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(54) RECIPROCATING COMPRESSOR

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(57) ABSTRACT

A reciprocating compressor provided with a piston, which is compact and having high sealing performance is provided. A reciprocating compressor includes: a low-pressure compressor unit having a low-pressure piston and a low-pressure cylinder for compressing air with the low-pressure piston reciprocating while oscillating within the low-pressure cylinder; a high-pressure compressor unit having a high-pressure piston and a high-pressure cylinder for further compressing the air compressed in the low-pressure compressor unit with the high-pressure piston reciprocating while oscillating within the high-pressure cylinder; and a motor for driving the low-pressure compressor unit and the high-pressure compressor unit. A maximum tilt angle during oscillation of the low-pressure piston is made larger than a maximum tilt angle during oscillation of the high-pressure piston.

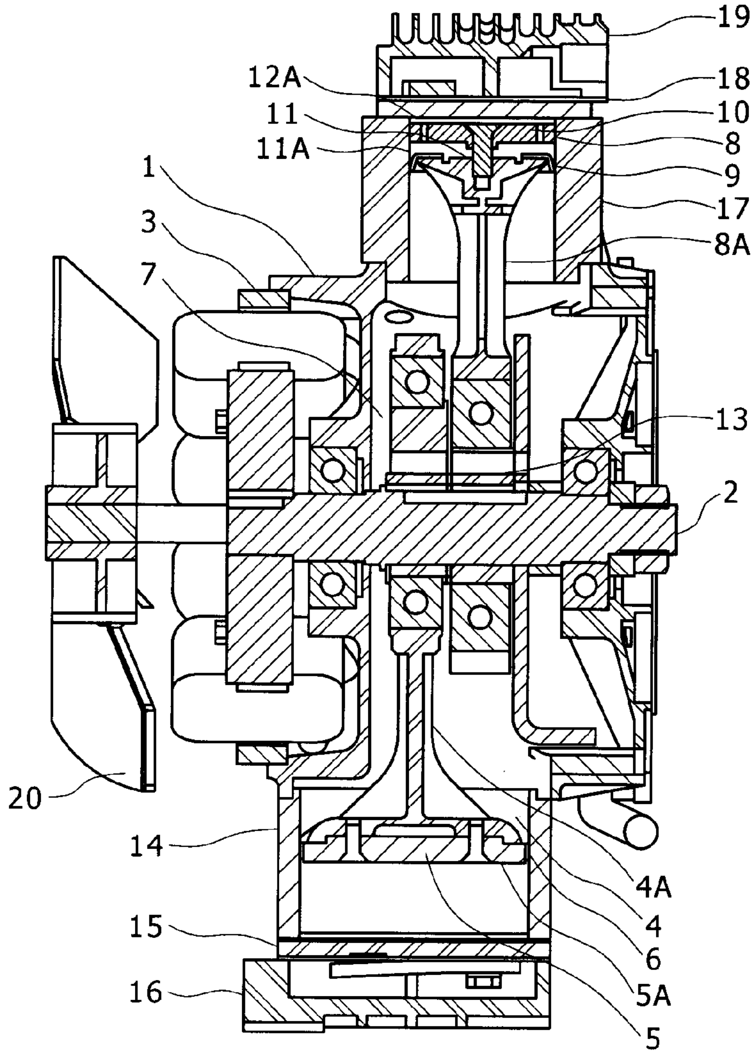


FIG. 1

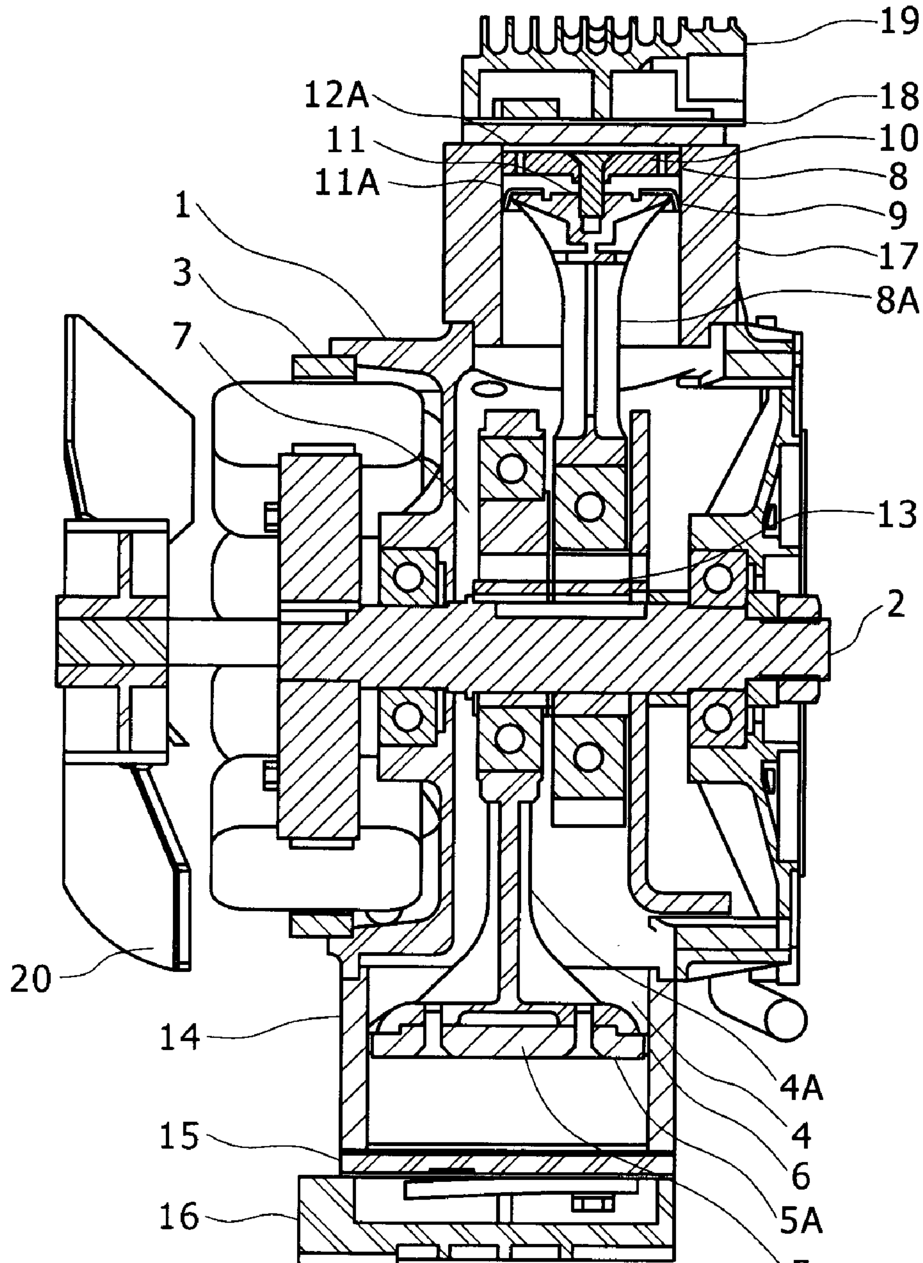


FIG. 2

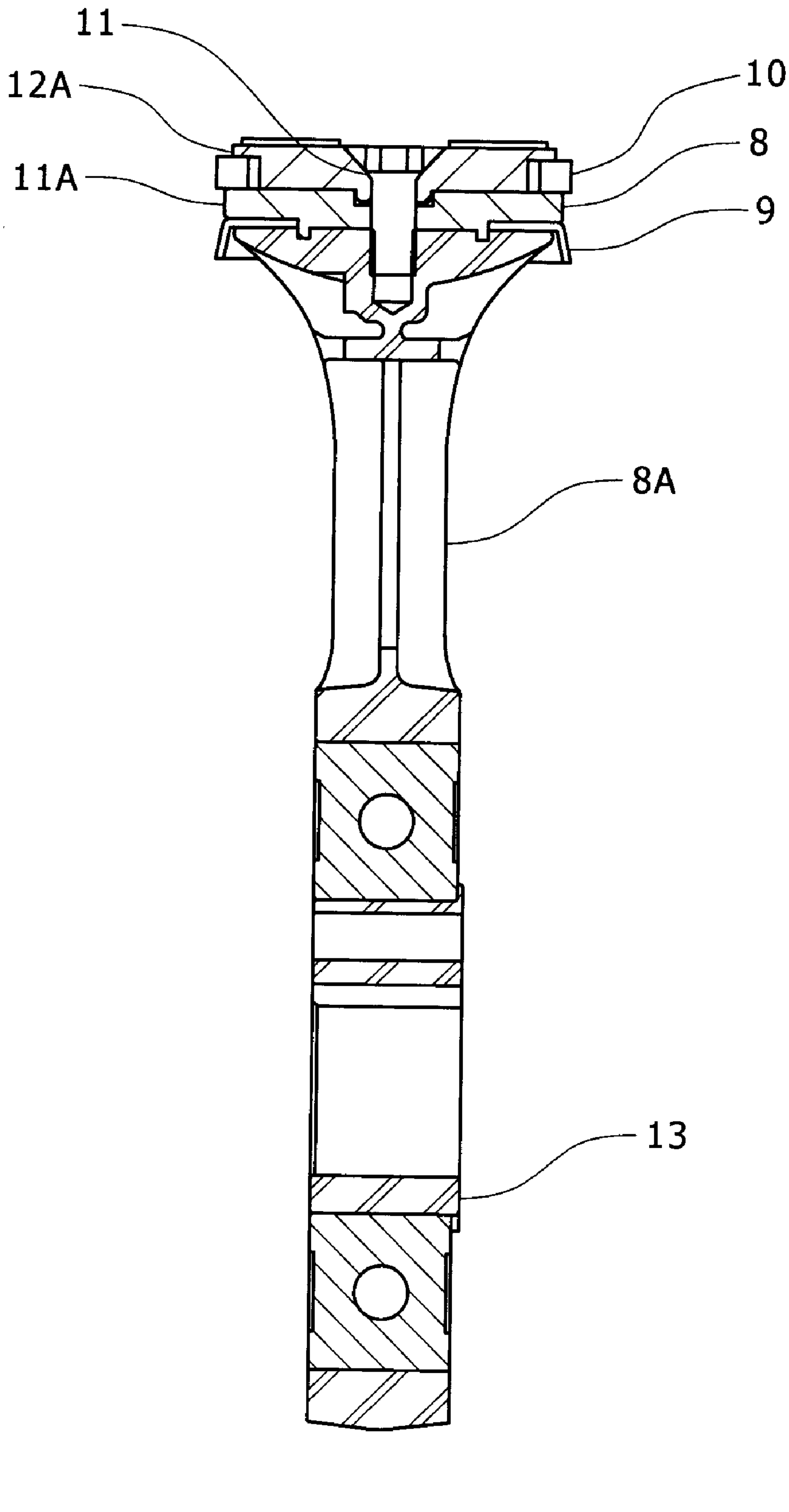


FIG. 3

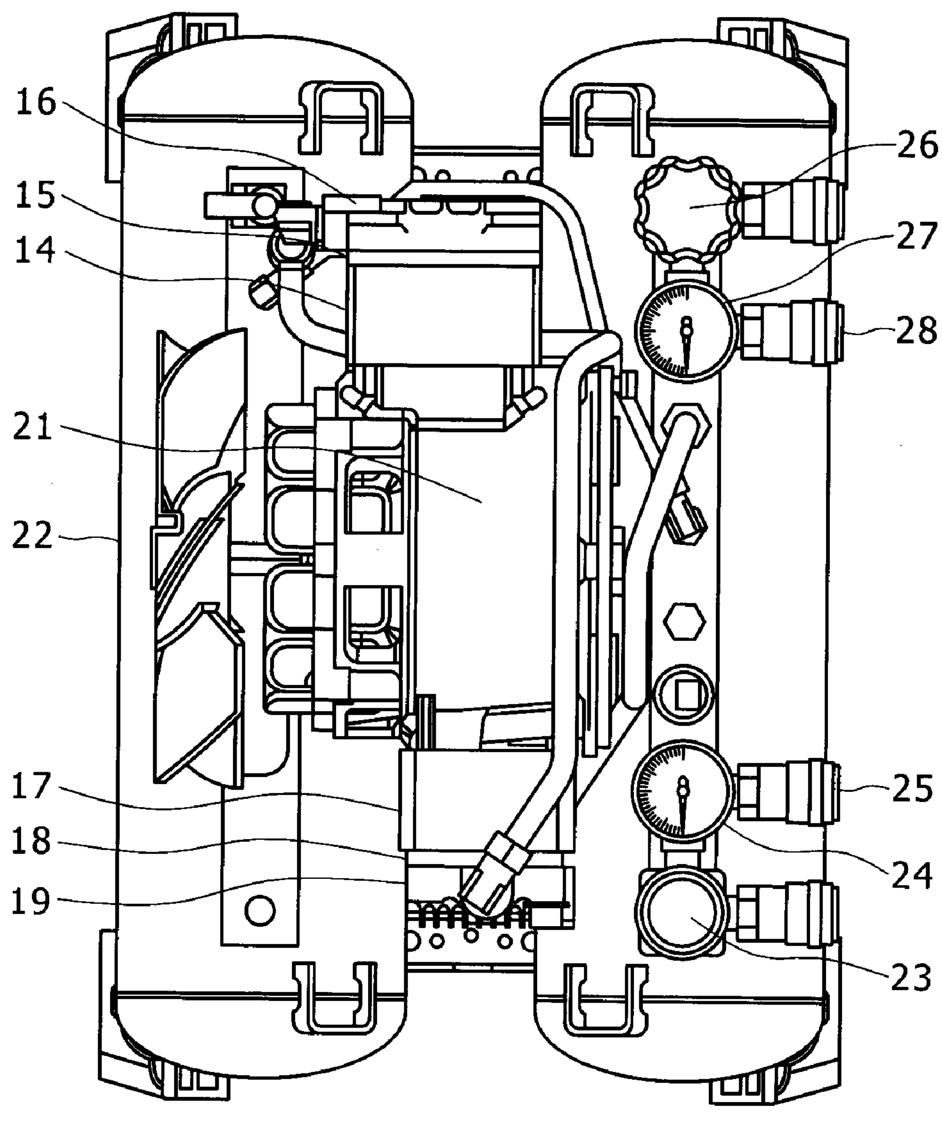
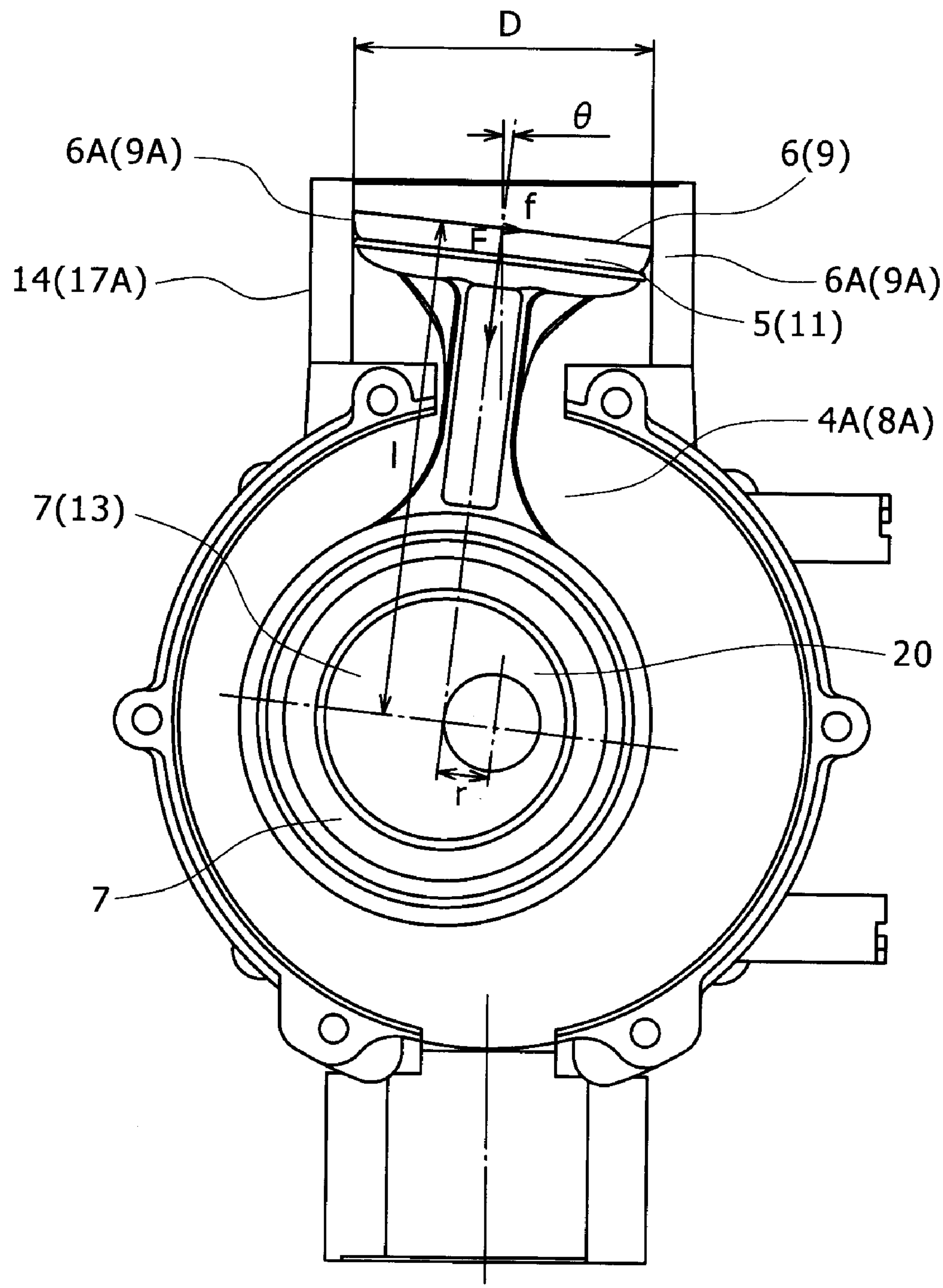


FIG. 4



RECIPROCATING COMPRESSOR

[0001] This application claims the priority of Japanese Patent Application No. JP 2010-092732, filed Apr. 14, 2010, the disclosure of which is expressly incorporated by reference herein in its entirety.

BACKGROUND OF THE INVENTION

[0002] 1. Field of the Invention

[0003] The present invention relates to a reciprocating compressor in which pistons reciprocate within cylinders.

[0004] 2. Description of the Related Art

[0005] According to an oscillating compressor disclosed in Japanese Patent Application Laid-Open Publication No. 2007-32532, in a two-stage reciprocating compressor in which air compressed in a low-pressure compressor unit is compressed in a high-pressure compressor unit, a piston reciprocates while oscillating within a cylinder, thereby compressing air in each of the low-pressure compressor unit and the high-pressure compressor unit.

[0006] According to a two-stage compressor disclosed in Japanese Patent Application Laid-Open Publication No. H11-62822, in a two-stage reciprocating compressor in which air compressed in a low-pressure compressor unit is compressed in a high-pressure compressor unit, a piston with a piston body fixed to a connecting rod is used in the low-pressure compressor unit, so that the piston reciprocates while oscillating (tilting) within a cylinder. On the other hand, a piston with a piston body oscillating with respect to a leading end of a connecting rod is used in the high-pressure compressor unit, so that a leading end of the piston reciprocates without tilting within a cylinder.

[0007] In the oscillating compressor disclosed in Japanese Patent Application Laid-Open Publication No. 2007-32532, the piston is formed without consideration of a tilt angle of the piston within the low-pressure or high-pressure cylinder. It is therefore not possible to improve the sealing performance and lifetime of the piston.

[0008] In the two-stage compressor disclosed in Japanese Patent Application Laid-Open Publication No. H11-62822, the piston with the piston body oscillating with respect to the leading end of the connecting rod is used in the high-pressure compressor unit. This causes an increase in the axial length of the piston, and therefore it is difficult to downsize the piston.

SUMMARY OF THE INVENTION

[0009] Accordingly, in view of the above-described problems, an object of the present invention is to provide a reciprocating compressor with pistons of low- and high-pressure compressor units being compact and having high sealing performance by forming the pistons in consideration of tilt angles of the pistons within cylinders.

[0010] In order to address the problems discussed above, according to one aspect of the present invention, there is provided a reciprocating compressor including: a low-pressure compressor unit having a low-pressure piston and a low-pressure cylinder for compressing air with the low-pressure piston reciprocating while oscillating within the low-pressure cylinder; a high-pressure compressor unit having a high-pressure piston and a high-pressure cylinder for further compressing the air compressed in the low-pressure compressor unit with the high-pressure piston reciprocating while oscillating within the high-pressure cylinder; and a motor for driving the low-pressure compressor unit and the high-pressure compressor unit. A maximum tilt angle during oscillation of the low-pressure piston is made larger than a maximum tilt angle during oscillation of the high-pressure piston.

[0011] According to another aspect of the present invention, there is provided a reciprocating compressor including: a motor having a rotating shaft; a low-pressure compressor unit having a low-pressure cylinder and a low-pressure piston for compressing air; and a high-pressure compressor unit having a high-pressure cylinder and a high-pressure piston for further compressing the air compressed in the low-pressure compressor unit. The low-pressure piston and the high-pressure piston each include: an eccentric portion for performing eccentric motion with rotation of the rotating shaft of the motor; a connecting rod extending from the eccentric portion; and a piston body provided on a leading end of the connecting rod. When an eccentric amount of the eccentric portion of the low-pressure piston with respect to the rotating shaft of motor is represented by **r1** and an eccentric amount of the eccentric portion of the high-pressure piston with respect to the rotating shaft of motor is represented by **r2**, the low- and high-pressure pistons are formed in such a manner that $r1>r2$ is established.

[0012] According to the present invention, it is possible to provide a two-stage reciprocating compressor provided with a piston, which is compact and having high sealing performance.

BRIEF DESCRIPTION OF THE DRAWINGS

[0013] The present invention will become fully understood from the detailed description given hereinafter and the accompanying drawings, wherein:

[0014] FIG. **1** is a diagram showing a low-pressure compressor unit and a high-pressure compressor unit of a compressor according to an embodiment of the present invention;

[0015] FIG. **2** is an enlarged view of the high-pressure compressor unit according to the embodiment of the present invention;

[0016] FIG. **3** is a general view of the reciprocating compressor according to the embodiment of the present invention; and

[0017] FIG. **4** is a diagram showing oscillating motion of a piston according to the embodiment of the present invention.

DESCRIPTION OF PREFERRED EMBODIMENTS

[0018] An embodiment of the present invention will be described with reference to FIGS. **1** to **4**.

[0019] A compressor according to the embodiment of the present invention will be described with reference to FIG. **1**. The compressor according to this embodiment has a crankcase **1**. A motor **3** having a shaft (a rotating shaft) **2** is attached to the crankcase **1**. A low-pressure piston **4** including a connecting rod **4**A and a piston body **5**, and a high-pressure piston **8** including a connecting rod **8**A and a piston body **11** are attached to the shaft **2** of the motor **3** through eccentric portions **7** and **13** for converting a rotational motion to a reciprocating motion. The low-pressure piston body **5** is composed of a retainer **5**A, and the high-pressure piston body **11** is composed of a base **11**A and a top **12**A.

[0020] The low-pressure piston body **5** is provided with a lip ring **6**. The high-pressure piston body **11** is provided with a lip ring **9** and a piston ring **10**.

[0021] The lip ring **6** is provided with a skirt portion (a lip portion) for increasing a contact area with a cylinder **14**. The skirt portion is mounted facing a low-pressure compression chamber, thereby allowing the ensuring of the sealing performance between the piston **4** and the cylinder **14**.

[0022] Next, a description will be given with reference to the enlarged view of the high-pressure compressor unit shown in FIG. **2**. The lip ring **6**, the lip ring **9**, and the piston ring **10** are made of resin material having excellent resistance to abrasion and self-lubricating property and each formed in an approximately annular shape. The piston ring **10** is substantially rectangular in cross section, and has a uniform radial width over the almost entire circumference thereof. Also, an abutment joint (not shown) is circumferentially formed on the piston ring **10**. This abutment joint allows expansion and contraction in diameter while maintaining the sealing performance. Additionally, when the high-pressure piston **8** is at the top dead center position or the bottom dead center position, the inner diameter of the piston ring **10** in a state brought into contact with an inner peripheral surface of a high-pressure cylinder **17** to be described later is larger than the smallest diameter of a piston ring groove into which the piston ring **10** is mounted. Thus, the piston ring **10** is radially movable with respect to the high-pressure piston **8**.

[0023] The lip ring **9** is mounted in the direction opposite to the lip ring **6** (the skirt portion of the lip ring **9** faces the crankcase), so that the centers of the lip ring **9** and the piston body **11** are aligned. Also, when the cylinder **17** is assembled in the crankcase **1**, the lip ring **9** is brought into contact with an inner wall surface of the cylinder **17** to define an assembly position of the cylinder **17**, so that the centers of the cylinder **17** and the piston body **11** are aligned. Thus, centering of the piston ring **10** mounted on the piston body **11**, and the cylinder **17** can be performed. Further, the lip ring **9** can prevent contact between the piston body **11** and the cylinder **17** when the piston ring **10** becomes worn, and therefore the lifetime of the piston body **11** and the cylinder **17** can be improved. In addition, the lip ring **9** is interposed between the base **11**A and the connecting rod **8**A. Thus, compression heat generated in the cylinder **17** can be prevented from being transmitted from the piston body **11** to the connecting rod **8**A, so that the temperature of a large end can be lowered. It is therefore possible to improve the lives of bearings provided on outer peripheries of the eccentric portions **7** and **13**.

[0024] The lip ring **6** is mounted on the low-pressure piston **4**, thereby allowing reduction in manufacturing cost. Also, the lip ring **9** and the piston ring **10** are mounted on the high-pressure piston **8**, thereby allowing improvement in assembly performance, sealing performance, and resistance to abrasion.

[0025] The crankcase **1** is mounted with the low-pressure cylinder **14**, an air valve **15**, a cylinder head **16**, the high-pressure cylinder **17**, an air valve **18**, and a cylinder head **19**. With the low-pressure piston **4** including the low-pressure connecting rod **4**A, and the high-pressure piston **8** including the high-pressure connecting rod **8**A, compression chambers are formed within the respective cylinders.

[0026] In the compressor according to the embodiment of the present invention, when the motor **3** is driven to rotate the shaft **2**, the eccentric portions **7** and **13** allow the low- and high-pressure pistons **4** and **8** to reciprocate within the cylinders **14** and **17**, respectively, thereby driving the low-pressure compressor unit including the low-pressure piston **4** and cylinder **14**, and the high-pressure compressor unit including the high-pressure piston **8** and cylinder **17**. Also, the compressor according to the embodiment of the present invention has a two-stage compression structure in which the low-pressure compressor unit sucks air into the cylinder **14** through the cylinder head **16** and the air valve **15** from the atmospheric pressure and compresses the sucked air to discharge the compressed air to the high-pressure compressor unit through a pipe (not shown), and then the high-pressure compressor unit sucks the compressed air discharged from the low-pressure compressor unit and further compresses the air to discharge the air to a storage tank. At this time, the low- and high-pressure compressor units and the pipe is air-cooled by the rotation of a fan **20** mounted coaxially with the shaft **2**.

[0027] In general, in the case of air compressors which perform single-stage compression, air is sucked from the atmospheric pressure and compressed to the maximum pressure by a piston to be discharged. However, due to compression heat generated at the time of compression, discharge efficiency decreases, or, noise, vibration and the like due to torque fluctuations of a motor are generated. Therefore, the motor needs to have the capability for providing a high power output, resulting in an increase in size. Also, under high pressure, a pressure ratio (discharge absolute pressure/suction absolute pressure) increases, thereby raising the discharge temperature and deteriorating the suction efficiency. Therefore, it becomes difficult to ensure the discharge air quantity. Furthermore, the rise of the discharge temperature might cause an increase in distortion and leakage of an air valve. Also, in compressors which perform intermittent running, moisture in the suction air is condensed, thereby increasing the occurrence of drain, which might cause a failure.

[0028] Therefore, for example, in the compressor disclosed in Japanese Patent Application Laid-Open Publication No. H11-62822, the above-described problems are avoided by adopting the following two-stage compression structure. For example, in order to raise the pressure to about 1 MPa or more, on the low-pressure side, air is sucked from the atmospheric pressure to be subjected to pressure rising and temporarily intermediate cooling, and then on the high-pressure side, the air subjected to pressure rising and intermediate cooling by the low-pressure side is sucked and further subjected to pressure rising to be discharged. In particular, portable small compressors adopt the two-stage compression structure in view of low vibration, low noise, miniaturization (downsizing of a motor, weight reduction of a compressor unit).

[0029] In the two-stage compression, the respective pressure ratios on the low- and high-pressure sides are reduced, relative to the case of the single-stage compression, thereby allowing improvement in efficiency, reduction in heat generation, and reduction in performance deterioration caused by the heat generation described above. Furthermore, torque fluctuations of the motor can be reduced and therefore low vibration and low noise can be realized.

[0030] Next, the design of a typical two-stage compressor will be described.

[0031] The two-stage compressor disclosed in Japanese Patent Application Laid-Open Publication No. H11-62822 adopts a reciprocating piston structure in which a connecting rod is connected to a piston through a needle bearing on the high-pressure side. In the embodiment of the present invention, on the other hand, such needle bearing is removed, and a rocking piston mechanism with the connecting rod and the piston body integrally formed is employed, thereby realizing durability improvement by reducing movable portions, weight reduction and low noise, and reduction in cost by reducing the number of components.

[0032] Also, the oscillating compressor disclosed in Japanese Patent Application Laid-Open Publication No. 2007-32532 includes the pistons formed without relatively considering the angles at which the pistons are tilted within the low- and high-pressure cylinders. Consequently, in the low-pressure compressor unit, the piston tilt angle is minimized, in spite of the fact that, even when the piston tilt angle is made larger than that of the high-pressure side, its influence on the sealing performance and abrasion property is small. For this reason, the downsizing and weight reduction of the piston cannot be sufficiently achieved. Also, in the high-pressure compressor unit, the piston tilt angle is not designed to be made sufficiently small, in spite of the fact that, when the piston tilt angle is made too large, its influence on the sealing performance and abrasion property increases. For this reason, the sealing performance and lifetime of the piston cannot be achieved.

[0033] In view of the foregoing, in the embodiment of the present invention, the low-pressure piston **4** and the high-pressure piston **8** include the rocking piston mechanism in which: the connecting rods **4**A and **8**A and the piston bodies **5** and **11** are integrally formed; the piston bodies **5** and **11** are tilted as the connecting rods **4**A and **8**A are tilted; and the piston bodies **5** and **11** reciprocate while oscillating within the cylinders **14** and **17**. Also, the compressor according to this embodiment is designed in consideration of the angles at which the low-pressure piston **4** and the high-pressure piston **8** are tilted within the cylinders **14** and **17**, respectively.

[0034] Hereinafter, the design of the two-stage compressor according to the embodiment of the present invention will be described in detail.

[0035] The motor powers, that is, the shaft powers (the workloads w) of the low- and high-pressure sides increase with an increase in the respective pressure ratios of suction pressure to discharge pressure. The shaft power of the two-stage compressor depends on the sum of the shaft powers of the low- and high-pressure sides. Also, the shaft power decreases as the pressure ratios on the low- and high-pressure sides decrease. That is to say, when the pressure ratios on the low- and high-pressure sides become equal, the shaft power becomes minimum. In view of the foregoing, in the embodiment of the present invention, various sizes (such as bore diameter and stroke) of the low- and high-pressure pistons **4** and **8** are designed based on the shaft power in consideration of the maximum pressure of the compressor.

[0036] Here, the motor shaft powers of the low- and high-pressure sides are calculated. Required shaft power Ls and theoretical adiabatic aerodynamic force Lad are expressed by the following equations (1) and (2):

$$Ls = \frac{Lad}{\eta ad} \tag{1}$$

where Ls represents the required shaft power; and η_{ad} represents the overall adiabatic efficiency.

$$Lad - \frac{k}{k-1} \times \frac{P_s Q_s}{0.060} \times \left\{ \left(\frac{Pd}{Ps} \right)^{\frac{k-1}{k}} - 1 \right\} \tag{2}$$

where Lad represents the theoretical adiabatic aerodynamic force; Qs represents the actual air volume in suction condition; Ps represents the suction absolute pressure; Pd represents the discharge absolute pressure; and κ represents the specific heat ratio.

[0037] Here, the actual air volume Qs shown in the equation (2) is determined by the parameters of the members composing the compressor and the efficiency of the compressor, as expressed by the following equation (3):

$$Qs = \frac{\pi}{4} \times D^2 \times S \times N \times \eta v \tag{3}$$

where D represents the bore diameter; S represents the stroke; N represents the rotational speed; and ηv represents the volumetric efficiency.

[0038] As described above, the motor power (the total shaft power) decreases as the pressure ratios on the low- and high-pressure sides in the two-stage compression and the difference in shaft power therebetween decrease. In the embodiment of the present invention, therefore, in order to minimize the motor power, assuming that the pressure ratios on the low- and high-pressure sides in the two-stage compression are equal, the various sizes of the low- and high-pressure pistons **4** and **8** are calculated as expressed by the following equations (4) and (5):

$$L = Ls1 + Ls2 \tag{4}$$

where L represents the total shaft power; Ls**1** represents the low-pressure required shaft power; and Ls**2** represents the high-pressure required shaft power.

$$Pm/P1 = P2/Pm \tag{5}$$

where Pm represents the intermediate absolute pressure; P**1** represents the low-pressure suction absolute pressure; and P**2** represents the high-pressure discharge absolute pressure.

[0039] On the basis of the equations (1) and (2), Ls**1** and Ls**2** are expressed as follows:

$$Ls1 = \frac{1}{\eta ad} \frac{k}{k-1} \times \frac{P1Qs1}{0.060} \times \left\{ \left(\frac{Pm}{P1} \right)^{\frac{k-1}{k}} - 1 \right\} \tag{6}$$

where Qs**1** represents the actual air volume in suction condition on the low-pressure side.

$$Ls2 = \frac{1}{\eta ad} \frac{k}{k-1} \times \frac{PmQs2}{0.060} \times \left\{ \left(\frac{P2}{Pm} \right)^{\frac{k-1}{k}} - 1 \right\} \tag{7}$$

where Qs**2** represents the actual air volume in suction condition on the high-pressure side.

[0040] Here, in consideration of the equation (5), using the constant K, the equations (6) and (7) are transformed as follows:

$$Ls1=P1\cdot Qs1\times K \tag{8}$$

$$Ls2=Pm\cdot Qs2\times K \tag{9}$$

[0041] The equation (3) is substituted into the equations (8) and (9) as:

$$Ls1=D1^2\times S1\times P1\times k \tag{10}$$

$$Ls2=D2^2\times S2\times Pm\times k \tag{11}$$

where D**1** represents the low-pressure bore diameter; D**2** represents the high-pressure bore diameter; S**1** represents the low-pressure stroke; and S**2** represents the high-pressure stroke.

[0042] The motor shaft power becomes minimum when the low- and high-pressure sides are equal in pressure ratio and shaft power. Therefore, the following equations are obtained by the equations (10) and (11):

$$D1^2\times S1\times P1 = D2^2\times S2\times Pm \tag{12}$$

$$\frac{Pm}{P1} = \frac{P2}{Pm} \quad Pm = \sqrt{\frac{P2}{P1}} \tag{13}$$

[0043] Based on the above, in the embodiment of the present invention, respective bore diameters and strokes of the low-pressure piston **4** and the high-pressure piston **8** are determined by the equations (12) and (13).

[0044] Note that, in the equations (12) and (13), P**2** must be made sufficiently large relative to P**1** to obtain compressed air at high pressure. That is to say, in the equation (12), Pm must be made sufficiently large relative to P**1**. In this case, D**1** must be made sufficiently large relative to D**2**, or S**1** must be made sufficiently large relative to S**2** to satisfy the equation (12) (to at least approximate the left- and right-hand values).

[0045] Here, the relationship between the low-pressure bore diameter D**1** and the high-pressure bore diameter D**2** will be described.

[0046] Firstly, the low-pressure bore diameter D**1** and the high-pressure bore diameter D**2** are preferably set to satisfy D**1**>D**2**. This is because, in the case of the two-stage compression, the pressure in the high-pressure compression chamber is higher than that in the low-pressure compression chamber, and therefore the area of the high-pressure piston **8** is reduced to reduce a load thereon so as to reduce a load F of the piston **8** applied in the longitudinal direction of the connecting rod **8**A from a top surface of the piston **8** and downsize the bearing provided on the outer periphery of the eccentric portion **13** on the center side of the connecting rod **8**A.

[0047] Meanwhile, as will be described later, in the embodiment of the present invention, in consideration of the center-of-gravity balance of the product, it is necessary to prevent the high-pressure bore diameter D**2** from being extremely small relative to the low-pressure bore diameter D**1**.

[0048] The center-of-gravity balance of the product is important, especially in portable air compressors. As shown in FIG. **3**, the portable air compressor is mounted with a compressor body **21** including the low- and high-pressure compressor units described in FIG. **1**, and the motor **3**, for example, on a pair of air tanks **22** (roughly in the center thereof). Also, various auxiliary components, in particular, a reducing valve **23** (**26**), a pressure gauge **24** (**27**), and an air outlet coupler **25** (**28**), which are large in mass, are mounted symmetrically with respect to the compressor body **21**, thereby performing layout in consideration of the center-of-gravity balance of the product.

[0049] With regard to the compressor body **21** with the largest mass thereamong, the center-of-gravity balance of the compressor body itself is also important.

[0050] As described above, the low- and high-pressure bore diameters D**1** and D**2** and the low- and high-pressure strokes S**1** and S**2** are determined so that the low- and high-pressure sides are made equal in shaft power and pressure ratio. The two-stage compression structure according to the embodiment of the present invention as shown in FIG. **1** includes the opposed two cylinders, and therefore the center-of-gravity balance of the compressor body **21** is greatly influenced by the lengths and sizes of the low- and high-pressure cylinders **14** and **17**, air valves **15** and **18**, cylinder heads **16** and **19** protruding from the crankcase **1**. As described above, it is a common practice to make the high-pressure bore diameter D**2** smaller than the low-pressure bore diameter D**1** to reduce the high-pressure piston load F. However, an extreme difference between the bore diameters D**1** and D**2** causes differences in size, not only between the connecting rods **4**A and **8**A but also between the cylinders **14** and **17** forming the respective compression chambers, the air valves **15** and **18**, and the cylinder heads **16** and **19**, thereby losing the lateral balance of the compressor body **21**. As a result, the center-of-gravity balance of the overall product is deteriorated. This also occurs in the case where there is a difference in lengths l (the lengths from the centers of the eccentric portions **7** and **13** to the leading ends, i.e. top surfaces, of the piston bodies **5** and **11**) of the low- and high-pressure connecting rods **4**A and **8**A. In this case, in addition to the deterioration in the center-of-gravity balance, the air cooled by the cooling fan **20** mounted on the compressor body **21** is less likely to reach the larger one in the length from the compressor body **21** to the cylinder head **16** (**19**), thereby producing an increase in temperature. As a result, performance deterioration and lifetime shortening might be caused. Also, when the compressor body **21** is mounted on the air tanks **22**, the cylinder head of the larger one in length protrudes from between the air tanks **22**. As a result, a disadvantage of an increase in dimension of the product might occur.

[0051] Therefore, in the light of the balance of the product, making an extreme difference in bore diameter D and connecting rod length l between the low- and high-pressure sides is preferably avoided.

[0052] Referring to FIG. **3**, the compressor body **21** is mounted in such a manner that the axial direction of the shaft **2** and the longitudinal direction of the air tanks **22** are perpendicular to each other, thereby satisfying both miniaturization and weight balance. However, the present invention is not limited thereto, and the compressor body **21** may be mounted in such a manner that the axial direction of the shaft **2** and the longitudinal direction of the air tanks **22** face in the same direction. In this case, by locating the shaft **2** in between the two air tanks **22**, the center-of-gravity balance can be ensured. Furthermore, the size is determined so that the cylinder head is prevented from protruding from between the air tanks **22**, thereby allowing realization of miniaturization.

[0053] In view of the above, it is necessary to make the high-pressure bore diameter D**2** smaller than the low-pressure bore diameter D**1** to reduce the piston load on the high-pressure side, while it is necessary to prevent D**1** from being extremely large relative to D**2** to keep balance. In the embodiment of the present invention, therefore, D**1**, D**2**, S**1**, and S**2** are determined in consideration of the foregoing.

[0054] In the equation (12), when S**1**<S**2** is set without making a large difference between D**1** and D**2**, Pm is not made sufficiently large relative to P**1**. Thus, in the equation (13), P**2** is not made sufficiently large relative to P**1**. More specifically, no compressed air at high pressure is obtainable. In the embodiment of the present invention, therefore, S**1**>S**2** is set, thereby making the low- and high-pressure shaft powers subequal and preventing the center-of-gravity balance between the high- and low-pressure sides from being largely lost, so that compressed air at high pressure is obtainable.

[0055] For example, in order to prevent the center-of-gravity balance from being largely lost, D**1** is preferably set twice or less as large as D**2**. In this case, if S**1**<S**2** is set, Pm becomes four times or less as large as P**1**, and thus compressed air at sufficiently high pressure cannot be obtained. Therefore, in order to obtain compressed air at sufficiently high pressure, S**1** must be designed large relative to S**2**, and S**1**>S**2** must be set.

[0056] Next, the oscillating motion of the piston **4** (**8**) and the connecting rod **4**A (**8**A) in the rocking piston mechanism will be described with reference to FIG. **4**.

[0057] As shown in FIG. **4**, while the piston **4** (**8**) and the connecting rod **4**A (**8**A) move toward the top dead center position and the bottom dead center position during the suction and discharge processes, the connecting rod **4**A (**8**A) becomes oblique with respect to a central shaft **20** of the cylinder **14** (**17**) by the eccentricity of the eccentric portion **7** (**13**).

[0058] The maximum tilt angle of the piston **4** (**8**) with respect to the cylinder **14** (**17**) during oscillation will be described. During oscillation of the piston **4** (**8**) within the cylinder **14** (**17**), when the maximum angle at which the longitudinal axis of the connecting rod **4**A (**8**A) is tilted with respect to the cylinder central shaft **20**, or the maximum angle at which a virtual plane normal to the cylinder central shaft **20** and the top surface of the piston attached to a connecting rod upper portion are tilted, is set as a tilt angle θ, the tilt angle θ is determined by, the length l of the connecting rod **4**A (**8**A), that is, the length from the center of the eccentric portion **7** (**13**) to the leading end (top surface) of the piston body **5** (**11**), and the eccentric amount r of the eccentric portion **7** (**13**) with respect to the shaft **2** of the motor **3**, as expressed by the following equation (14):

$$\theta = \tan^{-1}\left(\frac{r}{l}\right) \tag{14}$$

where l represents the connecting rod length; and r represents the eccentric amount (=stroke S/2).

[0059] Since S**1**>S**2** is set as described above, when the low-pressure eccentric amount is represented by r**1** and the high-pressure eccentric amount is represented by r**2**, the relationship therebetween is r**1**>r**2**. Also, since the low-pressure connecting rod length l**1** and the high-pressure connecting rod length l**2** are made subequal to stabilize the center of gravity of the compressor including the low- and high-pressure compressor units, the relationship of r**1**/l**1**>r**2**/l**2** is established. Thus, on the basis of the equation (14), when the low-pressure maximum tilt angle is represented by θ**1** and the high-pressure maximum tilt angle is represented by θ**2**, the relationship of θ**1**>θ**2** is established.

[0060] Therefore, regarding the center-of-gravity balance of the compressor body as described above, the tilt angle θ is minimized while considering differences in the bore diameter D and the connecting rod length l between the low- and high-pressure sides, and the low- and high-pressure bore diameters D and strokes S (the stroke S equals two multiplied by the eccentric amount r), and the connecting rod lengths l are determined by the foregoing equations while preventing the low-pressure maximum tilt angle θ**1** from becoming smaller than the high-pressure maximum tilt angle θ**2**. Thus, occurrence of the performance deterioration due to leakage of the compressed air is avoided, in particular, the performance deterioration on the high-pressure side can be prevented. Further, by equalizing the center-of-gravity balance of the compressor body, compressed air at high pressure can be obtained while preventing the deterioration in the center-of-gravity balance of the product and unequal cooling.

[0061] Hereinafter, further advantages obtained by designing the compressor so as to prevent the maximum tilt angle θ**2** of the high-pressure piston **8** with respect to the high-pressure cylinder **17** from becoming larger than the maximum tilt angle θ**1** of the low-pressure piston **4** with respect to the low-pressure cylinder **14** will be described.

[0062] In the embodiment of the present invention, the piston body **5** of the low-pressure piston **4** is mounted with the lip ring **6** that is flexible and has high followability to a change in the gap between the cylinder **14** and the lip ring **6**, thereby reducing the tendency to cause the performance deterioration due to leakage of compressed air from a gap formed in an oscillating direction of the piston **4**. On the other hand, the piston body **11** of the high-pressure piston **8** is mounted with the piston ring **10** requiring stiffness under high pressure and temperature. The piston ring **10** is poor in followability to a change in the gap relative to the lip ring **6**, and thus the performance deterioration due to leakage of compressed air is likely to be caused. It is therefore necessary to prevent the high-pressure side from being affected by leakage of compressed air, in consideration of the angle at which the piston **8** is tilted with respect to the cylinder **17**.

[0063] While the connecting rod **4**A (**8**A) moves toward the top dead center position and the bottom dead center position during the suction and discharge processes, the connecting rod **4**A (**8**A) becomes oblique with respect to the central shaft **20** of the cylinder **14** (**17**) by the eccentricity of the eccentric portion **7** (**13**). At this time, the contact surface between the lip ring **6** (the piston ring **10**) and the cylinder **14** (**17**) is formed in an elliptical shape (as viewed from the upper side of the cylinder central shaft) with the oscillating direction (the horizontal direction in FIG. **4**) corresponding to the long axis. Thus, a gap between a side **6**A (**9**A) in the oscillating direction of the lip ring **6** (the piston ring **10**) and the cylinder **14** (**17**) is likely to be formed. In particular, during the compression process in which the piston **4** (**8**) moves toward the top dead center position, the performance deterioration is likely to be caused due to leakage of compressed air from the gap.

[0064] Therefore, in the embodiment of the present invention, as described above, the compressor is designed so as to prevent the high-pressure maximum tilt angle θ**2** from becoming larger than the low-pressure maximum tilt angle

$\theta 1$. Therefore, a maximum gap T**2** formed between the high-pressure piston **8** and the cylinder **17** can be prevented from becoming larger than a maximum gap T**1** formed between the low-pressure piston **4** and the cylinder **14**. This allows prevention of occurrence of the performance deterioration due to leakage of compressed air, in particular, on the high-pressure compressor unit.

[0065] It should be noted that, in the embodiment of the present invention, the piston ring **10** having stiffness higher than the lip ring is mounted on the high-pressure piston body **11**, thereby also reducing the tendency to the performance deterioration due to abrasion.

[0066] Also, although the embodiment of the present invention has been described by using the case where the lip ring **6** is provided on the low-pressure piston body **5**, a piston ring may be provided in place of the lip ring, in the same manner as the high pressure side. In this case, followability to the gap is reduced on the low-pressure compressor unit, however, since pressure in the compression chamber of the low-pressure compressor unit is lower than that of the high-pressure compressor unit, the low-pressure piston body **5** with the piston ring may be adopted by taking a measure such as reducing the thickness of the piston ring to thereby increase the followability. When the low-pressure piston body **5** is provided with the piston ring, the performance deterioration due to abrasion of the piston **4** can be also prevented on the low-pressure compressor unit.

[0067] Furthermore, although the embodiment of the present invention adopting both the lip ring **9** and the piston ring **10** for use in the high-pressure piston body **11** has been described, if the performance deterioration due to abrasion becomes no problem, in place of the piston ring **10**, a single lip ring may be provided facing the high-pressure compression chamber, in the same manner as the low-pressure piston body **5**. Alternatively, such lip ring may be used together with the lip ring **9**. In this case, further reduction in weight and cost of the high-pressure piston **8** can be realized.

[0068] Moreover, regarding the pressure in the compression chamber, the gas load F to be applied toward the connecting rod central axis from the piston top surface (in the case where the piston body **5** (**11**) is the upper side and the connecting rod **4**A (**8**A) is the lower side), and a horizontal component f of the gas load F occur, thereby pressing the lip ring **6** (**9**) against the cylinder **14** (**17**). Thus, the surface abrasion of the lip ring **6** (**9**) or the cylinder **14** (**17**) advances, which can result in performance deterioration. Since the gas load F is large especially on the high-pressure compressor unit, it is necessary to reduce the surface abrasion of the lip ring **9**, the piston ring **10**, and the cylinder **17** on the high-pressure compressor unit to prevent the performance deterioration.

[0069] The horizontal component f of the gas load F increases as the tilt angle θ corresponding to the angle at which the piston **4** (**8**) is tilted with respect to the cylinder **14** (**17**) increases. In the embodiment of the present invention, as described above, the compressor is designed so as to prevent the high-pressure maximum tilt angle $\theta 2$ from becoming larger than the low-pressure maximum tilt angle $\theta 1$. Therefore, especially on the high-pressure compressor unit in which the surface abrasions of the lip ring **9**, the piston ring **10**, and the cylinder **17** could be a problem, it is possible to reduce the abrasions thereof and prevent the performance deterioration.

[0070] As described above, in the embodiment of the present invention, the low-pressure piston **4** and the high-pressure piston **5** are constructed according to the above-described dimensional relationship therebetween, thereby minimizing the tilt angle θ. Also, the low- and high-pressure bore diameters D and strokes S (the stroke S equals two multiplied by the eccentric amount r), and connecting rod lengths l are determined by the foregoing equations (1) to (14) so as to prevent the high-pressure maximum tilt angle $\theta 2$ from becoming larger than the low-pressure maximum tilt angle $\theta 1$, thereby allowing prevention of the performance deterioration due to leakage of compressed air, especially on the high-pressure side.

[0071] It should be understood that the foregoing description is only illustrative of the preferred embodiment of the present invention, and should not be taken as a limitation of the technical scope of the invention. That is to say, various embodiments of the invention are possible without departing from the technical principles or essential of the invention.

1. A reciprocating compressor comprising:

a low-pressure compressor unit having a low-pressure piston and a low-pressure cylinder for compressing air with the low-pressure piston reciprocating while oscillating within the low-pressure cylinder;

a high-pressure compressor unit having a high-pressure piston and a high-pressure cylinder for further compressing the air compressed in the low-pressure compressor unit with the high-pressure piston reciprocating while oscillating within the high-pressure cylinder; and

a motor for driving the low-pressure compressor unit and the high-pressure compressor unit,

wherein a maximum tilt angle during oscillation of the high-pressure piston is prevented from becoming larger than a maximum tilt angle during oscillation of the low-pressure piston.

2. The reciprocating compressor according to claim **1**, wherein a maximum gap formed between the high-pressure piston and the high-pressure cylinder during oscillation of the high-pressure piston is prevented from becoming larger than a maximum gap formed between the low-pressure piston and the low-pressure cylinder during oscillation of the low-pressure piston.

3. The reciprocating compressor according to claim **1**, wherein the low-pressure piston and the high-pressure piston each include: an eccentric portion coupled to a rotating shaft of the motor for performing rotational motion; a piston body for compressing air within the cylinder; and a connecting rod for connecting the eccentric portion and the piston body, the piston body being fixed to the connecting rod.

4. The reciprocating compressor according to claim **3**, wherein the piston body performs oscillating motion with the rotational motion of the eccentric portion.

5. The reciprocating compressor according to claim **1**, wherein a bore diameter of the high-pressure piston is prevented from being made larger than a bore diameter of the low-pressure piston.

6. The reciprocating compressor according to claim **3**, wherein a length from a center of the eccentric portion to a leading end of the low-pressure piston and a length from a center of the eccentric portion to a leading end of the high-pressure piston are made subequal.

7. The reciprocating compressor according to claim **1**, wherein the low-pressure piston and the high-pressure piston each include a piston body for compressing air within the cylinder, and

wherein the piston body of the low-pressure piston is provided with a lip ring, and the piston body of the high-pressure piston is provided with a piston ring.

8. The reciprocating compressor according to claim **1**, wherein the low-pressure piston and the high-pressure piston each include a piston body for compressing air within the cylinder, and

wherein the piston body of the low-pressure piston is provided with a piston ring, and the piston body of the high-pressure piston is provided with a piston ring.

9. A reciprocating compressor comprising:

a motor having a rotating shaft;

a low-pressure compressor unit having a low-pressure cylinder and a low-pressure piston for compressing air; and

a high-pressure compressor unit having a high-pressure cylinder and a high-pressure piston for further compressing the air compressed in the low-pressure compressor unit,

the low-pressure piston and the high-pressure piston each including: an eccentric portion for performing eccentric motion with rotation of the rotating shaft of the motor; a connecting rod extending from the eccentric portion; and a piston body provided on a leading end of the connecting rod,

wherein, when an eccentric amount of the eccentric portion of the low-pressure piston with respect to the rotating shaft of motor is represented by **r1** and an eccentric amount of the eccentric portion of the high-pressure piston with respect to the rotating shaft of motor is represented by r**2**, the low- and high-pressure pistons are formed in such a manner that **r1**>r**2** is established.

10. The reciprocating compressor according to claim **9**, wherein a maximum gap formed between the high-pressure piston and the high-pressure cylinder during oscillation of the high-pressure piston is made smaller than a maximum gap formed between the low-pressure piston and the low-pressure cylinder during oscillation of the low-pressure piston.

11. The reciprocating compressor according to claim **9**, wherein the low-pressure piston and the high-pressure piston each include the piston body fixed to the connecting rod.

12. The reciprocating compressor according to claim **9**, wherein the respective piston bodies reciprocate while oscillating within the high- and low-pressure cylinders with eccentric motion of the eccentric portion.

13. The reciprocating compressor according to claim **9**, wherein a bore diameter of the high-pressure piston is made smaller than a bore diameter of the low-pressure piston.

14. The reciprocating compressor according to claim **9**, wherein, when a length from a center of the eccentric portion of the low-pressure piston to a leading end of the piston body of the low-pressure piston is represented by l**1** and a length from a center of the eccentric portion of the high-pressure piston to a leading end of the piston body of the high-pressure piston is represented by l**2**, the low- and high-pressure pistons are formed so that **r1**/l**1**>r**2**/l**2** is established.

15. The reciprocating compressor according to claim **9**, wherein the piston body of the low-pressure piston is provided with a lip ring, and the piston body of the high-pressure piston is provided with a piston ring.

16. The reciprocating compressor according to claim **9**, wherein the piston body of the low-pressure piston is provided with a piston ring, and the piston body of the high-pressure piston is provided with a piston ring.

17. The reciprocating compressor according to claim **2**, wherein the low-pressure piston and the high-pressure piston each include: an eccentric portion coupled to a rotating shaft of the motor for performing rotational motion; a piston body for compressing air within the cylinder; and a connecting rod for connecting the eccentric portion and the piston body, the piston body being fixed to the connecting rod.

18. The reciprocating compressor according to claim **10**, wherein the low-pressure piston and the high-pressure piston each include the piston body fixed to the connecting rod.

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