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(54) Suction opening of a screw compressor

Ansaugöffnung eines Schraubenverdichters

Ouverture d'aspiration d'un compresseur rotatif à vis

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Description

BACKGROUND OF THE INVENTION

Field of the invention

[0001] The present invention relates to a screw compressor, and particularly relates to a shape of an axial suction port of an oil free type screw compressor including a timing gear.

Description of related art

[0002] In order to enhance energy efficiency and volume efficiency of a screw compressor, a great deal of effort has been spent so far. While various factors determine the performance, it has been learned from the recent study that a contour shape of an axial suction port affects the volume efficiency of the screw compressor. In general, when a sufficient opening area or an opening time cannot be ensured in an axial suction port, a suction flow rate decreases to cause reduction in volume efficiency, but when an excessive opening area and an excessive opening time are set on the contrary, the fluid which is once sucked into a working chamber flows back in a compression process, and the suction flow rate decreases to reduce the volume efficiency as a result.

[0003] JP-A-06-288369 describes a contour shape preferable for an axial suction port of a screw compressor including a working chamber in which a volumetric change is temporarily discontinued. Further, JP-A-10-9164 describes a method for increasing a suction flow rate by performing suction operation intermittently.

[0004] JP-A-06-288369 and JP-A-10-9164 have not been applied widely and generally because certain conditions are required for the structure of the screw compressor to which those can be applied and for the usage thereof. Accordingly, there has been the problem of embodying the structure for increasing the suction amount that can be applied to many screw compressors.

[0005] US 2457314 A discloses a rotary screw wheel device in which intermeshing rotors having helical lands and grooves cooperate with each other and with an enclosing casing to form working chambers which vary in volume as the rotors revolve.

[0006] US 2481527 A discloses rotary multiple helical rotor machine which provides working chamber between intermeshing rotors of the screw wheel type mounted in a suitable casing.

[0007] The teaching of the two latter documents generally relates to the effect of the shape of the suction port opening and closing on pressure waves travelling along the working chambers.

BRIEF SUMMARY OF THE INVENTION

[0008] Thus, in view of the above described problem, an object of the present invention is to propose a configuration which can enhance energy efficiency and volume efficiency in a screw compressor of an ordinary configuration.

[0009] The above object is achieved by the features of claim 1.

[0010] In order to solve the above described problem, the invention provides a screw compressor including a male rotor having a helical lobe, a female rotor having a helical lobe, a casing forming a bore for accommodating the lobes of the male female rotors in a state where the lobes mesh with each other, an axial suction port provided on a suction side of the casing, and a delivery port provided on a delivery side of the casing, wherein the axial suction port is configured by a contour line including a line along a male lobe profile and a line along a female lobe profile, and the line along the male lobe profile is disposed at a predetermined displacement angle toward a rotational direction side of the male rotor from a following side contour line position of a male lobe groove in the rotational angle of the male rotor, at which following side contour line position the volume of a working chamber formed by being enclosed by the male lobe groove of the male rotor, a female lobe groove of the female rotor and the bore is maximum.

[0011] Further, the invention provides a screw compressor including a male rotor having a helical lobe, a female rotor having a helical lobe, a casing forming a bore for accommodating the lobes of the male and female rotors in a state where the lobes mesh with each other, an axial suction port provided on a suction side of the casing, and a delivery port provided on a delivery side of the casing, wherein the axial suction port is configured by a contour line including a line along a male lobe profile and a line along a female lobe profile, and the line along the female lobe profile is disposed at a predetermined displacement angle toward a rotational direction side of the female rotor from a following side contour line position of a female lobe groove in the rotational angle of the female rotor, at which following side contour line the volume of a working chamber formed by being enclosed by a male lobe groove of the male rotor, the female lobe groove of the female rotor and the bore is maximum.

[0012] According to the present invention, the compressor which increases the suction amount and is highly efficient can be realized.

[0013] Other objects, features and advantages of the invention will become apparent from the following description of the embodiments of the invention taken in conjunction with the accompanying drawings.

BRIEF DESCRIPTION OF THE SEVERAL VIEWS OF THE DRAWINGS

[0014]

Figs. 1A and 1B show a pair of rotors and a contour of a suction port of embodiment 1;

Fig. 2 is a development of a bore outer circumferential surface of embodiment 1;

Fig. 3 is a schematic sectional view of an ordinary screw compressor;

Figs. 4A and 4B show a pair of rotors and a contour of a suction port of an ordinary screw compressor;

Fig. 5 shows graphs of a volumetric change and a suction flow rate of a working chamber of a screw compressor; and

Fig. 6 is a view explaining an inflow sectional area shape.

DETAILED DESCRIPTION OF THE INVENTION

[0015] Before describing an embodiment of the present invention, a general configuration of a screw compressor will be described by using Figs. 3, 4A and 4B.

[0016] Fig. 3 is a schematic sectional view of a screw compressor. Reference numeral 1 designates a male rotor having a helical lobe, and reference numeral 2 designates a female rotor having a helical lobe. As shown here, in a screw compressor, the male rotor 1 and the female rotor 2 are housed in a bore 4 formed inside a casing 3 in a state where those mesh with each other. The bore 4 is of a shape of two cylinders of which parts overlapping each other, and more specifically, is configured by four surfaces which are a cylindrical outer peripheral surface with which a lobe portion of the male rotor 1 is covered, a cylindrical outer peripheral surface with which a lobe portion of the female rotor 2 is covered, a suction end 7, and a delivery end 8. The suction end 7 is provided with an axial suction port 11 which will be described later, and the delivery end 8 is provided with a delivery port not illustrated. Further, a timing gear 10 which is provided at one end of a shaft of the male rotor 1 and a timing gear 10 which is provided at one end of a shaft of the female rotor 2 are disposed to mesh with each other, so that the female rotor 2 is configured to be rotationally driven synchronously with rotational drive of the male rotor 1. A space where a male lobe groove 5 of the male rotor 1 and a female lobe groove 6 of the female rotor 2 communicate with each other is covered with the bore 4, and forms a working chamber 9 which is a closed space. Since there are a plurality of spaces where the male lobe groove 5 and the female lobe groove 6 communicate with each other, a plurality of working chambers 9 are formed. Each of the working chambers 9 moves toward the direction of the delivery end 8 from the suction end 7 in accordance with rotation of both rotors. In an oil free type screw compressor, backlash of the timing gears 10 is designed to be smaller than backlash of the lobe portions of the male rotor 1 and the female rotor 2, and the lobe portions of the male rotor 1 and the female rotor 2 are not brought into contact with each other. Therefore, each of the working chambers 9 is not a closed space in strict meaning, and accordingly is connected to the adjacent working chambers through a clearance. However, the amount of gas which leaks into the adjacent working chamber 9 through the clearance is so small that it can be ignored. Further, in a non-oil free type screw compressor, the clearance is generally sealed by oil, water or the like, and the effect of leakage between the adjacent working chambers 9 can be ignored. Therefore, hereinafter, description will be made with the assumption that each of the working chambers 9 is an independent space.

[0017] Figs. 4A and 4B are views explaining an ordinary shape of the axial suction port 11 which is provided in the suction end 7 of the bore 4. As shown by the arrows, the male rotor 1 rotates clockwise, whereas the female rotor 2 rotates counterclockwise. The working chamber 9 which is generated in a position where both rotors on the suction end 7 are in contact with each other enlarges its inner volume while a leading head of the working chamber 9 moves toward the delivery end 8 in accordance with rotation of both rotors. During this time period, gas to be compressed is supplied to the working chamber 9 via the axial suction port 11. A contour line 15 on the male rotor 1 side and a contour line 18 on the female rotor 2 side of the axial suction port 11 are provided at a position where the inner volume of the working chamber 9 is maximum. More specifically, during the time period in which the inner volume of the working chamber 9 is enlarged, the gas to be compressed is supplied to the working chamber 9 communicating with the axial suction port 11, and during the time period in which the inner volume of the working chamber 9 is reduced, new gas to be compressed is not supplied to the working chamber 9 which does not communicate with the axial suction port 11. Thereafter, in accordance with rotation of both rotors, the inner volume of the working chamber 9 is reduced, and therefore, the gas to be compressed in the working chamber 9 is compressed. The gas to be compressed which has been compressed is discharged from the delivery port which is provided in the delivery end 8.

(Embodiment 1)

[0018] Embodiment 1 of the present invention will be described using Figs. 1A and 1B. The components equivalent to those of Figs. 3 and 4 are assigned with the same reference numerals, and the description thereof will be omitted. Fig. 1A is a view explaining a shape in embodiment 1 of the axial suction port 11 provided in the suction end 7 of the bore 4. As shown by the arrows, the male rotor 1 rotates clockwise, whereas the female rotor 2 rotates counterclockwise. The working chamber 9 which is the nearest to the delivery side and shown by the oblique lines is in a position where the working chamber 9 has the maximum volume, and the working chambers which are located on the suction side from that position are in the suction process by enlargement of the volume.

[0019] As shown in Fig. 1B, the contour line of the axial suction port 11 is configured by seven contour lines that are a line 13 of a meshing portion, a circular arc 14 along a male lobe bottom diameter, the line 15 along a male lobe profile, a circular arc 16 along a male lobe tip diameter, a circular arc 17 along a female lobe tip diameter, the line 18 along a female lobe profile and a circular arc 19 along a female lobe bottom diameter. Among those, the line 15 along the male lobe profile and the line 18 along the female lobe profile, which are the contour lines that have a large effect on the performance of the compressor, will be described in detail.

[0020] The working chamber 9 with the maximum volume shown by hatching in Fig. 1A is in contact with the suction end 7 and the delivery end 8 at both male and female sides. The contour of the groove of each of the rotors in the suction end 7 will be studied by dividing the contour into two on an advance side and a following side with respect to the rotor rotation. Among those, the following side has a large effect on the performance of the compressor, and therefore, attention will be paid on a following side contour line 21 of the male lobe groove 5 and a following side contour line 22 of the female lobe groove 6 hereinafter.

[0021] As shown in Fig. 1B, the line 15 along the male lobe profile is provided at a position which is displaced clockwise by $\Delta f1$ from a position of the following side contour line 21 at the timing when the working chamber 9 has the maximum volume. The positional precision of the line 15 along the male lobe profile has to be within 1/20 of the rotor diameter.

[0022] Further, the line 18 along the female lobe profile is provided at a position which is displaced counterclockwise by $\Delta f2$ from a position of the following side contour line 22 at the timing when the working chamber 9 has the maximum volume. The positional precision of the line 18 along the female lobe profile has to be within 1/20 of the rotor diameter.

[0023] Next, the positional relation of the working chamber 9 and the axial suction port 11 will be described by using the development of the bore 4 as shown in Fig. 2. In the development of the bore 4 shown in Fig. 2, the right side is a development of the male side cylinder, and the left side is a development of the female side cylinder. A lower end of the development is the suction end 7, whereas an upper end is the delivery end 8. The axial suction port 11 which is adjacent to the suction end 7, and has both ends defined by the line 15 along the male lobe profile and the line 18 along the female lobe profile is opened.

[0024] A vertical line in the center which is designated by reference numeral 31 is an expansion side cusp which is on an expansion side, of intersection lines of the male side cylinder and the female side cylinder. Vertical lines at both left and right sides designated by reference numerals 32 are compression side cusps which are on a compression side, of the intersection lines of the male side cylinder and the female side cylinder of the bore 4. Further, Oblique lines 24 and 25 and the oblique lines parallel with them show lobe tip lines of each of the rotors. Among the working chambers formed between the respective lobe tip lines, the working chambers facing the axial suction port 11 take the gas to be compressed in, and the working chambers which do not face the axial suction port 11 do not take the gas to be compressed in.

[0025] When both male and female rotors rotate, the working chambers move upward as shown by the arrows, and the internal volumes are enlarged or decreased.

[0026] With use of Fig. 5, a suction stroke of the gas to be compressed to the working chamber of the screw compressor having the configuration described above will be described. (a) shows the volume of the working chamber when the rotors are rotated. (b) shows the volumetric change rate of the working chamber that is obtained by differentiating (a). (c) shows a volume flow rate which is sucked by the working chamber. In Fig. 5, it is shown that the axial suction port 11 of the present embodiment is larger than the ordinary axial suction port by the fact that the axial suction port opening time period is longer than the working chamber volume increasing time period.

[0027] First, a rotational angle of the rotors that is generated at a contact point of the suction end 7 and the expansion side cusp 31 by the working chamber formed by the male lobe groove and the female lobe groove communicating with each other is set as θ_0. In the time period from the rotational angles θ_0 to θ_1, the male lobe groove and the female lobe groove are connected to each other at the opening of the working chamber in the suction end 7. At the time of the rotational angle θ_1, the opening of the working chamber in the suction end 7 is separated to a male lobe groove side and a female lobe groove side. As shown in (c) of Fig. 5, during the time period from the rotational angles θ_1 to θ_2, the volume flow rate which is sucked into the working chamber is large, but since the gas to be compressed is sucked into the working chamber through two openings at the male rotor side and the female rotor side, the passage pressure loss is small, and smooth suction can be realized.

[0028] When the tip end of the working chamber reaches the delivery end 8 at the timing of the rotational angle θ_2, the change rate of the volume of the working chamber and the volume flow rate to be sucked are gradually reduced as shown in (b) and (c) of Fig. 5. In the meanwhile, the volume of the working chamber becomes maximum at the timing of the rotational angle θ_3, and after θ_3, the volume of the working chamber changes and decreases. What should be noted here is the point that even if the volume of the working chamber 9 changes and decreases, suction of the gas to be compressed to the working chamber continues up to the rotational angle θ_4.

[0029] The reason of the above will be described by using Fig. 6. Fig. 6 shows the state of the flow of the gas to be compressed which is sucked to the rotors through the suction port from the suction side casing. Since suction in the state close to a stationary state where the inertia effect accompanying the suction is small is conventionally assumed, the flow velocity of the gas to be compressed sucked from the rotors has a sufficiently large inertia effect, and in the present embodiment, the volume efficiency is enhanced by increasing the suction quantity by adopting a shape of the axial suction port 11 which can utilize the inertia effect.

[0030] Hereinafter, the reason why the suction of the gas to be compressed is possible even after the working chamber volume changes and decreases will be discussed in detail.

[0031] When a radius of the male rotor groove bottom is Rm, a male rotor wrap angle is θm, a radius of the female rotor groove bottom is Rf, a female rotor wrap angle is θf, and a axial length of the rotor is L, a groove length L' in the axial direction of the working chamber can be expressed assuming that the groove bottom radius is minimum as follows (θ is in a radian unit).

$$L' = \sqrt{\ }\{(2\pi R \times \theta/2\pi)^2 + L^2\}$$

[0032] The gas to be compressed which has flown into the working chamber is considered to have a temperature and a pressure substantially equal to a port temperature and a port pressure before shifting to the compression process. A sound velocity of the gas to be compressed at this time is set as "a". With respect to the case of a non oil free type, the sound velocity is defined by the sound velocity corrected with a vapor quantity of oil, water or the like.

[0033] The working chamber volume is enlarged the stage at which the working chamber volume is minimum in accordance with rotation of the rotor from, and the volume enlargement of the working chamber becomes maximum at the stage at which the working chamber reaches the delivery end. At this time, the fact that the working chamber reaches the delivery end is transmitted to the suction side under the sound velocity condition of the gas to be compressed. A time lag Δt based on the compressibility of the gas is expressed by using the groove length L' in the axial direction of the working chamber, for example, as follows.

$$\Delta t = \sqrt{\ }\{(2\pi R \times \theta/2\pi)^2 + L^2\}/a$$

[0034] When the rotational speed is set as ω, a displacement angle Δf is expressed as follows.

$$\Delta f = \omega \times \sqrt{\ }\{(2\pi R \times \theta/2\pi)^2 + L^2\}/a$$

[0035] More specifically, in the case that a male rotor side groove length L'm is shorter than a female rotor side groove length L'f, for example, $\Delta t = 3.7\times10^{-4}$sec where the female rotor side groove bottom radius Rf=30mm, the wrap angle θf=150°, the rotor axial length L=100 mm, air is used as the gas to be compressed, and the sound velocity a=340 m/s, and if the rotational speed of the female rotor is set as 200 rev/sec, the angle Δf which moves during the delay time is Δt=27°. Accordingly, by delaying the closing timing of the axial port theoretically in the range of Δf=27° or less from the angle at which the working chamber forms the maximum volume, the suction opening sectional area and the suction time can be increased, and the suction volume efficiency can be enhanced.

[0036] According to the screw compressor of embodiment 1 described above, the working chamber in the suction process can suck the gas to be compressed smoothly with low pressure loss, and the suction quantity can be increased while backflow of the gas to be compressed which is once sucked from the working chamber is prevented.

[0037] As for the kind of the gas to be compressed, the present invention is applicable to any kind of gas. Further, the condition is defined by the shape (mainly, the length) of the working chamber 9 formed by the male rotor 1 and the female rotor 2, the present invention is applicable to various rotor lobe shapes, and the material of the rotor is not limited.

(Example 1 not falling within the scope of the claims)

[0038] In embodiment 1, the axial suction port 11 is closed on the male and female sides at the same timing, but in example 1 only the contour line 15 which is the male side contour line of the axial suction port 11 is closed at the closing timing described in embodiment 1.

[0039] By this configuration, even in the case where "the maximum displacement angle Δf at the female rotor side << the maximum displacement angle Δf at the male rotor side", enhancement in efficiency can be realized.

(Example 2 not falling within the scope of the claims)

[0040] In embodiment 1, the axial suction port 11 is closed on the male and female sides at the same timing, whereas in example 2 only the contour line 18 which is the female side contour line of the axial suction port 11 is closed at the closing timing described in embodiment 1.

[0041] By this configuration, even in the case where "the maximum displacement angle Δf at the male rotor side <<than the maximum displacement angle Δf at the female rotor side", enhancement in efficiency can be realized.

Claims

1. A screw compressor, comprising:

a male rotor (1) having a helical lobe;
a female rotor (2) having a helical lobe;
a casing (3) in which a bore (4) is formed for accommodating the lobe of the male rotor and the lobe of the female rotor in a state where the lobes mesh with each other;
an axial suction port (11) provided on a suction side of the casing; and

a delivery port provided on a delivery side of the casing, wherein
the axial suction port (11) is defined by a contour line including a line (15) along a male lobe profile and a line (18) along a female lobe profile,
characterized in that
the contour line of the axial suction port is configured by seven contour lines consisting of a line (13) of a meshing portion, a circular arc (14) along a male lobe bottom diameter, the line (15) along the male lobe profile, a circular arc (16) along a male lobe tip diameter, a circular arc (17) along a female lobe tip diameter, the line (18) along the female lobe profile and a circular arc (19) along a female lobe bottom diameter,
the line (15) along the male lobe profile is arranged so as to be displaced by a predetermined displacement angle toward a rotational direction side of the male rotor from a following side contour line position (21) of a male lobe groove (5) at a rotational angle of the male rotor (1) at the time when the volume of a working chamber (9) formed by being enclosed by the male lobe groove (5) of the male rotor, a female lobe groove (6) of the female rotor and the bore (4) is maximum,
the line (18) along the female lobe profile is arranged so as to be displaced by a predetermined displacement angle toward a rotational direction side of the female rotor from a following side contour line position (22) of the female lobe groove (6) at a rotational angle of the female rotor (2) at the time when the volume of the working chamber (9) is maximum,
the axial suction port (11) is structured to close the suction port after a time required for gas to be compressed to move by a lobe portion length of the rotor at the speed of sound from the time when the working chamber reaches a delivery end (8), further from a position in a volume reduction process through the rotational angle at which the volume of the working chamber is substantially maximum, and the contour line of the axial suction port (11) is structured to include both of a portion along a contour line in the end that is on a rear side of rotation of a male rotor groove which forms a part of the working chamber (9), and a portion along a contour line in the end that is on a rear side of rotation of a female rotor groove, from a rotational angle of the rotors where the working chamber (9) is closed after a time required for gas to be compressed to move at the speed of sound by a lobe portion length of the rotor after the working chamber (9) reaches a delivery end after the rotational angle of the rotor at which the volume of the working chamber is substantially maximum, and is structured to be opened on an opposite side to the rotating direction of both male and female rotors from the contour lines and to be closed on a rotating direction side with a bore end.

Patentansprüche

1. Schraubenkompressor, der aufweist:

einen Rotor (1) mit einem konvexen Schraubenprofil;
einen Rotor (2) mit einem konkaven Schraubenprofil;
ein Gehäuse (3), in dem eine Bohrung (4) ausgebildet ist zur Aufnahme des Schraubenprofils des Rotors mit konvexem Profil und des Schraubenprofils des Rotors mit konkavem Profil in einem Zustand, in dem die Schraubenprofile miteinander in Eingriff sind;
einer axialen Ansaugöffnung (11), die auf der Ansaugseite des Gehäuses vorgesehen ist, und eine Auslassöffnung, die auf der Auslassseite des Gehäuses vorgesehen ist, wobei
die axiale Ansaugöffnung (11) durch eine Konturlinie definiert ist, die eine Linie (15) entlang einem konvexen Schraubenprofil und eine Linie (18) entlang einem konkaven Schraubenprofil einschließt,
dadurch gekennzeichnet, dass
die Konturlinie der axialen Ansaugöffnung aus sieben Konturlinien aufgebaut ist, die bestehen aus einer Linie (13) eines Eingriffs-Abschnitts, einem Kreisbogen (14) entlang einem Durchmesser des Bodens des konvexen Schraubenprofils, der Linie (15) entlang dem konvexen Schraubenprofil, einem Kreisbogen (16) entlang einem Durchmesser der Spitze des konvexen Schraubenprofils, einem Kreisbogen (17) entlang einem Durchmesser der Spitze des konkaven Schraubenprofils, der Linie (18) entlang dem konkaven Schraubenprofil und einem Kreisbogen (19) entlang einem Durchmesser des Bodens des konkaven Schraubenprofils,
die Linie (15) entlang dem konvexen Schraubenprofil so ausgebildet ist, dass sie um einen vorgegebenen Versetzungswinkel zu der Drehrichtungsseite des konvexen Rotors hin von einer folgeseitigen Konturlinienposition (21) einer Vertiefung (5) des konvexen Profils bei einem Drehwinkel des konvexen Rotors (1) zu der Zeit versetzt ist, wenn das Volumen einer Arbeitskammer (9), die durch Einschließen durch die Vertiefung (5) des konvexen Profils des konvexen Rotors, eine Vertiefung (6) des konkaven Profils des konkaven Rotors und die Bohrung (4) gebildet wird, maximal ist,
die Linie (18) entlang dem konkaven Schraubenprofil so ausgebildet ist, dass sie um einen

vorgegebenen Versetzungswinkel zu der Drehrichtungsseite des konkaven Rotors hin von einer folgeseitigen Konturlinienposition (22) einer Vertiefung (6) des konkaven Profils bei einem Drehwinkel des konkaven Rotors (2) zu der Zeit versetzt ist, wenn das Volumen der Arbeitskammer (9) maximal ist,
die axiale Ansaugöffnung (11) so ausgebildet ist, dass die Ansaugöffnung nach einer Zeit geschlossen ist, die erforderlich ist, um das zu komprimierende Gas durch die Länge eines Profilabschnitts des Rotors bei der Schallgeschwindigkeit von der Zeit zu bewegen, wenn die Arbeitskammer das Auslassende (8) erreicht, ferner von einer Position in einem Prozess der Volumenverringerung durch den Drehwinkel, bei dem das Volumen der Arbeitskammer im Wesentlichen maximal ist, und
die Konturlinie der axialen Ansaugöffnung (11) so ausgebildet ist, dass sie sowohl einen Abschnitt entlang einer Konturlinie an dem Ende, das sich auf der Rückseite der Rotation einer Vertiefung des konvexen Rotors befindet, die eine Teil der Arbeitskammer (9) bildet, als auch einen Abschnitt entlang einer Konturlinie an dem Ende, das sich auf der Rückseite der Rotation einer Vertiefung des konkaven Rotors befindet, einschließt, von einem Drehwinkel der Rotoren, wo die Arbeitskammer (9) nach einer Zeit geschlossen ist, die erforderlich ist, um das zu komprimierende Gas bei der Schallgeschwindigkeit durch die Länge eines Profilabschnitts des Rotors zu bewegen, nachdem die Arbeitskammer (9) das Auslassende erreicht nach dem Drehwinkel des Rotors, bei dem das Volumen der Arbeitskammer im Wesentlichen maximal ist, und sie so ausgebildet ist, dass sie auf der der Drehrichtung entgegengesetzten Seite des konvexen und des konkaven Rotors von den Konturlinien offen ist und auf der Seite der Drehrichtung mit dem Ende der Bohrung geschlossen ist.

Revendications

1. Compresseur à vis, comprenant :

un rotor mâle (1) ayant un lobe hélicoïdal ;
un rotor femelle (2) ayant un lobe hélicoïdal ;
un carter (3) dans lequel un alésage (4) est formé pour recevoir le lobe du rotor mâle et le lobe du rotor femelle dans un état où les lobes s'engrènent l'un avec l'autre ;
un orifice d'aspiration axial (11) prévu sur un côté d'aspiration du carter ; et
un orifice de refoulement prévu sur un coté de refoulement du carter, dans lequel
l'orifice d'aspiration axial (11) est défini par une ligne de contour incluant une ligne (15) le long d'un profil de lobe mâle et une ligne (18) le long d'un profil de lobe femelle,
caractérisé en ce que
la ligne de contour de l'orifice d'aspiration axial est configuré par sept lignes de contour consistant en une ligne (13) d'une partie d'engrenage, un arc circulaire (14) le long d'un diamètre de fond de lobe mâle, la ligne (15) le long du profil de lobe mâle, un arc circulaire (16) le long d'un diamètre de pointe de lobe mâle, un arc circulaire (17) le long d'un diamètre de pointe de lobe femelle, la ligne (18) le long du profil de lobe femelle et un arc circulaire (19) le long d'un diamètre de fond de lobe femelle,
la ligne (15) le long du profil de lobe mâle est agencée de manière à être déplacée d'un angle de déplacement prédéterminé vers un côté de sens de rotation du rotor mâle depuis une position (21) de ligne de contour de côté suivant d'une rainure (5) de lobe mâle à un angle de rotation du rotor mâle (1) au moment où le volume d'une chambre de travail (9) formée en étant renfermée par la rainure (5) de lobe mâle du rotor mâle, une rainure (6) de lobe femelle du rotor femelle et l'alésage (4) est maximum,
la ligne (18) le long du profil de lobe femelle est agencée de manière à être déplacée d'un angle de déplacement prédéterminé vers un côté de sens de rotation du rotor femelle depuis une position (22) de ligne de contour de côté suivant de la rainure (6) de lobe femelle à un angle de rotation du rotor femelle (2) au moment où le volume de la chambre de travail (9) est maximum,
l'orifice d'aspiration axial (11) est structuré pour fermer l'orifice d'aspiration après une durée requise pour qu'un gaz devant être compressé se déplace d'une longueur de partie de lobe du rotor à la vitesse du son depuis un moment où la chambre de travail atteint une extrémité de refoulement (8), en outre depuis une position dans un processus de réduction de volume par l'intermédiaire de l'angle de rotation auquel le volume de la chambre de travail est sensiblement maximum, et
la ligne de contour de l'orifice d'aspiration axial (11) est structurée pour inclure les deux d'une partie le long d'une ligne de contour dans l'extrémité qui est sur un côté arrière de rotation d'une rainure de rotor mâle qui forme une partie de la chambre de travail (9), et d'une partie le long d'une ligne de contour dans l'extrémité qui est sur un côté arrière de rotation d'une rainure de rotor femelle, depuis un angle de rotation des rotors où la chambre de travail (9) est fermée après une durée requise pour qu'un gaz devant

être compressé se déplace à la vitesse du son d'une longueur de partie de lobe du rotor après que la chambre de travail (9) a atteint une extrémité de refoulement après que l'angle de rotation du rotor auquel le volume de la chambre de travail est sensiblement maximum, et est structurée pour être ouverte sur un côté opposé au sens de rotation des deux rotors mâle et femelle depuis les lignes de contour et pour être fermée sur un côté de sens de rotation avec une extrémité d'alésage.

FIG.1A

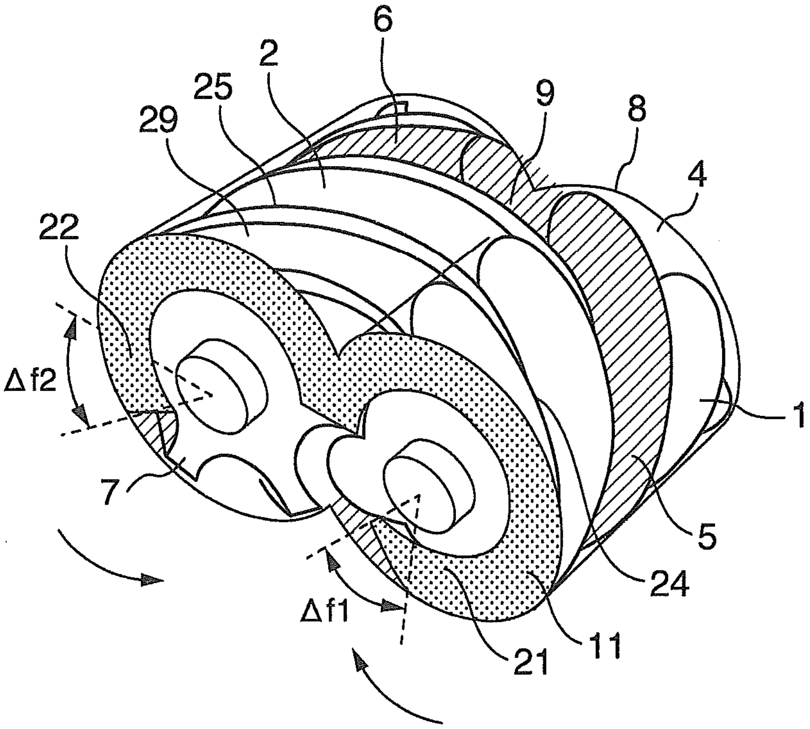


FIG.1B

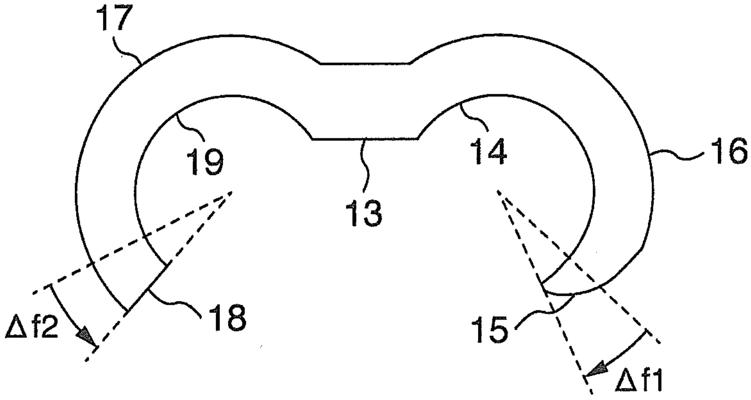


FIG.2

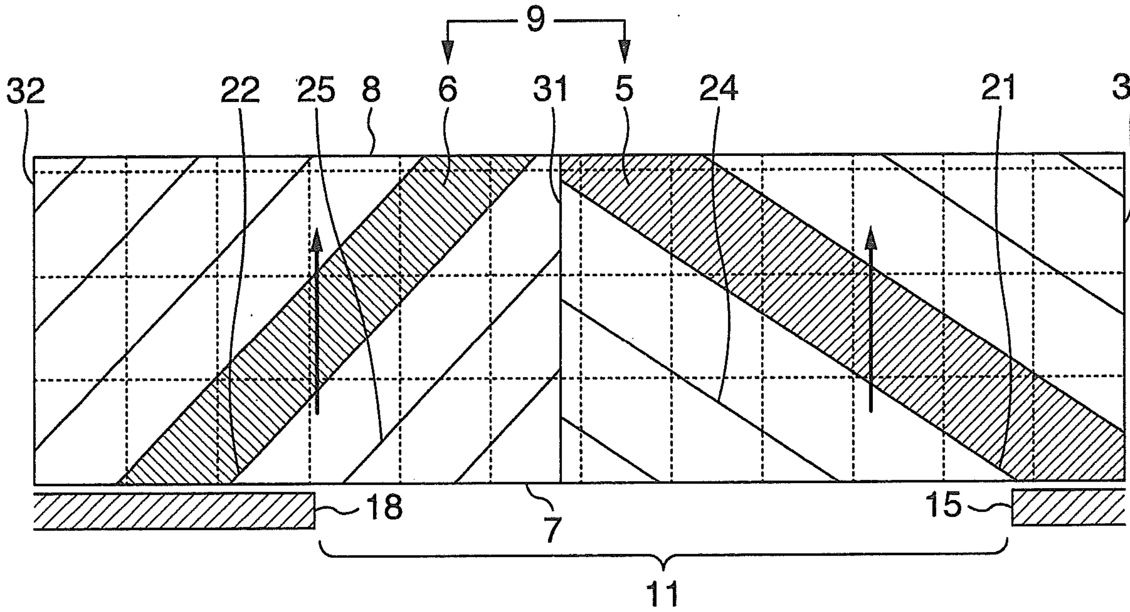


FIG.3

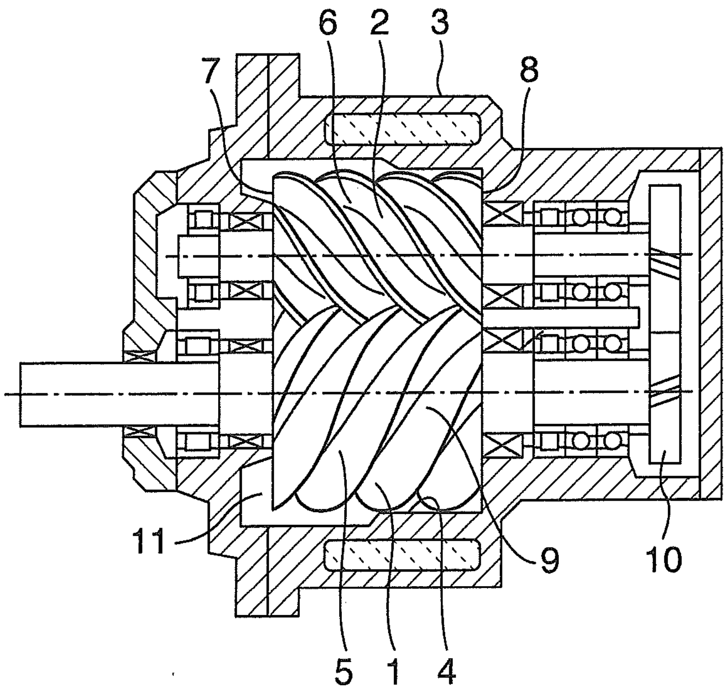


FIG.4A

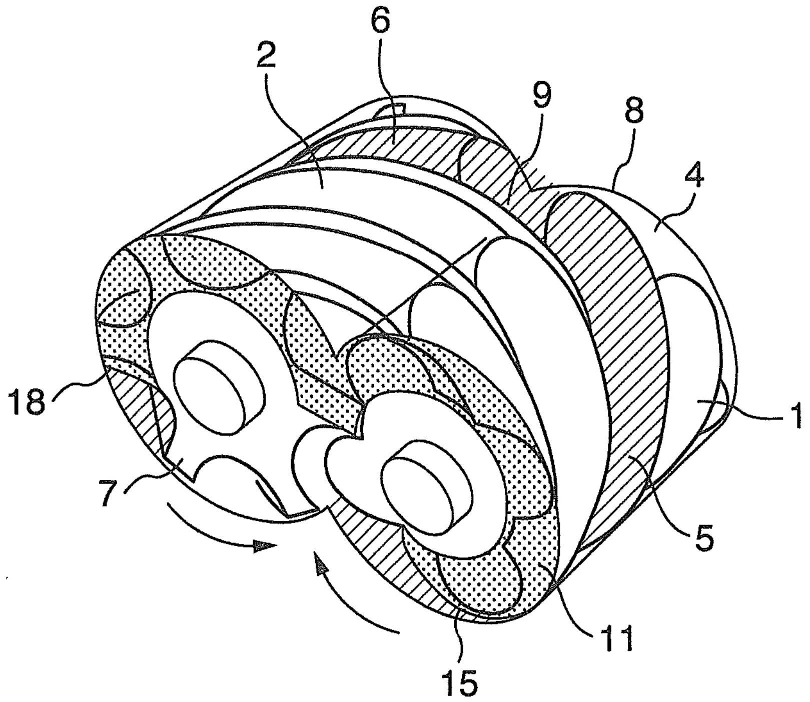


FIG.4B

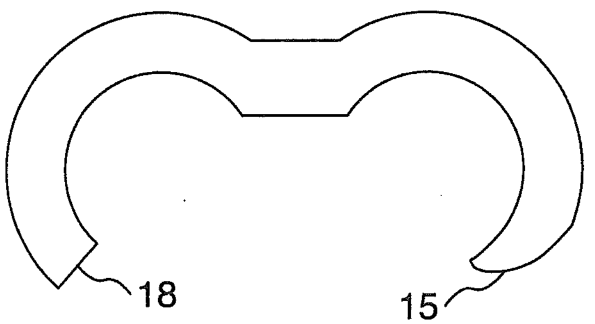


FIG.5

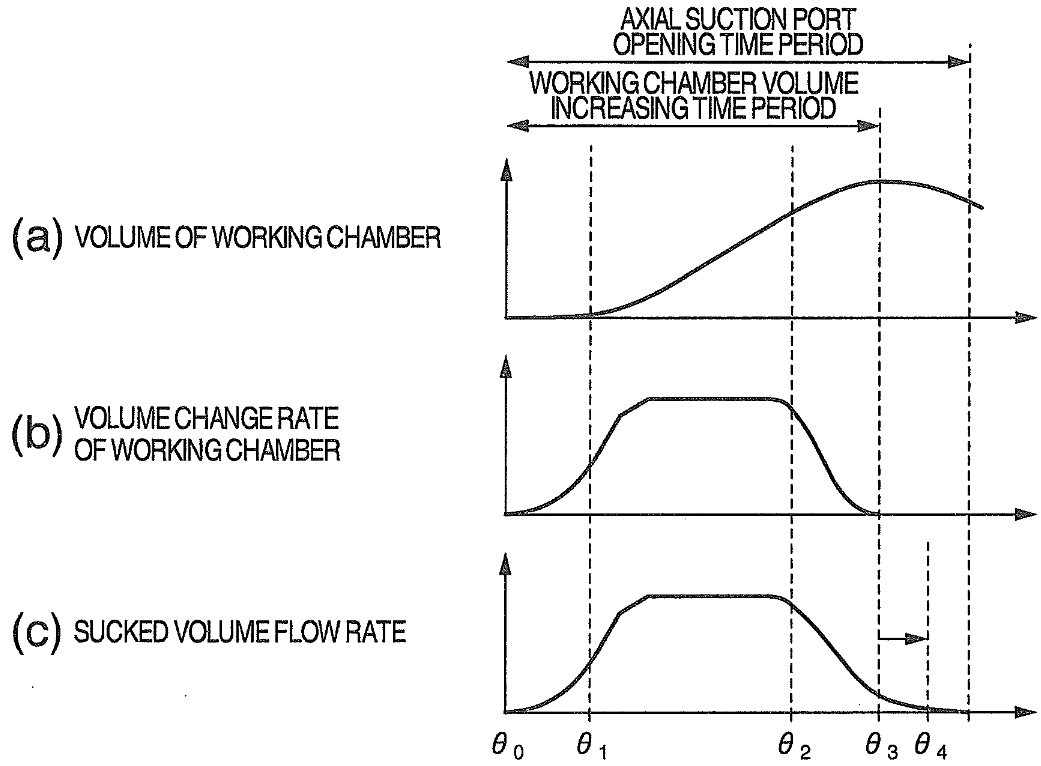
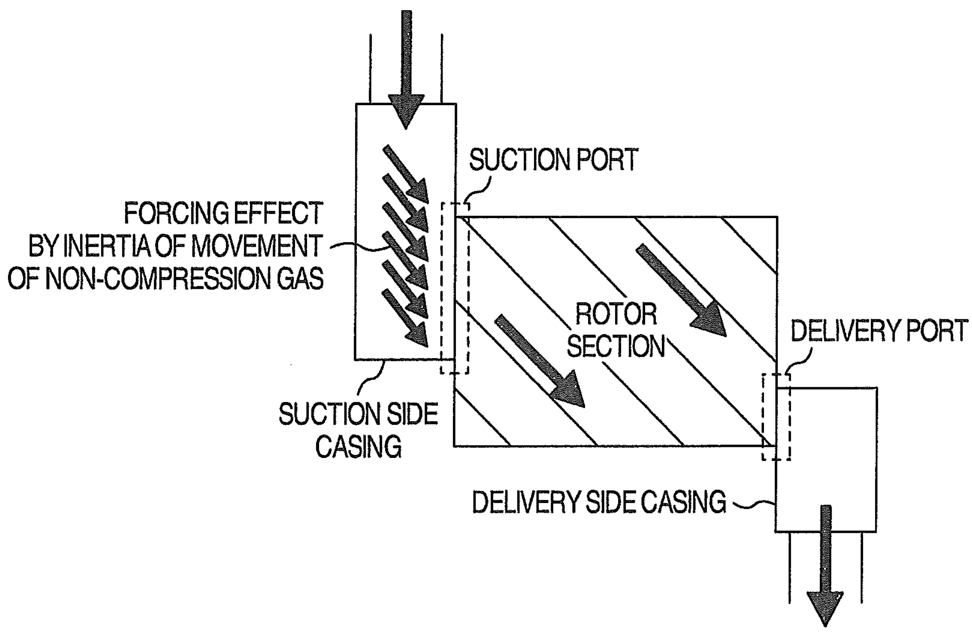

AXIAL SUCTION PORT OPENING TIME PERIOD
WORKING CHAMBER VOLUME INCREASING TIME PERIOD
(a) VOLUME OF WORKING CHAMBER
(b) VOLUME CHANGE RATE OF WORKING CHAMBER
(c) SUCKED VOLUME FLOW RATE
θ_0
θ_1
θ_2
θ_3
θ_4

FIG.6


SUCTION PORT
FORCING EFFECT BY INERTIA OF MOVEMENT OF NON-COMPRESSION GAS
ROTOR SECTION
DELIVERY PORT
SUCTION SIDE CASING
DELIVERY SIDE CASING

REFERENCES CITED IN THE DESCRIPTION

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- US 2457314 A **[0005]**
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