinder so as to be eccentric to a rotational shaft, and a blade which divides a fluid chamber formed between the piston and the cylinder into a high pressure side and a low pressure side of a fluid, the expander recovering power of a fluid depressurized in the fluid chamber, wherein an inner diameter of the cylinder is D_c [mm], and a diameter D_i [mm] of the inflow port satisfies a relationship

$$0.065 \times D_c \leq D_i \leq 0.13 \times D_c$$

2. The expander of claim 1, wherein a density of a fluid in the inflow port is ρ_i [kg/m³], and a density of a fluid in the outflow port is ρ_o [kg/m³], and a diameter D_o [mm] of the outflow port satisfies a relationship

$$D_o = D_i \times (\rho_i/\rho_o)^2$$

3. An expander having a cylinder at which an inflow port and an outflow port are formed, a piston accommodated in the cylinder so as to be eccentric to a rotational shaft, and a blade which divides a fluid chamber formed between the piston and the cylinder into a high pressure side and a low pressure side of a fluid, the expander recovering power of a fluid depressurized in the fluid chamber, wherein an inner diameter of the cylinder is D_c [mm], and a diameter D_o [mm] of the outflow port satisfies a relationship

$$0.065 \times D_c \leq D_o \leq 0.13 \times D_c$$

4. The expander of claim 3, wherein a density of a fluid in the inflow port is ρ_i [kg/m³], and a density of a fluid in the outflow port is ρ_o [kg/m³], and a diameter D_i [mm] of the inflow port satisfies a relationship

$$D_i = D_o \times (\rho_o/\rho_i)^2$$

FIG.1

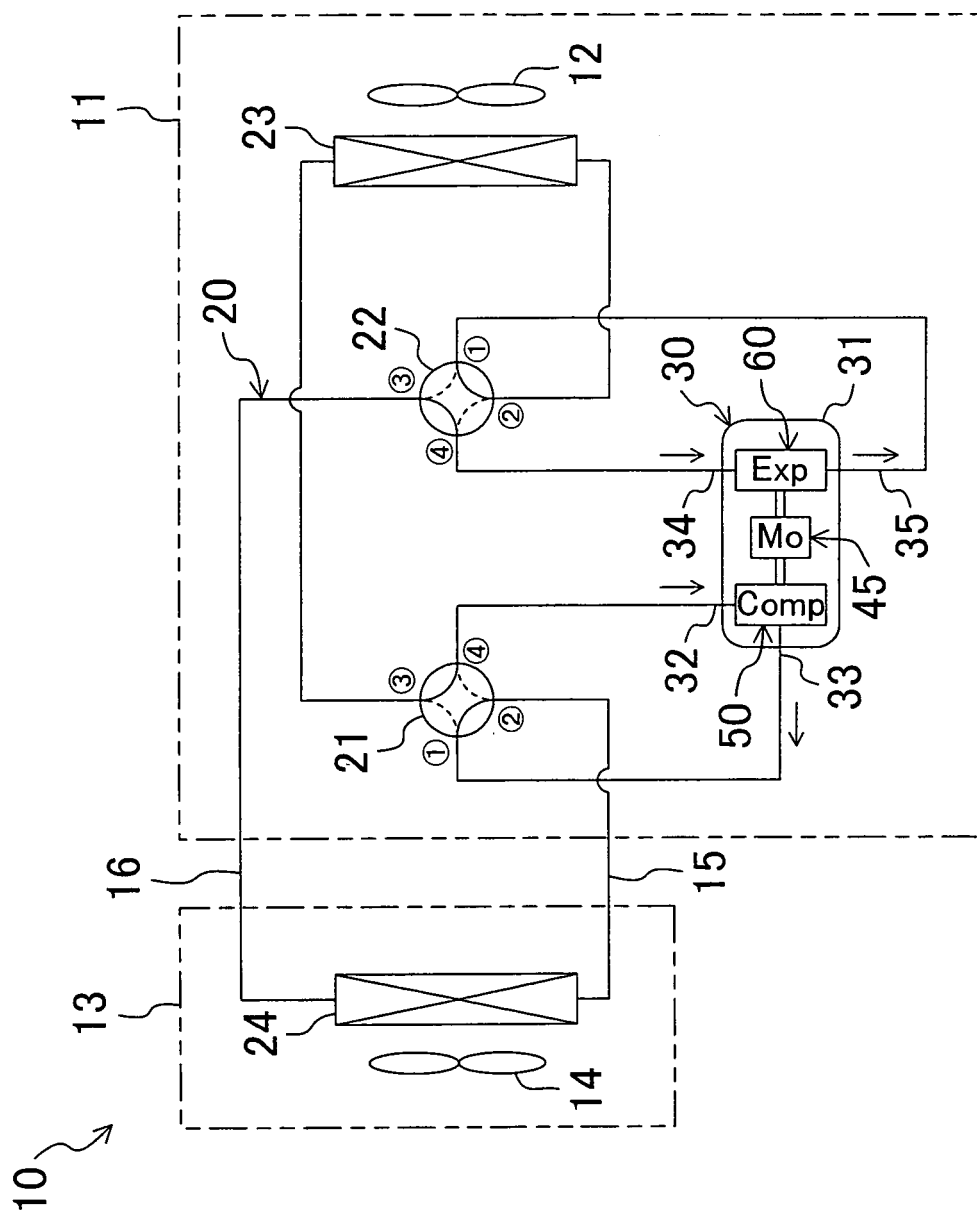


FIG.2

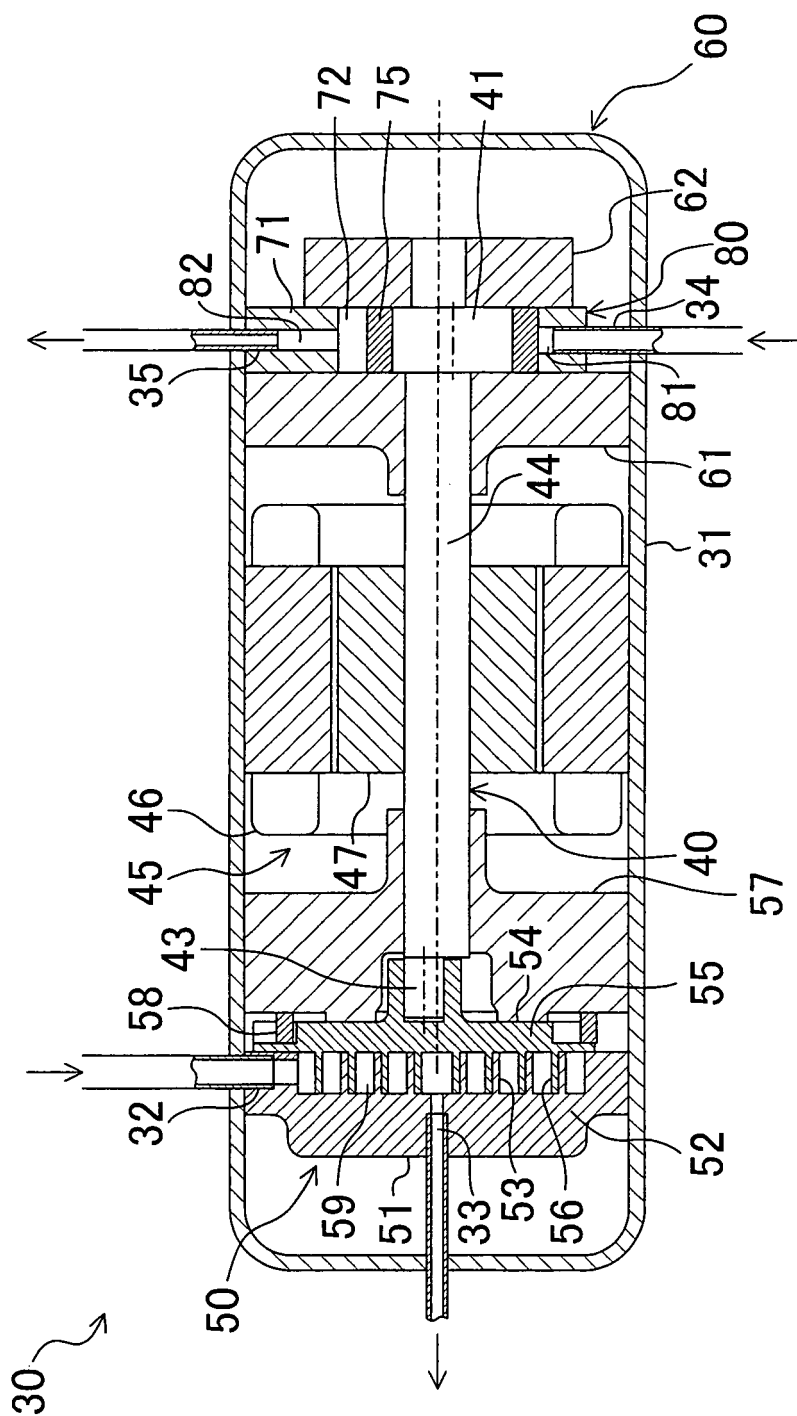


FIG.3

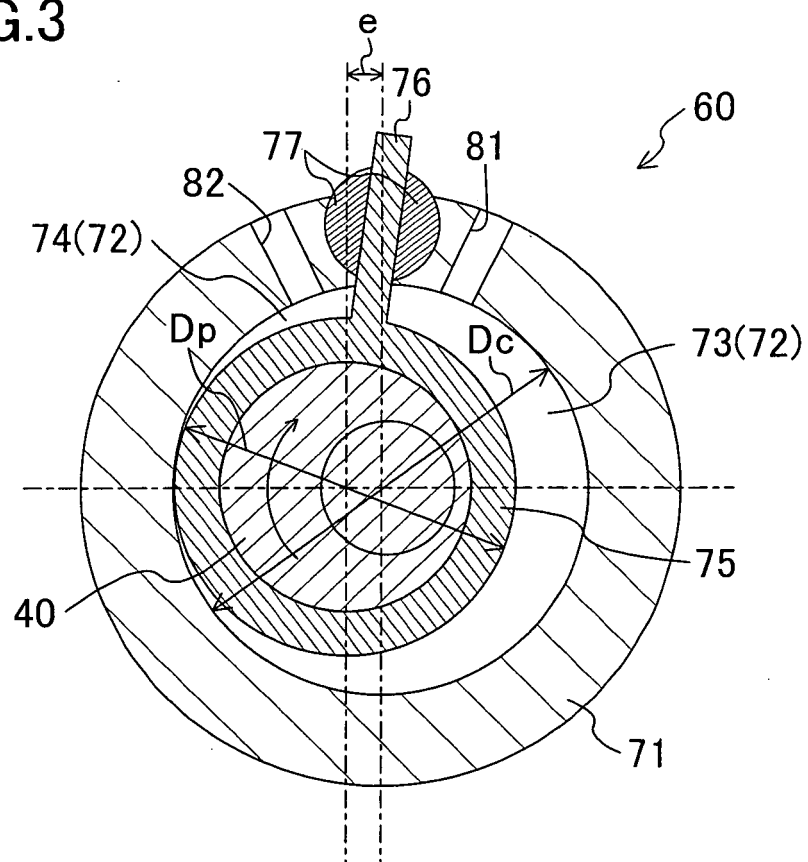


FIG.4

Inflow Refrigerant Pressure	P_i	[MPa]	11.5
Outflow Refrigerant Pressure	P_o	[MPa]	3.5
Inflow Refrigerant Temperature	T_i	[°C]	10.0
Outflow Refrigerant Temperature	T_o	[°C]	0.0
Inflow Refrigerant Density	ρ_i	[kg/m ³]	931.7
Outflow Refrigerant Density	ρ_o	[kg/m ³]	713.3

FIG.5

			A	B	C
Cylinder' s Capacity	V_{cc}	[cc]	0.50	0.75	1.00
Cylinder' s Inner Diameter	D_c	[mm]	22.00	30.00	38.00
Cylinder' s Height	D_h	[mm]	9.00	9.50	10.00
Piston' s Outer Diameter	D_p	[mm]	20.33	28.27	36.28
Piston' s Eccentricity	e	[mm]	0.84	0.87	0.86
Inflow Port Diameter	D_i	[mm]	1.9	2.6	3.2
Outflow Port Diameter	D_o	[mm]	3.2	4.4	5.5

FIG. 6

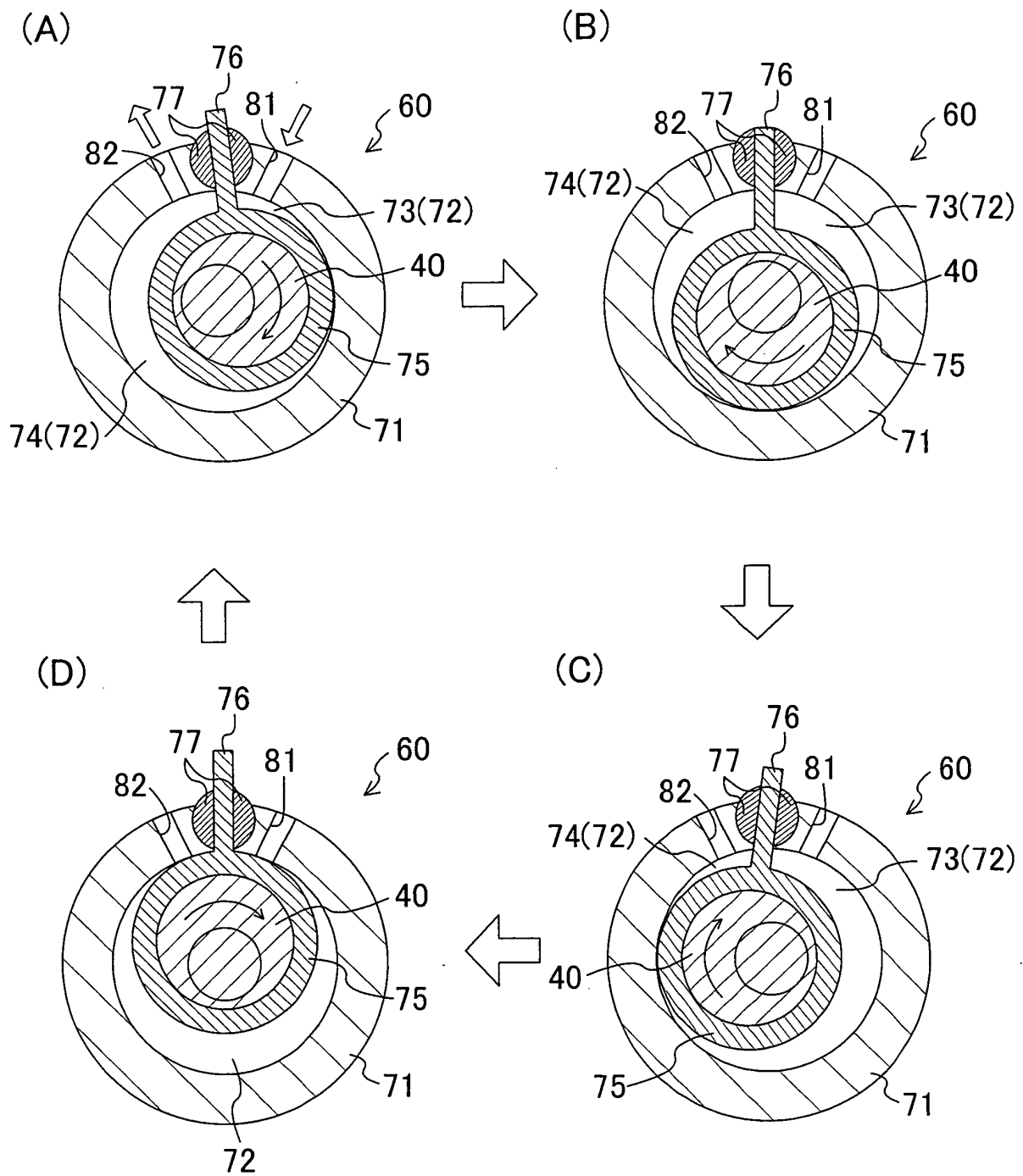


FIG.7

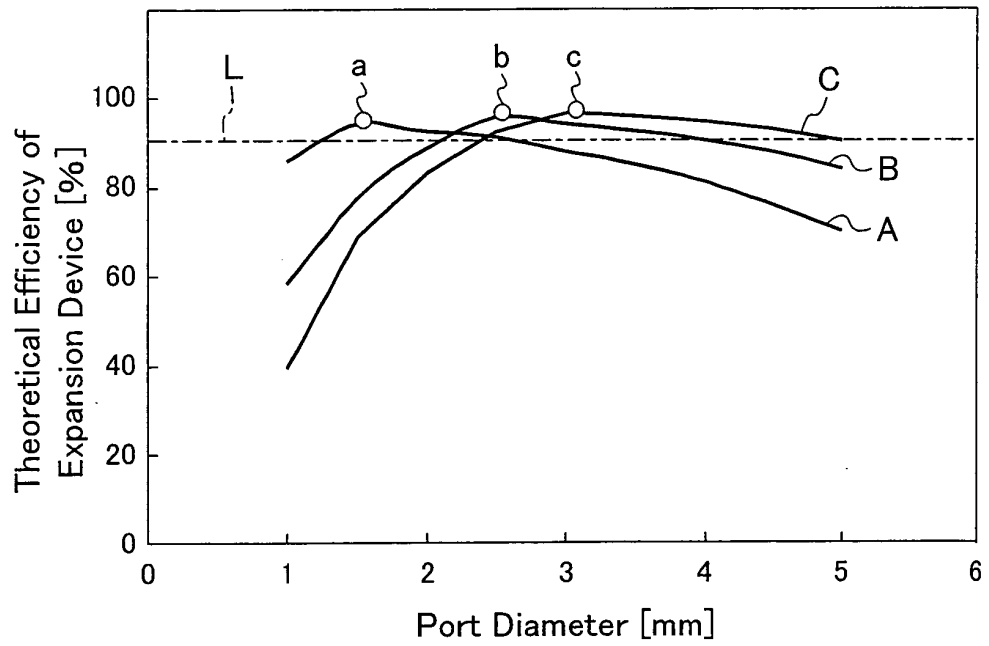


FIG.8

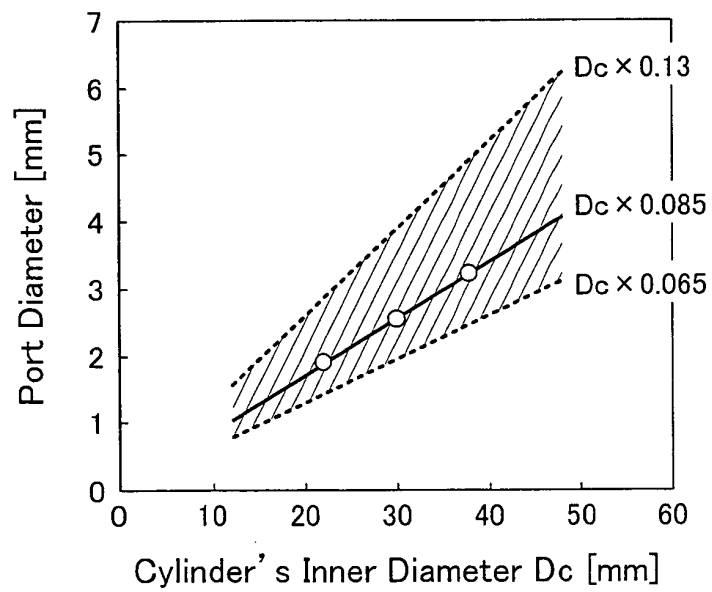
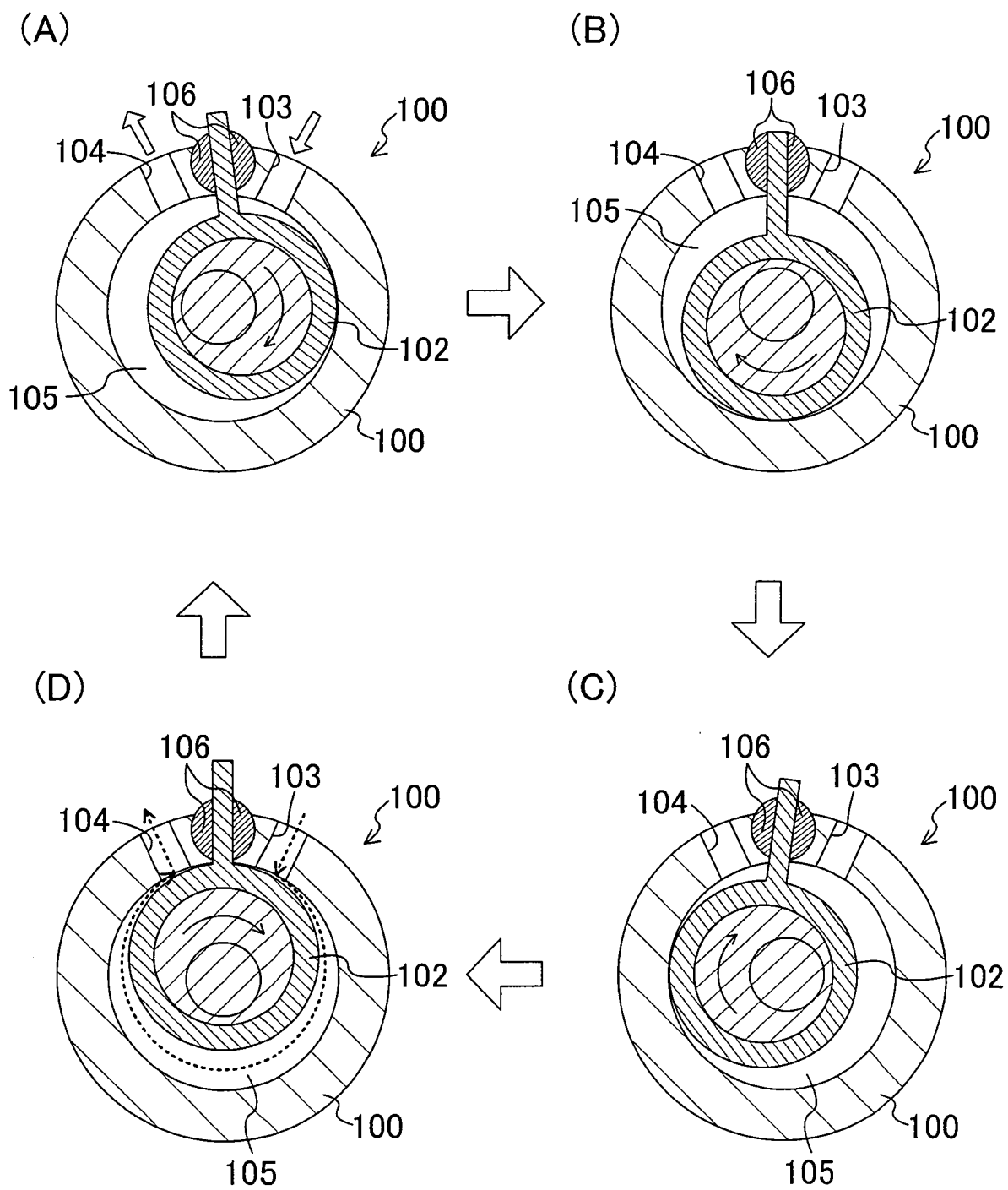


FIG.9



INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2009/000859

A. CLASSIFICATION OF SUBJECT MATTER

F01C1/356 (2006.01) i, F01C21/18 (2006.01) i

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

F01C1/356, F01C21/18

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Jitsuyo Shinan Koho 1922-1996 Jitsuyo Shinan Toroku Koho 1996-2009

Kokai Jitsuyo Shinan Koho 1971-2009 Toroku Jitsuyo Shinan Koho 1994-2009

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	WO 2008/026428 A1 (Matsushita Electric Industrial Co., Ltd.), 06 March, 2008 (06.03.08), Claim 1; Par. No. [0034]; all drawings & JP 4143685 B2	1-4
P, A	WO 2008/050654 A1 (Matsushita Electric Industrial Co., Ltd.), 02 May, 2008 (02.05.08), Par. No. [0081]; Fig. 4A (Family: none)	2, 4

☐ Further documents are listed in the continuation of Box C.☐ See patent family annex.

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"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

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"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art

"&" document member of the same patent family

Date of the actual completion of the international search

15 May, 2009 (15.05.09)

Date of mailing of the international search report

26 May, 2009 (26.05.09)

Name and mailing address of the ISA/
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REFERENCES CITED IN THE DESCRIPTION

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Patent documents cited in the description

- JP H08338356 B [0005]