

DESCRIPTION

LIQUID-COOLED SCREW COMPRESSOR

TECHNICAL FIELD

[0001] The present invention relates to a liquid-cooled screw compressor.

BACKGROUND ART

[0002] In a liquid-cooled screw compressor such as an oil-cooled screw compressor, a liquid (for example, oil) is supplied to a rotor chamber for lubrication and cooling of compressed air, and the liquid is mixed into the compressed air that is in a process of compression created by male and female rotors rotating while meshing with each other. An oil supply port to the rotor chamber provided in the oil-cooled screw compressor disclosed in Patent Document 1 is a so-called drill hole, and has a straight-pipe shape having a constant diameter between both ends.

PRIOR ART DOCUMENT

PATENT DOCUMENT

[0003] Patent Document 1: JP 2014-214740 A

SUMMARY OF THE INVENTION

PROBLEMS TO BE SOLVED BY THE INVENTION

[0004] In the liquid-cooled screw compressor, there occurs a loss in power other than power for compressing air, such as a power loss due to

agitation of the liquid and a power loss due to liquid viscosity in a narrow gap between the rotor chamber and a tooth portion of the screw rotor. Due to these power losses, the liquid cooling type may rather have reduced efficiency.

[0005] Further, in a case where the liquid supply port has a straight-pipe shape like the oil supply port of Patent Document 1, when the screw rotor passes immediately above the liquid supply port in the operation, the flow of the liquid supplied from the liquid supply port to the rotor chamber is obstructed and the velocity is decreased. Due to such a decrease in velocity, the dispersible performance of the liquid in the tooth groove may decrease and the cooling efficiency for the compressed air may decrease.

[0006] For the above reasons, typical liquid-cooled screw compressors have room for improvement in reducing the power necessary for compressing the air to a demanded pressure and improving the cooling efficiency.

[0007] The present invention has an object to reduce the power necessary for compressing the air to a demanded pressure and to improve the cooling efficiency in a liquid-cooled screw compressor.

MEANS FOR SOLVING THE PROBLEMS

[0008] According to an aspect of the present invention, there is provided a liquid-cooled screw compressor including: a rotor chamber provided in a casing and accommodating a pair of screw rotors; and a liquid supply port provided in the casing to supply a liquid supplied from a liquid

supply line to the rotor chamber, wherein the liquid supply port includes an inlet portion that fluidly communicates with the liquid supply line, an injection portion that fluidly communicates with the rotor chamber, and an intermediate portion that fluidly connects the inlet portion and the injection portion and has a fluid passage cross-sectional area that is constant, and wherein a fluid passage cross-sectional area at an opening portion of the injection portion facing the rotor chamber is larger than the fluid passage cross-sectional area of the intermediate portion.

[0009] The opening portion of the injection portion of the liquid supply port facing the rotor chamber has the fluid passage cross-sectional area larger than the fluid passage cross-sectional area of the intermediate portion of the liquid supply port. With such a configuration, as compared with the case where the oil supply port has a straight-pipe shape (so-called drill hole) in which the fluid passage cross-sectional area is constant between both ends, at a moment when a tooth portion of the female rotor passes immediately above the oil supply port in the operation, a distance between the end portion of the intermediate portion on the rotor chamber side and a tooth tip of the tooth portion becomes longer. In other words, when the oil that has been injected from the oil supply port passes through a gap between the tooth portion (tooth tip) of the screw rotor and the casing, a cross-sectional area through which the oil can pass is enlarged. Therefore, a pressure loss at the time of oil injection into the rotor chamber is reduced. Due to such a reduction in the pressure loss, a flow velocity of the liquid injected into the rotor chamber can be made higher while a volumetric flow rate of the oil supplied to the

liquid supply port remains identical to that of the case where the liquid supply port is the drill hole. In other words, without increasing the volumetric flow rate, the flow velocity of the liquid injected into the rotor chamber can be made higher than that of the case where the liquid supply port is the drill hole. Such a higher flow velocity promotes atomization occurring when a liquid column collides with a tooth surface of the tooth portion of the screw rotor, increases heat transfer areas of liquid droplets, and improves the cooling efficiency for the compressed air. Therefore, the power necessary for compressing the air to a demanded pressure can be reduced.

[0010] Supposing that the volumetric flow rate of the liquid supplied to the oil supply port is increased due to an increase in the flow velocity of the liquid injected into the rotor chamber, this will increase a power loss due to the agitation of the liquid by the tooth portion of the screw rotor and a power loss due to the viscosity of the liquid in a narrow gap between the tooth portion (tooth tip) of the screw rotor and the casing. However, as described above, according to an aspect of the present invention, the flow velocity of the liquid injected into the rotor chamber can be made higher while the volumetric flow rate of the liquid supplied to the liquid supply port remains identical. Therefore, no increase in the power loss occurs.

[0011] Preferably, a diameter at the opening portion of the injection portion is larger than an axially-perpendicular tooth tip width of a tooth portion of one of the pair of screw rotors.

[0012] In the present description, the “axially-perpendicular tooth tip

width” of the tooth portion of the screw rotor means a width of a smooth surface at a tip of the tooth portion in a cross section where the tooth portion is orthogonal to a rotor shaft. By setting the diameter at the opening portion to be larger than the axially-perpendicular tooth tip width, the smooth surface does not close the injection portion of the liquid supply port when the tooth portion of the screw rotor passes immediately above the liquid supply port. Hence, the pressure loss at the time of liquid injection into the rotor chamber can be reduced in a more effective manner.

[0013] A diameter of the intermediate portion may be equal to or larger than 0.7 mm and equal to or smaller than 18 mm, and a diameter at the opening portion of the injection portion may be equal to or less than 4.0 times the diameter of the intermediate portion.

[0014] In particular, the diameter at the opening portion of the injection portion may be equal to or more than 1.5 times and equal to or less than 3.0 times the diameter of the intermediate portion.

[0015] The injection portion may have a reverse tapered shape in which the fluid passage cross-sectional area gradually increases from a portion connected with the intermediate portion toward the opening portion.

[0016] The injection portion may have the fluid passage cross-sectional area that is constant from a portion connected with the intermediate portion to the opening portion, and a step, in which the fluid passage cross-sectional area increases discontinuously, may be formed in a portion of the injection portion connected with the intermediate portion.

[0017] The liquid supply port may include a pipe member, both ends

of which are opened and which is inserted into a mounting hole that penetrates through from the liquid supply line provided in the casing to the rotor chamber, the intermediate portion may be defined by the pipe member, an end face of the pipe member closer to the rotor chamber is located in the mounting hole, and the injection portion may be defined by the end face of the pipe member and a hole peripheral wall of the mounting hole.

[0018] With such a configuration, dimension control in processing for providing the liquid supply port in the casing becomes easier than a case of processing the casing itself by drilling or the like to form the injection portion having the fluid passage cross-sectional area larger than the fluid passage cross-sectional area of the intermediate portion at the opening portion facing the rotor chamber. By making the dimension control easier during the processing, a higher level of quality stability can be ensured. Further, by using a different pipe member, the diameter of the intermediate portion of the liquid supply port can be easily changed according to the product use. Further, as the processing performed on the casing to provide the liquid supply port, processing of a step in which the fluid passage cross-sectional area increases discontinuously or processing a tapered shape in which the fluid passage cross-sectional area continuously increases is no longer necessary. Since only the formation of the mounting hole (through-hole) is demanded, the man-hours can be reduced.

[0019] The inlet portion may have a tapered shape in which the fluid passage cross-sectional area gradually decreases from a portion

connected with the liquid supply line toward the intermediate portion.

[0020] The inlet portion may have a fluid passage cross-sectional area that is constant and larger than the fluid passage cross-sectional area of the intermediate portion, from a portion connected with the liquid supply line to a portion connected with the intermediate portion, and a step may be formed in a portion of the inlet portion connected with the intermediate portion, the step having the fluid passage cross-sectional area that decreases discontinuously.

[0021] With these configurations, a sharp reduction is alleviated in the fluid passage cross-sectional area when the liquid flows from the liquid supply line into the inlet portion of the liquid supply port, and the pressure loss is reduced. As a result, the flow velocity of the liquid injected into the rotor chamber can be increased in a more effective manner without increasing the volumetric flow rate of the liquid supplied to the liquid supply port.

EFFECT OF THE INVENTION

[0022] According to a liquid-cooled screw compressor according to the present invention, the cooling efficiency for the compressed air can be improved, and the power necessary for compressing the air to a demanded pressure can be reduced, that is, an improvement in efficiency can be achieved.

BRIEF DESCRIPTION OF THE DRAWINGS

[0023]

Fig. 1 is a schematic plan view of an oil-cooled screw compressor according to a first embodiment of the present invention;

Fig. 2 is a cross-sectional view taken along line II-II of Fig. 1;

Fig. 3 is a cross-sectional view taken along line III-III of Fig. 1;

Fig. 4 is a schematic view of a compressor system including the oil-cooled screw compressor according to the first embodiment of the present invention;

Fig. 5 is an enlarged view of a part V in Fig. 3;

Fig. 6 is a cross-sectional view of the part V of Fig. 3 in a different cross section;

Fig. 7 is a view of an oil supply port when viewed from a female rotor chamber;

Fig. 8 is a cross-sectional view similar to Fig. 5 of a typical oil-cooled compressor;

Fig. 9 is a cross-sectional view similar to Fig. 6 according to a modification of the first embodiment;

Fig. 10 is a cross-sectional view similar to Fig. 6 according to another modification of the first embodiment;

Fig. 11 is a cross-sectional view, similar to Fig. 3, of an oil-cooled screw compressor according to a second embodiment of the present invention;

Fig. 12 is a cross-sectional view similar to Fig. 6 according to the second embodiment;

Fig. 13 is a cross-sectional view similar to Fig. 6 according to a modification of the second embodiment;

Fig. 14 is a cross-sectional view similar to Fig. 6 according to another modification of the second embodiment;

Fig. 15 is a cross-sectional view, similar to Fig. 3, of an oil-cooled screw compressor according to a third embodiment of the present invention;

Fig. 16 is a cross-sectional view similar to Fig. 6 according to the third embodiment;

Fig. 17 is a perspective view of an orifice pipe; and

Fig. 18 is a graph showing relationships between an oil amount and a cross-sectional efficiency.

DESCRIPTION OF EMBODIMENTS

[0024] (First Embodiment)

Referring to Figs. 1 to 3, an oil-cooled screw compressor (liquid-cooled screw compressor) 1 according to a first embodiment of the present invention includes a casing 2 in which a male rotor chamber 2a and a female rotor chamber 2b spatially communicating with each other are formed. A male rotor 3 is accommodated in the male rotor chamber 2a, and a female rotor 4 is accommodated in the female rotor chamber 2b. Further, the casing 2 is provided with a suction port 2c and a discharge port 2d which spatially communicate with the rotor chambers 2a and 2b.

[0025] The male rotor chamber 2a is defined by a cylindrical surface 2e and a pair of end faces 2g and 2h. In addition, the female rotor chamber 2b is defined by a cylindrical surface 2f and by the pair of end

faces 2g and 2h which are common to the male rotor chamber 2a.

[0026] The male rotor (screw rotor) 3 includes a rotor shaft 3a and a plurality of spiral tooth portions 3b provided on the outer periphery of the rotor shaft 3a. Similarly, the female rotor 4 includes a rotor shaft 4a and a plurality of spiral tooth portions 4b provided on the outer periphery of the rotor shaft 4a. Spiral tooth grooves 4c are respectively defined between pairs of the tooth portions 4b adjacent to each other. The rotor shaft 3a of the male rotor 3 is supported by bearings 5A and 5B while the male rotor 3 is rotatable around its own axis Lm. The rotor shaft 4a of the female rotor 4 is also instructed, by bearings 6A and 6B, to be rotatable around its own axis Lf.

[0027] A drive mechanism 7 including a motor is mechanically connected with the rotor shaft 3a on the suction port 2c side of the male rotor 3. When the male rotor 3 is rotated by the drive mechanism 7, the tooth portion 3b of the male rotor 3 meshes with the tooth groove 4c of the female rotor 4 while being inserted. Accordingly, the male rotor 3 and the female rotor 4 rotate synchronously. Instead of the male rotor 3, the female rotor 4 may be driven to rotate by a drive mechanism.

[0028] Gas (air in the present embodiment) that has been sucked from the suction port 2c is confined in a confined space defined by the tooth portion 3b of the male rotor 3 and the tooth groove 4c of the female rotor 4, is compressed while moving in the directions of the axes Lm and Lf in accordance with the rotation of the rotors 3 and 4, and is discharged from the discharge port 2d.

[0029] The casing 2 is provided with three oil supply ports (liquid supply

ports) 11A, 11B, and 11C for supplying oil (liquid) for cooling, lubrication, and the like to the female rotor chamber 2b. These oil supply ports 11A to 11C are opened at a bottom portion of the female rotor chamber 2b, and are arranged on a straight line along the axis Lf of the female rotor 4 (which also serves as the axis of the female rotor chamber 2b). The number of the oil supply ports may be one or two, or four or more. Further, an oil supply port for the male rotor chamber 2a may be provided instead of or in combination with the oil supply port for the female rotor chamber 2b. The oil supply ports 11A to 11C will be described in detail later.

[0030] Since the oil is supplied to the female rotor chamber 2b from the oil supply ports 11A to 11C, air discharged from the discharge port 2d contains the oil. With also reference to Fig. 4, the air discharged from the discharge port 2d is introduced into a separator 13 via air piping 12A. In the separator 13, the air and the oil are separated. The air from which the oil has been separated is fed from air piping 12B to a device or equipment that needs the compressed air. The oil that has been separated from the air by the separator 13 is fed to an oil supply line (liquid supply line) 15 provided in the casing 2 (an elongated hole drilled in the casing 2 in the present embodiment) via oil supply piping 14. The oil is supplied from the oil supply line 15 to the female rotor chamber 2b via the oil supply ports 11A to 11C. In this manner, the oil circulates between the oil-cooled screw compressor 1 and the separator 13. In the present embodiment, an oil pump 16 for supplying the oil from the separator 13 to the oil-cooled screw compressor 1 is provided in the oil

supply piping 14.

[0031] Next, the oil supply ports 11A to 11C will be described in detail. In the following description, reference numeral 11 is used for one oil supply port, unless it is necessary to distinguish among the three oil supply ports 11A to 11C in particular.

[0032] Referring to Figs. 5 to 7, the oil supply line 15 and the female rotor chamber 2b fluidly communicate with each other via the oil supply port 11. The oil supply port 11, as a whole, extends straight in a direction orthogonal to the axis L_m of the female rotor 4 (that is also the axis of the female rotor chamber 2b).

[0033] The oil supply port 11 includes an inlet portion 21 that fluidly communicates with the oil supply line 15, an injection portion 22 that fluidly communicates with the female rotor chamber 2b, and an intermediate portion 23 that fluidly connects the inlet portion 21 and the injection portion 22.

[0034] In the present embodiment, the cross-sectional shape of the inlet portion 21 and the intermediate portion 23 orthogonal to an axis L_i of the oil supply port 11 is circular. This cross-sectional shape may be other than circular. Further, the inlet portion 21 has a constant diameter D_e and the intermediate portion 23 also has a constant diameter D_m . These diameters D_e and D_m are identical to each other. In other words, fluid passage cross-sectional areas A_e and A_m are constant over the entire area from the inlet portion 21 to the intermediate portion 23.

[0035] The injection portion 22 includes an opening portion 22a that opens on the female rotor chamber 2b, more specifically, on the

cylindrical surface 2f of the casing 2 that defines the female rotor chamber 2b. In the present embodiment, the cross-sectional shape of the injection portion 22 orthogonal to the axis L_i of the oil supply port 11 is circular. This cross-sectional shape may be other than circular. In the present embodiment, the injection portion 22 has a diameter D_i that gradually increases from a portion 22b connected with the intermediate portion 23 toward the opening portion 22a. In other words, the injection portion 22 has a reverse tapered shape in which a fluid passage cross-sectional area A_i gradually increases from the portion 22b where the injection portion 22 is connected with the intermediate portion 23 toward the opening portion 22a. Due to such a reverse tapered shape, the fluid passage cross-sectional area A_i at the opening portion 22a of the injection portion 22 is larger than the fluid passage cross-sectional area A_m of the intermediate portion 23.

[0036] The diameter D_i at the opening portion 22a of the injection portion 22 is larger than an axially-perpendicular tooth tip width W_t of the tooth portion 4b of the female rotor 4. The “axially-perpendicular tooth tip width” means a width of a smooth surface 4d at a tip of the tooth portion 4b in a cross section where the tooth portion 4b is orthogonal to the axis L_f of the rotor shaft 4a.

[0037] The diameter D_m of the intermediate portion 23 can be set to equal to or larger than 0.7 mm and equal to or smaller than 18 mm. The diameter D_i at the opening portion 22a of the injection portion 22 can be set to be equal to or less than 4 times the diameter D_m of the intermediate portion 23. In particular, the diameter D_i at the opening portion 22a of

the injection portion 22 can be set to equal to or larger than 1.5 times and equal to and less than 3.0 times the diameter D_m of the intermediate portion 23.

[0038] The oil supplied from the oil supply line 15 enters the oil supply port 11 at the inlet portion 21, flows into the intermediate portion 23, and is injected from an end portion 23a on the female rotor chamber 2b side of the intermediate portion 23. The injected oil is supplied to the female rotor chamber 2b through the injection portion 22.

[0039] The diameter D_i at the opening portion 22a of the injection portion 22 of the oil supply port 11 facing the female rotor chamber 2b, that is the fluid passage cross-sectional area A_i is larger than the diameter D_m of the intermediate portion 23 of the oil supply port 11, that is the fluid passage cross-sectional area A_m . Therefore, as compared with the case where the oil supply port 11 has a straight-pipe shape (so-called drill hole) in which the fluid passage cross-sectional area is constant between both ends, as shown in Fig. 8, at a moment when the tooth portion 4b of the female rotor 4 passes immediately above the oil supply port 11 in the operation, a distance between the end portion 23a of the intermediate portion 23 on the female rotor chamber 2b side and the tooth tip of the tooth portion 4b becomes longer. In other words, when the oil that has been injected from the oil supply port 11 passes through a gap between the tooth tip of the tooth portion 4b of the female rotor 4 and the cylindrical surface 2f of the casing 2, a cross-sectional area through which the oil can pass is enlarged. Therefore, a pressure loss at the time of oil injection into the female rotor chamber 2b is reduced. Due

to such a reduction in the pressure loss, a flow velocity of the oil injected into the female rotor chamber 2b can be made higher while a volumetric flow rate of the oil supplied to the oil supply port 11 remains identical to that of the case where the oil supply port 11 is the drill hole as shown in Fig. 8. In other words, without increasing the volumetric flow rate, the flow velocity of the oil injected into the female rotor chamber 2b can be made higher than that of the case where the oil supply port 11 is the drill hole. Such a higher flow velocity promotes atomization occurring when a liquid column collides with a tooth surface of the tooth portion 4b of the female rotor 4, increases heat transfer areas of oil droplets, and improves the cooling efficiency for the compressed air. Therefore, the power necessary for compressing the air to a demanded pressure can be reduced.

[0040] Supposing that the volumetric flow rate of the liquid supplied to the oil supply port 11 is increased due to a higher flow velocity of the oil injected into the female rotor chamber 2b, this will increase a power loss due to the agitation of the oil by the tooth portion 4b of the female rotor 4 and a power loss due to the viscosity of the oil in a narrow gap between the tooth portion 4b (tooth tip) of the female rotor 4 and the cylindrical surface 2f of the casing 2. However, in the present embodiment, the flow velocity of the oil injected into the female rotor chamber 2b can be increased while the volumetric flow rate of the oil supplied to the oil supply port 11 remains identical. Therefore, no increase in the power loss occurs.

[0041] As described above, the diameter D_i at the opening portion 22a

of the injection portion 22 is larger than the axially-perpendicular tooth tip width W_t of the tooth portion 4b of the female rotor 4. Therefore, when the tooth portion 4b of the female rotor 4 passes immediately above the oil supply port 11, the smooth surface 4d of the tip of the tooth portion 4b does not close the injection portion 22 of the oil supply port 11, and the pressure loss at the time of oil injection into the female rotor chamber 2b can be reduced in a more effective manner.

[0042] Figs. 9 and 10 show modifications of the first embodiment.

[0043] In a modification of Fig. 9, the inlet portion 21 of the oil supply port 11 has a tapered shape in which the diameter D_e from a portion 21a connected with the oil supply line 15 to a portion 21b connected with the intermediate portion 23, that is the fluid passage cross-sectional area A_e gradually decreases.

[0044] In a modification of Fig. 10, the inlet portion 21 of the oil supply port 11 has the diameter D_e , which is constant and larger than the diameter D_m of the intermediate portion 23, from the portion 21a connected with the oil supply line 15 to the portion 21b connected with the intermediate portion 23. In other words, the inlet portion 21 has the fluid passage cross-sectional area A_e , which is constant and larger than the fluid passage cross-sectional area A_m of the intermediate portion 23, from the portion 21a connected with the oil supply line 15 to the portion 21b connected with the intermediate portion 23. Therefore, a step 25, in which the fluid passage cross-sectional area decreases sharply or discontinuously is formed, in a portion where the inlet portion 21 is connected with the intermediate portion 23.

[0045] According to the configurations shown in Figs. 9 and 10, a sharp reduction is alleviated in the fluid passage cross-sectional area when the oil flows from the oil supply line 15 into the inlet portion 21 of the oil supply port 11, and the pressure loss is reduced. As a result, the flow velocity of the oil injected into the female rotor chamber 2b can be increased more effectively, without increasing the volumetric flow rate of the oil supplied to the oil supply port 11.

[0046] In second and third embodiments below, differences from the first embodiment will be described. The structures, functions, and the like for which any of these embodiments are not particularly mentioned are the same as those of the first embodiment. Further, in the drawings relating to these embodiments, the same or similar elements as those of the first embodiment are designated by the same reference numerals. Further, the overall configuration of the oil-cooled compressor according to the first embodiment shown in Figs. 1, 2 and 4 is the same also in these embodiments.

[0047] (Second Embodiment)

In an oil-cooled screw compressor 1 according to a second embodiment of the present invention shown in Figs. 11 and 12, the injection portion 22 of the oil supply port 11 has a diameter D_i , which is constant and larger than a diameter D_m of the intermediate portion 23, from the portion 22b connected with the intermediate portion 23 to the opening portion 22a. In other words, the injection portion 22 has a fluid passage cross-sectional area A_i , which is constant and larger than the fluid passage cross-sectional area A_m of the intermediate portion 23 from

the portion 22b connected with the intermediate portion 23 to the opening portion 22a. Therefore, a step 26, in which the fluid passage cross-sectional area increases sharply or discontinuously, is formed in a portion where the injection portion 22 is connected with the intermediate portion 23.

[0048] The opening portion 22a of the injection portion 22 facing the female rotor chamber 2b has the fluid passage cross-sectional area A_i larger than the fluid passage cross-sectional area A_m of the intermediate portion 23. Therefore, the flow velocity of oil injected into the female rotor chamber 2b can be increased without increasing the volumetric flow rate. Such an increase in velocity improves the cooling efficiency for the compressed air, and reduces the power necessary for compressing the air to a demanded pressure. Further, the flow velocity of the liquid injected into the female rotor chamber can be increased while the volumetric flow rate of the oil supplied to the oil supply port 11 remains identical. Therefore, no increase in the power loss occurs. Further, the diameter D_i at the opening portion 22a of the injection portion 22 is larger than the axially-perpendicular tooth tip width W_t of the tooth portion 4b of the female rotor 4. Therefore, the smooth surface 4d of the tip surface of the tooth portion 4b does not close the injection portion 22, and the pressure loss at the time of oil injection into the female rotor chamber 2b can be reduced more effectively.

[0049] Figs. 13 and 14 show modifications of the second embodiment.

[0050] In a modification of Fig. 13, the inlet portion 21 of the oil supply port 11 has the fluid passage cross-sectional area A_e , which is constant

and larger than the fluid passage cross-sectional area A_m of the intermediate portion 23, from the portion 21a connected with the oil supply line 15 to the portion 21b connected with the intermediate portion 23. Therefore, the step 25, in which the fluid passage cross-sectional area decreases discontinuously, is formed in a portion where the inlet portion 21 is connected with the intermediate portion 23. A thickness TH_m of the intermediate portion 23 is larger than the sum of a thickness TH_e of the inlet portion 21 and a thickness TH_i of the injection portion 22.

[0051] In a modification of Fig. 14, the inlet portion 21 of the oil supply port 11 has a tapered shape in which the diameter D_e gradually decreases from the portion 21a connected with the oil supply line 15 to the portion 21b connected with the intermediate portion 23, that is, the fluid passage cross-sectional area A_e gradually decreases. According to the configurations of Figs. 13 and 14, a sharp reduction is alleviated in the fluid passage cross-sectional area when the oil flows from the oil supply line 15 into the inlet portion 21 of the oil supply port 11, and the pressure loss is reduced. As a result, the flow velocity of the oil injected into the female rotor chamber 2b can be increased more effectively, without increasing the volumetric flow rate of the oil supplied to the oil supply port 11.

[0052] (Third Embodiment)

In an oil-cooled screw compressor 1 according to a third embodiment of the present invention shown in Figs. 15 and 16, the oil supply port 11 is provided by attaching a separate member to the casing

2.

[0053] The casing 2 is provided with a mounting hole 31 penetrating through from the oil supply line 15 to the female rotor chamber 2b. In the present embodiment, the mounting hole 31 is circular and has a constant diameter D_a . An orifice pipe (pipe member) 32 having openings at both ends, that is, an axial bore 32a in the center as shown in Fig. 17 is inserted or fitted into the mounting hole 31, and is fixed to the casing 2.

[0054] An end face 32b of the orifice pipe 32 on the female rotor chamber 2b side is located in the mounting hole 31, and is recessed with respect to the cylindrical surface 2f defining the female rotor chamber 2b. Further, an end surface 32c of the orifice pipe 32 on the oil supply line 15 side is also located in the mounting hole 31. The inlet portion 21 of the oil supply port 11 is defined by the end surface 32c of the orifice pipe 32 and a hole peripheral wall 31a of the mounting hole 31. Further, the axial bore 32a of the orifice pipe 32 constitutes the intermediate portion 23 of the oil supply port 11. Further, the injection portion 22 of the oil supply port 11 is defined by the end face 32b of the orifice pipe 32 and the hole peripheral wall 31a of the mounting hole 31.

[0055] The injection portion 22 in the present embodiment has the same shape as that of the second embodiment. In particular, the step 26, in which the fluid passage cross-sectional area increases discontinuously, is formed in the intermediate portion 23, which is a portion where the axial bore 32a of the orifice pipe 32 and the injection portion 22 are connected with each other.

[0056] The inlet portion 21 in the present embodiment has the same shape as that shown in Fig. 13. In particular, the step 25, in which the fluid passage cross-sectional area decreases discontinuously, is formed in the intermediate portion 23, which is a portion where the inlet portion 21 is connected with the axial bore 32a of the orifice pipe 32.

[0057] In addition to the same effects as those of the first and second embodiments and their modifications, the present embodiment has the following effects. First, the dimension control at the time of processing for providing the oil supply port 11 in the casing 2 becomes easier, as compared to a case of processing the casing 2 itself by drilling or the like to form the intermediate portion 23 and the injection portion 22 having a fluid passage cross-sectional area larger than the fluid passage cross-sectional area of the intermediate portion 23 at an opening portion facing the rotor chamber. By making the dimension control easier during the processing, a higher level of quality stability can be ensured. Further, by using a different orifice pipe 32, the diameter D_m of the intermediate portion 23 of the oil supply port 11 can be easily changed according to the product use. Further, as the processing to be performed on the casing 2 to provide the oil supply port 11, processing of a step in which the fluid passage cross-sectional area increases discontinuously or processing a tapered shape in which the fluid passage cross-sectional area continuously increases is no longer necessary. Since only the formation of the mounting hole 31 is demanded, the man-hours can be reduced.

[0058] (Test)

A test was conducted to confirm the effects of the present invention. In this test, the oil-cooled screw compressor 1 in the third embodiment (the orifice pipe 32 constitutes the oil supply port 11 as shown in Fig. 15) and the oil-cooled screw compressor 1 shown in Fig. 8 (the oil supply port 11 is a drill hole with a constant diameter) as a comparative example were used. Each of the oil-cooled screw compressors 1 (75KW) was actually operated with two different lubrication oil amounts, and the heat insulation efficiency (the ratio of the theoretical consumption power with respect to the actual consumption power) was calculated. The discharge pressure of the oil-cooled screw compressors 1 were each set to 0.7 MPa. The ambient temperature was 20°C.

[0059] In both the third embodiment and the comparative example, the diameters D_m of the intermediate portions 23 of the three oil supply ports 11A to 11C were 4.4 times the axially-perpendicular tooth tip width W_t of the tooth portion 4b of the female rotor 4 ($D_m = 4.4 \times W_t$).

[0060] Regarding the third embodiment, the heat insulation efficiency was calculated when operated at a predetermined oil amount Q_0 (L/min) and about $1.4 \times Q_0$ (L/min) of the lubrication oil amounts. In the case where the lubrication oil amount was Q_0 (L/min), the diameters D_i of the injection portions 22 of the oil supply ports 11A, 11B, and 11C were respectively 2.2 D_m ($D_i = 2.2 \times D_m$), 2.9 D_m ($D_i = 2.9 \times D_m$), and 2.9 D_m ($D_i = 2.9 \times D_m$). On the other hand, when the amount of lubricating oil is $1.4 \times Q_0$ (L/min), the diameters D_i of the injection portion 22 of the oil supply ports 11A, 11B, and 11C were respectively 1.7 D_m ($D_i = 1.7 \times D_m$), 2.2 D_m ($D_i = 2.2 \times D_m$), and 2.2 D_m ($D_i = 2.2 \times D_m$).

[0061] Fig. 18 shows test results. As shown in Fig. 18, it was confirmed that the oil-cooled screw compressor 1 in the third embodiment had improved the heat insulation efficiency by about 1% as compared with that of the comparative example.

REFERENCE SIGNS LIST

[0062]

1: Oil-cooled screw compressor (liquid-cooled screw compressor), 2: Casing, 2a: Male rotor chamber, 2b: Female rotor chamber, 2c: Suction port, 2d: Discharge port, 2e; 2f: Cylindrical surface, 2g; 2h: End face, 3: Male rotor, 3a: Rotor shaft, 3b: Tooth portion, 4: Female rotor, 4a: Rotor shaft, 4b: Tooth portion, 4c: Tooth groove, 4d: Smooth surface, 5A; 5B; 6A; 6B: Bearing, 7: Drive mechanism, 11A; 11B; 11C: Oil supply port (liquid supply port), 12A; 12B: Air piping, 13: Separator, 14: Oil supply piping, 15: Oil supply line (liquid supply line), 16: Oil pump, 21: Inlet portion, 21a; 21b: Portion, 22: Injection portion, 22a: Opening portion, 22b: Portion, 23: Intermediate portion, 23a: End portion, 25; 26: Step, 31: mounting hole, 31a: Hole peripheral wall, 32: Orifice pipe, 32a: Axial bore, 32b, 32c: End face

CLAIMS

1. A liquid-cooled screw compressor comprising:
 - a rotor chamber provided in a casing and accommodating a pair of screw rotors; and
 - a liquid supply port provided in the casing to supply a liquid supplied from a liquid supply line to the rotor chamber, wherein the liquid supply port includes:
 - an inlet portion that fluidly communicates with the liquid supply line;
 - an injection portion that fluidly communicates with the rotor chamber; and
 - an intermediate portion that fluidly connects the inlet portion and the injection portion, the intermediate portion having a fluid passage cross-sectional area that is constant, and
 - wherein a fluid passage cross-sectional area at an opening portion of the injection portion facing the rotor chamber is larger than the fluid passage cross-sectional area of the intermediate portion.
2. The liquid-cooled screw compressor according to claim 1, wherein a diameter at the opening portion of the injection portion is larger than an axially-perpendicular tooth tip width of a tooth portion of one the pair of screw rotors.
3. The liquid-cooled screw compressor according to claim 1 or 2, wherein a diameter of the intermediate portion is equal to or larger

than 0.7 mm and equal to or smaller than 18 mm, and

wherein a diameter at the opening portion of the injection portion is equal to or less than 4.0 times the diameter of the intermediate portion.

4. The liquid-cooled screw compressor according to claim 3, wherein the diameter at the opening portion of the injection portion is equal to or more than 1.5 times and equal to or less than 3.0 times the diameter of the intermediate portion.

5. The liquid-cooled screw compressor according to claim 1 or 2, wherein the injection portion has a reverse tapered shape in which the fluid passage cross-sectional area gradually increases from a portion connected with the intermediate portion toward the opening portion.

6. The liquid-cooled screw compressor according to claim 1 or 2, wherein the injection portion has the fluid passage cross-sectional area that is constant from a portion connected with the intermediate portion to the opening portion, and

wherein a step is formed in a portion of the injection portion connected with the intermediate portion, the step having the fluid passage cross-sectional area that increases discontinuously.

7. The liquid-cooled screw compressor according to claim 6, wherein the liquid supply port includes a pipe member, both ends of which are opened and which is inserted into a mounting hole that

penetrates through from the liquid supply line provided in the casing to the rotor chamber,

wherein the intermediate portion is defined by the pipe member,

wherein an end face of the pipe member closer to the rotor chamber is located in the mounting hole, and

wherein the injection portion is defined by the end face of the pipe member and a hole peripheral wall of the mounting hole.

8. The liquid-cooled screw compressor according to claim 1 or 2, wherein the inlet portion has a tapered shape in which the fluid passage cross-sectional area gradually decreases from a portion connected with the liquid supply line toward the intermediate portion.

9. The liquid-cooled screw compressor according to claim 1 or 2, wherein the inlet portion has a fluid passage cross-sectional area that is constant and larger than the fluid passage cross-sectional area of the intermediate portion, from a portion connected with the liquid supply line to a portion connected with the intermediate portion, and

wherein a step is formed in a portion of the inlet portion connected with the intermediate portion, the step having the fluid passage cross-sectional area that decreases discontinuously.