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(54) **REFRIGERATING SYSTEM AND CONDENSER FOR DECOMPRESSION TUBE SYSTEM**

(57) This refrigeration system is an orifice-tube system constituting a refrigeration cycle in which a refrigerant passes through a compressor 1, a condenser 10, an orifice-tube 3, an evaporator 4, and an accumulator 5 in this order and then returns to the compressor 1. The condenser 10 is constituted by the so-called multi-flow type heat exchanger having a plurality of passes P1-P3. The intermediate pass P2 is constituted as a decompression pass for decompressing the refrigerant. After condensing the refrigerant by the first pass P1, the condensed refrigerant is decompressed and evaporated by the decompression pass P2, and then the evaporated refrigerant is re-condensed by the third pass P3. This refrigeration system is excellent in response characteristic to thermal load fluctuations and in refrigeration performance.

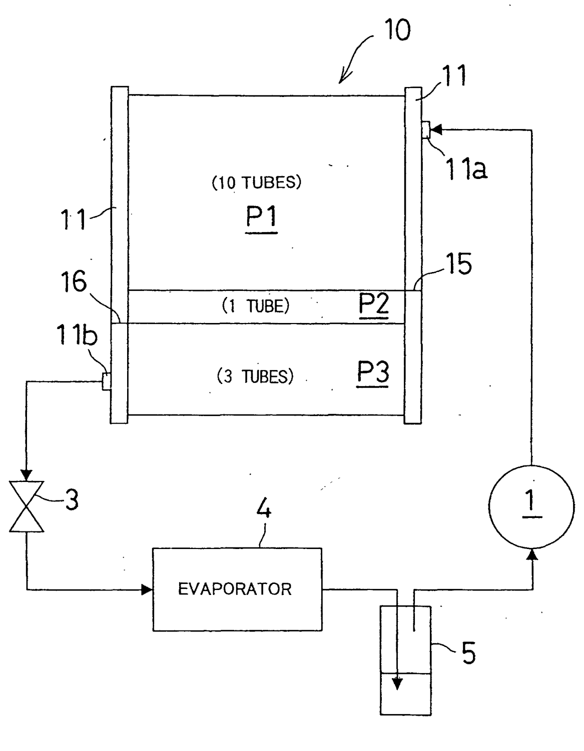

FIG.1

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Description

[0001] This application claims priority to Japanese Patent Application No. 2001-278975 filed on September 14, 2001 and U.S. Provisional Application No. 60/324,542 filed on September 26, 2001, the disclosure of which is incorporated by reference in its entirety.

Technical Field

[0002] The present invention relates to a refrigeration system for car air-conditioners including a refrigeration cycle which employs a decompressing tube as decompressing means such as an orifice-tube or a capillary tube, and also relates to a condenser for use in a decompressing-tube system.

Background Art

[0003] Generally, as a refrigeration system to be adopted for car air-conditioners or the like, the following refrigeration systems are well known: that is, an expansion-valve type refrigeration system including an automatic thermal expansion valve (TXV) as a decompressing means (hereinafter referred to as "expansion valve system"); and an orifice-tube type refrigeration system (CCOT) including a decompressing tube as decompressing means such as an orifice-tube or a capillary tube (hereinafter referred to as "orifice-tube system" or "decompressing-tube system").

[0004] As shown in Fig. 13, in the orifice-tube system, the gaseous refrigerant of high temperature and high pressure from the compressor 1 flows into the condenser 2 to be condensed therein. Then, the condensed refrigerant passes through the orifice-tube 3 to be decompressed and then flows into the evaporator 4. In the evaporator 4, the condensed refrigerant exchanges heat with the ambient air to be evaporated, and then is introduced into the accumulator 5. In the accumulator 5, only the gaseous refrigerant is separated from the refrigerant introduced in the accumulator 5, and the gaseous refrigerant returns to the aforementioned compressor 1. Thus, a refrigeration cycle is formed.

[0005] As compared with the expansion valve system, this orifice-tube system has fewer components and can be fabricated by fewer steps. Furthermore, the orifice-tube system is simple in structure and low in manufacturing cost.

[0006] The orifice-tube system is, however, inferior in response to load fluctuations.

[0007] That is, in the orifice-tube system, a liquefied refrigerant R stagnates at the subcooling area ranging from near the inlet of the orifice-tube 3 to the outlet of the condenser 2. This liquefied refrigerant R increases when the thermal load of the condenser 2 is small. For example, when an automobile mounting this refrigeration system is running at a high speed, the thermal load of the condenser 2 is small because of an enough amount of ventilation. In this case, the condenser performance can be fully demonstrated, resulting in an enhanced condensation of the refrigerant therein.

[0008] By the way, the amount of refrigerant passing through the orifice-tube 3 (i.e., circulation amount of refrigerant) is constant, and the amount of refrigerant passing through the orifice-tube 3 is limited. Accordingly, in cases where the thermal load of the condenser decreases suddenly, for example, when the running speed of the car is changed from a low-speed to a high-speed, the amount of liquefied refrigerant increases suddenly, and the subcooling area spreads even in the condenser 2. As a result, a large amount of liquefied refrigerant is temporarily accumulated in the condenser 2. When a large amount of liquefied refrigerant is accumulated in the condenser 2, the condensation of refrigerant will not be performed in the liquefied refrigerant stagnated portion. Accordingly, the effective area for condensing the refrigerant decreases, which in turn decreases the condenser performance.

[0009] To the contrary, in cases where the thermal load of the condenser increases suddenly, for example, when the running speed of the car is changed from a high-speed to a low-speed, the refrigerant is not condensed smoothly in the condenser 2. As a result, the amount of liquefied refrigerant accumulated in the outlet side portion in the condenser 2 decreases, resulting in insufficient subcooling degree of the liquefied refrigerant. This deteriorates the condenser performance temporarily. As will be apparent from the above, the orifice-tube system is inferior in response characteristic to load fluctuations, and cannot obtain sufficient refrigeration performance.

[0010] It is an object of the present invention to provide a refrigeration system that is excellent in response characteristic to load fluctuations and can obtain sufficient refrigeration performance irrespective of load fluctuations.

[0011] It is an object of the present invention to provide a condenser for use in a decompressing-tube system that is excellent in response characteristic to load fluctuations and can obtain sufficient refrigeration performance irrespective of load fluctuations.

[0012] Another object of the present invention will be apparent from the following embodiments.

Disclosure of the Invention

[0013] According to the first aspect of the present invention, a refrigeration system having a refrigeration cycle, comprises:

- a compressor for compressing a refrigerant;
- a condenser for condensing the refrigerant compressed by the compressor;
- a decompressing tube for decompressing the refrigerant condensed by the condenser;
- an evaporator for evaporating the refrigerant de-

compressed by the decompressing tube; and
an accumulator for separating a gaseous refrigerant from the refrigerant evaporated by the evaporator,

wherein the condenser includes a refrigerant inlet for introducing the refrigerant compressed by the compressor, a refrigerant outlet for discharging the refrigerant condensed by the condenser, a refrigerant passage for leading the refrigerant introduced from the refrigerant inlet to the refrigerant outlet while condensing the refrigerant, and decompressing means provided at a part of the refrigerant passage to decompress the refrigerant passing through the decompressing means.

[0014] In this refrigeration system, when the thermal load of the condenser decreases, the condensation of refrigerant in the condenser is enhanced at the upstream side of the decompressing means, and therefore only the completely liquefied refrigerant passes through the decompressing means. Thus, the resistance of the refrigerant passing through the decompressing means decreases, thereby increasing the flow rate. Accordingly, at the upstream side of the decompressing means and the downstream side thereof, the condensation of refrigerant is performed efficiently. Thus, the performance of the condenser is sufficiently demonstrated.

[0015] To the contrary, when the thermal load of the condenser increases, the condensation of refrigerant in the condenser deteriorates at the upstream side of the decompressing means, and therefore incompletely liquefied refrigerant passes through the decompressing means. At this time, the amount of gas in the refrigerant increases, i.e., the volume of the refrigerant passing through the decompressing means increases, resulting in increased flow resistance of the refrigerant passing through the decompressing means, which in turn decreases the flow rate. As the flow rate decreases in this way, the condensation load at the upstream side of the decompressing means decreases. Accordingly, the condensation will be performed fully, resulting in enhanced condenser performance.

[0016] As will be apparent from the above, since the refrigerant flow rate can be appropriately adjusted in response to fluctuations of thermal load in the condenser, this refrigeration system is excellent in response characteristics to load fluctuations. Accordingly, sufficient refrigeration performance can be obtained.

[0017] In this refrigeration system, an orifice-tube can be suitably used as the decompressing tube.

[0018] Furthermore, in this refrigeration system, it is preferable that at least a part of the condensed refrigerant is evaporated by the decompressing means and then re-condensed.

[0019] That is, in this refrigeration system, it is preferable that at least a part of the refrigerant condensed at an upstream side of the decompressing means in the refrigerant passage is decompressed by the decompressing means into a low-pressure gaseous refrigerant, and the low-pressure gaseous refrigerant is re-condensed at a downstream side of the decompressing means in the refrigerant passage.

[0020] According to the second aspect of the present invention, a refrigeration system having a refrigeration cycle in which a refrigerant is compressed into a compressed refrigerant, the compressed refrigerant is condensed into a condensed refrigerant, the condensed refrigerant is decompressed by giving passage resistance into a decompressed refrigerant, the decompressed refrigerant is evaporated into an evaporated refrigerant, and then a gaseous refrigerant is separated from the evaporated refrigerant and re-compressed, wherein a decompressing passage for decompressing the refrigerant is provided at a part of a refrigerant passage in which the compressed refrigerant is condensed.

[0021] In this refrigeration system, in the same manner as in the aforementioned system, since the flow rate of the refrigerant is appropriately adjusted by the decompressing passage in response to fluctuations of thermal load, the response characteristics to load fluctuations is excellent, and sufficient refrigeration performance can be obtained.

[0022] According to the third aspect of the present invention, a refrigeration system having a refrigeration cycle in which a refrigerant is compressed into a compressed refrigerant, the compressed refrigerant is condensed into a condensed refrigerant, the condensed refrigerant is decompressed by giving passage resistance into a decompressed refrigerant, the decompressed refrigerant is evaporated into an evaporated refrigerant, and then a gaseous refrigerant is separated from the evaporated refrigerant and re-compressed, wherein a passage for giving passage resistance to the refrigerant is provided at a part of a refrigerant passage in which the compressed refrigerant is condensed.

[0023] In this refrigeration system, when the thermal load of the condenser decreases, the condensation of refrigerant in the condenser is enhanced at the upstream side of the passage for giving passage resistance, and therefore only the completely liquefied refrigerant passes through the passage for giving passage resistance. Thus, the resistance of the refrigerant passing through the passage for giving passage resistance decreases, thereby increasing the flow rate. Accordingly, at the upstream side of the passage for giving passage resistance and the downstream side thereof, the condensation of refrigerant is performed efficiently. Thus, the performance of the condenser is sufficiently demonstrated.

[0024] To the contrary, when the thermal load of the condenser increases, the condensation of refrigerant in the condenser deteriorates at the upstream side of the passage for giving passage resistance, and therefore incompletely liquefied refrigerant passes through the passage for giving passage resistance. At this time, the amount of gas in the refrigerant increases, i.e., the volume of the refrigerant passing through the passage for giving passage resistance increases, resulting in increased flow resistance of the refrigerant passing through the passage for giving passage resistance, which in turn decreases the flow rate. As the flow rate decreases in this way, the condensation load at the upstream side of the passage for giving passage resistance decreases. Accordingly, the condensation will be performed fully, resulting in enhanced condenser performance.

[0025] As will be apparent from the above, since the refrigerant flow rate can be appropriately adjusted in response to fluctuations of thermal load in the condenser, this refrigeration system is excellent in response characteristics to load fluctuations. Accordingly, sufficient refrigeration performance can be obtained.

[0026] According to the fourth aspect of the present invention, a refrigeration system having a refrigeration cycle in which a refrigerant is compressed into a compressed refrigerant, the compressed refrigerant is condensed into a condensed refrigerant, the condensed refrigerant is decompressed by giving passage resistance into a decompressed refrigerant, the decompressed refrigerant is evaporated into an evaporated refrigerant, and then a gaseous refrigerant is separated from the evaporated refrigerant and re-compressed, wherein a small-cross-sectional passage, i.e., a passage whose passage cross-sectional area is smaller than that of each passage located before and after the passage is provided at a part of a refrigerant passage in which the compressed refrigerant is condensed.

[0027] In this refrigeration system, when the thermal load of the condenser decreases, the condensation of refrigerant in the condenser is enhanced at the upstream side of the small-cross-sectional passage, and therefore only the completely liquefied refrigerant passes through the small-cross-sectional passage. Thus, the resistance of the refrigerant passing through the small-cross-sectional passage decreases, thereby increasing the flow rate. Accordingly, at the upstream side of the small-cross-sectional passage and the downstream side thereof, the condensation of refrigerant is performed efficiently. Thus, the performance of the condenser is sufficiently demonstrated.

[0028] To the contrary, when the thermal load of the condenser increases, the condensation of refrigerant in the condenser deteriorates at the upstream side of the small-cross-sectional passage, and therefore incompletely liquefied refrigerant passes through the small-cross-sectional passage. At this time, the amount of gas in the refrigerant increases, i.e., the volume of the refrigerant passing through the small-cross-sectional passage increases, resulting in increased flow resistance of the refrigerant passing through the small-cross-sectional passage, which in turn decreases the flow rate. As the flow rate decreases in this way, the condensation load at the upstream side of the small-cross-sectional passage decreases. Accordingly, the condensation will be performed fully, resulting in enhanced condenser performance.

[0029] As will be apparent from the above, since the refrigerant flow rate can be appropriately adjusted in response to fluctuations of thermal load in the condenser, this refrigeration system is excellent in response characteristics to load fluctuations. Accordingly, sufficient refrigeration performance can be obtained.

[0030] According to the fifth aspect of the present invention, as the condenser in the refrigeration system according to the aforementioned first and second aspects of the present invention, the so-called multi-flow type heat exchanger is employed.

[0031] That is, a refrigeration system having a refrigeration cycle, comprises:

a compressor for compressing a refrigerant;
a condenser for condensing the refrigerant compressed by the compressor;
a decompressing tube for decompressing the refrigerant condensed by the condenser;
an evaporator for evaporating the refrigerant decompressed by the decompressing tube; and
an accumulator for separating a gaseous refrigerant from the refrigerant evaporated by the evaporator,

wherein the condenser includes:

a pair of headers disposed in parallel with each other at a certain distance;
a plurality of heat exchanging tubes disposed between the pair of headers with opposite ends thereof connected with the headers;
a partition provided in the header to group the plurality of heat exchanging tubes into a plurality of passes;

whereby the plurality of passes constitute a refrigerant passage through which the refrigerant passes in turn, the plurality of passes including a first pass and a final pass; and further includes:

decompressing means which is disposed at a part of the refrigerant passage between the first pass and the final pass to decompress the refrigerant passing through the decompressing means.

[0032] In this case, in the same manner as in the aforementioned system, the flow rate of refrigerant is appropriately adjusted by the decompressing means in response to fluctuations of thermal load. Thus, the response characteristic is excellent, and sufficient refrigeration performance can be obtained.

[0033] In this invention, an orifice-tube can be suitably used as the decompressing tube.

[0034] Furthermore, in this invention, it is preferable that the plurality of passes include the first pass, the final pass and one or a plurality of intermediate passes located between the first pass and the final pass, and wherein the one or a plurality of intermediate passes constitute a decompressing pass constituting the decompressing means.

[0035] In this case, a heat exchanging tube can be used as the decompressing means as it is. Thus, it is not necessary to attach additional components, and therefore the structure can be simplified.

[0036] In this refrigeration system, it is preferable that the intermediate pass located immediately before the final pass constitutes the decompressing means, that a total passage cross-sectional area of the decompressing pass is smaller than that of each pass located immediately before and after the decompressing pass, and that the number of heat exchanging tubes constituting the decompressing pass is smaller than that of each pass located immediately before and after the decompressing pass.

[0037] In these cases, the decompression effects can be effectively obtained by the decompressing means.

[0038] According to the sixth aspect of the present invention, a condenser for use in a decompressing-tube system constituting a refrigeration cycle which includes a compressor, a decompressing tube, an evaporator and an accumulator, comprises:

a refrigerant inlet for introducing a refrigerant;
a refrigerant outlet for discharging the refrigerant;
a refrigerant passage for leading the refrigerant introduced from the refrigerant inlet to the refrigerant outlet while condensing the refrigerant; and
decompressing means which is provided at a part of the refrigerant passage to decompress the refrigerant passing through the decompressing means.

[0039] In this invention, in the same manner as in the aforementioned cases, the flow rate of refrigerant is appropriately adjusted by the decompressing means in response to fluctuations of thermal load. Thus, the response characteristic is excellent, and sufficient refrigeration performance can be obtained.

[0040] According to the seventh aspect of the present invention, a condenser for use in a decompressing-tube system constituting a refrigeration cycle which includes a compressor, a decompressing tube, an evaporator and an accumulator, comprises:

a refrigerant inlet for introducing a refrigerant;
a refrigerant outlet for discharging the refrigerant;
a refrigerant passage for leading the refrigerant introduced from the refrigerant inlet to the refrigerant outlet while condensing the refrigerant; and
means which is provided at a part of the refrigerant passage to give passage resistance to the refrigerant.

[0041] In this invention, in the same manner as in the aforementioned cases, the flow rate of refrigerant is appropriately adjusted by the means for giving passage resistance in response to fluctuations of thermal load. Thus, the response characteristic is excellent, and sufficient refrigeration performance can be obtained.

[0042] According to the eighth aspect of the present invention, a condenser for use in a decompressing-tube system constituting a refrigeration cycle which includes a compressor, a decompressing tube, an evaporator and an accumulator, comprises:

a refrigerant inlet for introducing a refrigerant;
a refrigerant outlet for discharging the refrigerant;
a refrigerant passage for leading the refrigerant introduced from the refrigerant inlet to the refrigerant outlet while condensing the refrigerant; and
diminishing means which is provided at a part of the refrigerant passage to diminish a cross-sectional area of the passage.

[0043] In this invention, in the same manner as in the aforementioned cases, the flow rate of refrigerant is appropriately adjusted by the diminishing means in response to fluctuations of thermal load. Thus, the response characteristic is excellent, and sufficient refrigeration performance can be obtained.

[0044] In this condenser, the so-called multi-flow type condenser can be used.

[0045] According to the ninth aspect of the present invention, a condenser for use in a decompressing-tube system constituting a refrigeration cycle which includes a compressor, a decompressing tube, an evaporator and an accumulator, comprises:

a pair of headers disposed in parallel with each other at a certain distance;
a plurality of heat exchanging tubes disposed between the pair of headers with opposite ends thereof connected with the headers;
a partition provided in the header;

wherein the plurality of heat exchanging tubes are grouped by the partition into a plurality of passes constituting a refrigerant passage through which the refrigerant passes in turn, the plurality of passes including a first pass and a final pass; and further comprises:

decompressing means which is disposed at a part of the refrigerant passage between the first pass and the final pass to decompress the refrigerant passing through the decompressing means.

[0046] In this invention, in the same manner as in the aforementioned cases, the response characteristic is excellent, and sufficient refrigeration performance can be obtained.

[0047] In this condenser, it is preferable that the plurality of passes include the first pass, the final pass and one or a plurality of intermediate passes located between the first pass and the final pass, and wherein the one or a plurality of intermediate passes constitute a decompressing pass constituting the decompressing means.

[0048] In this case, a heat exchanging tube can be used as the decompressing means as it is.

[0049] Other objects and the features will be apparent from the following detailed description of the present invention with reference to the attached drawings.

Brief Description of the Drawings

[0050] The present invention will be more fully described and better understood from the following description, taken with the appended drawings.

Fig. 1 shows a refrigerant circuit of a refrigeration system according to an embodiment of the present invention.
Fig. 2 is a front view showing the condenser employed in the refrigeration system of the embodiment.
Fig. 3 is a cross-sectional view showing a heat exchanging tube used in the condenser of the embodiment.
Fig. 4 is an exploded perspective view showing a heat exchanging tube for condensers according to a first modification of the present invention.
Fig. 5A is a side cross-sectional view showing the heat exchanging tube of the first modification, and Fig. 5B is the front cross-sectional view showing the heat exchanging tube of the first modification.
Fig. 6 is a cross-sectional view showing a heat exchanging tube for condensers according to a second modification of the present invention.
Fig. 7 is a Mollier diagram of the refrigeration cycle in the refrigeration system according to the present invention.
Fig. 8 shows a refrigerant circuit of a refrigeration system according to a third embodiment of the present invention.
Fig. 9 is a front view showing a condenser according to a fourth embodiment of the present invention.
Fig. 10 shows a refrigerant circuit of a refrigeration system according to the fourth embodiment of the present invention.
Fig. 11 is a cross-sectional view showing a heat exchanging tube for a decompression pass according to a fourth modification of the present invention.
Fig. 12 is a graph showing the relation of the cooling performance, the compressor discharge pressure and the coefficient of performance relative to the compressor rotating speed in the refrigeration system.
Fig. 13 shows a refrigerant circuit of a conventional orifice-tube system.

Best Mode for Carrying out the Invention

[0051] Fig. 1 shows a refrigerant circuit of the refrigeration system according to an embodiment of the present invention. Fig. 2 is a front view showing a condenser 10 employed in the refrigeration system.

[0052] As shown in these figures, this refrigeration system is an orifice-tube system. In this system, the gaseous refrigerant of high temperature and high pressure sent out from the compressor 1 is introduced into the condenser 10 and condensed therein. The condensed refrigerant is decompressed by the orifice-tube 3 and then introduced into the evaporator 4. In the evaporator 4, the refrigerant exchanges heat with the ambient air to be evaporated. Then, only the gaseous refrigerant is extracted by the accumulator 5 and then returns to the aforementioned compressor 1.

[0053] In this refrigeration system, the condenser 10 is the so-called multi-flow type heat exchanger, and is provided with a pair of right and left headers 11 and 11 vertically disposed at a certain distance. Between these headers 11 and 11, a plurality of heat exchanging flat tubes 12 are disposed horizontally in parallel at certain intervals in the direction of up-and-down with the opposite ends thereof connected with the headers 11 and 11. A corrugated fin 13 is disposed between the adjacent heat exchanging tubes 12 and 12. Furthermore, a corrugated fin 13 is arranged on each of the outermost heat exchanging tubes 12. Disposed on the outermost fin 13 is a side plate 14.

[0054] As the heat exchanging tube 12, as shown in Fig. 3, the so-called harmonica tube having a plurality of refrigerant passages 12a arranged side by side inside is generally used.

[0055] In the present invention, in place of the aforementioned heat exchanging tube, the heat exchanging tube as shown in Figs. 4 and 5 can also be preferably used. This heat exchanging tube 12 is provided with a plurality of refrigerant passages 12a. The adjacent refrigerant passages 12a and 12a are communicated with each other via a plurality of communication apertures 12c formed in the partition wall 12b which partitions the adjacent passages 12a and 12a. Furthermore, the heat exchanging tube 12 with numerous inner fins 12d as shown in Fig. 6 can also be suitably used. In this heat exchanging tube 12, a plurality of inner fins 12d are protruded from the inner surface of each refrigerant passage 12a.

[0056] As shown in Figs. 1 and 2, partitions 15 and 16 for dividing the inside of the header 11 are provided at the predetermined positions of the headers 11 and 11. In this embodiment, the 1st to 10th heat exchanging tubes counted from the uppermost tube constitute the first pass P1. The 11th heat exchanging tube 12 counted from the uppermost tube constitutes the second pass P2. The 12th to 14th heat exchanging tubes 12 counted from the uppermost tube constitute the third pass P3 which is the final pass.

[0057] In this embodiment, the first pass P1 constitutes a first condensing portion. The second pass P2 constitutes a decompressing pass (a decompressing means, a decompressing passage), and the third pass P3 constitutes a second condensing portion (re-condensing portion).

[0058] Furthermore, a refrigerant inlet 11a is provided at the upper portion of one of the headers 11 (right-hand side header), and a refrigerant outlet 11b is provided at the lower portion of the other header 11 (left-hand side header). The refrigerant introduced into the header 11 via the refrigerant inlet 11a passes through the first pass to the third pass in a meandering manner in turn, and then flows out of the refrigerant outlet 11b.

[0059] As shown in Fig. 1, this condenser 10 is connected with the compressor 1, the orifice-tube 3, the evaporator 4 and the accumulator 5 via refrigerant tubes to form a refrigeration system for automobiles.

[0060] Next, the operation of the refrigeration system of this embodiment will be explained with reference to the Mollier diagram shown in Fig. 7.

[0061] In this diagram, the refrigerant at the region on the left side of the liquidus curve is in a liquid phase state. The refrigerant at the region between the liquidus curve and the vapor line is in a vapor-liquid mixed phase state. The refrigerant at the region on the right side of the vapor line is in a vapor phase.

[0062] In this refrigeration system, the refrigerant compressed by the compressor 1 shifts from the point A to the point B to become a gaseous refrigerant of high temperature and high pressure. The gaseous refrigerant is then introduced into the condenser 10. In the condenser 10, the refrigerant passed through the first pass P1 is condensed, and shifts from the point B to the point C1. Subsequently, the liquefied refrigerant passes through the decompressing pass P2 to be decompressed, and shifts from the point C1 to the point C2. Thereafter, the refrigerant passes through the third pass P3 to be re-condensed, and shifts from the point C2 to the point C3.

[0063] The condensed refrigerant passes through the orifice-tube 3 to be decompressed, and shifts from the point C3 to the point D in which the refrigerant is in a vapor-liquid mixed phase state. Then, the refrigerant is sent to the evaporator 4, and exchanges heat with the ambient air therein to be evaporated. Thus, the refrigerant shifts from the point D to the point A, and then returns to the aforementioned compressor 1.

[0064] In this refrigeration system, when the thermal load of the condenser increases suddenly, the condensation of refrigerant in the first pass P1 deteriorates, and therefore incompletely liquefied refrigerant is introduced into the decompressing pass P2. At this time, the amount of gas in the refrigerant increases, i.e., the volume of the refrigerant passing through the decompressing pass P2 increases, resulting in increased flow resistance of the refrigerant passing through the decompressing pass P2, which in turn increases the flow resistance of the refrigerant to thereby decrease the flow rate. As the flow rate decreases in the decompressing pass P2, the condensation load at the upstream side of the decompressing means pass P2, i.e., the condensation load at the first pass P1 decreases. Accordingly, the condensation and decompression will be performed smoothly in each pass P1 to P3, resulting in enhanced condenser performance.

[0065] To the contrary, in this refrigeration system, when the thermal load of the condenser decreases suddenly, the condensation of refrigerant in the first pass P1 is performed fully, and therefore only the completely liquefied refrigerant is introduced into the decompressing pass P2. Thus, the resistance of the refrigerant passing through the decompressing pass P2 decreases, thereby increasing the flow rate. Accordingly, at the upstream side of the decompressing pass P2, i.e., at the fist pass P1, the condensation of refrigerant is performed efficiently. Thus, the refrigerant is effectively condensed or decompressed in each pass P1 to P3. Therefore, the performance of the condenser is sufficiently demonstrated.

[0066] Thus, in the refrigeration system of this embodiment, since the decompressing pass P2 has a self-control function for controlling the refrigerant flow rate in response to fluctuations of thermal load. Accordingly, the circulation flow rate of refrigerant in the refrigeration cycle can be adjusted appropriately. Therefore, the response characteristic to load fluctuations is excellent, and thus sufficient refrigeration performance can be obtained.

[0067] Furthermore, in the refrigeration system of this embodiment, the refrigerant is initially condensed in the first pass P1 of the condenser 10 to release the heat, and then secondarily condensed in the second pass P2 to release the heat. Therefore, sufficient heat release can be secured, which in turn can secure a large enthalpy difference (D-A) at the time of evaporation. Thus, outstanding refrigeration effects can be obtained.

[0068] Furthermore, in the condenser 10, since the amount of releasing heat is increased by the secondary condensation accompanying phase changes, the heat can be effectively released. In other words, in the condenser 10 of this embodiment, since almost the entire region thereof constitutes a condensing portion, the heat radiation of the refrigerant can be effectively performed, resulting in excellent condensing performance. Accordingly, the refrigerant can be condensed assuredly while preventing the rise of the refrigerant pressure within the refrigeration cycle. Thus, the load of the compressor 1 can be decreased. Accordingly, it becomes possible to prevent the enlargement of the compressor 1, resulting in a small and lightweight refrigeration system, an enhanced fuel consumption rate at the time of mounting the system on an automobile, a reduced amount of refrigerant and a decreased cost.

[0069] In the aforementioned embodiment, the number of passes and the number of tubes constituting each pass, especially the number of tubes constituting the decompressing pass, are not limited to the above. For example, as shown in Fig. 8, it is possible that four passes P1 to P4 are provided and the number of the third pass P3 constitutes the decompressing pass including two tubes.

[0070] Furthermore, in this invention, two or more decompressing passes may be provided. For example, as shown in Figs. 9 and 10, the headers 11 and 11 may be partitioned by partitions 15 to 17 to form four passes P1 to P4, and the second pass P2 and the third pass P3, each including one tube 12, may constitute a decompressing pass, respectively.

[0071] Furthermore, in the present invention, in order to enhance the decompression effects, a tube constituting a decompressing pass may be constituted by a tube different from the other tube in structure. For example, as shown in Fig. 11, the so-called harmonica tube having a plurality of refrigerant small circular passages 12a may be used as a heat exchanging tube for a decompressing pass.

[0072] Furthermore, as a tube constituting a decompressing pass, it is not necessary to use a straight tube. For example, it may be possible to employ a serpentine type tube bent in a zigzag manner for a serpentine type heat exchanger or a capillary tube.

[0073] Furthermore, it is not necessary to constitute the decompressing means by a heat exchanging tube. For example, it is possible to provide a decompressing means such as a partitioning plate with an orifice formed in a tube.

[0074] Furthermore, in the present invention, it is not necessary to provide a decompressing means in a heat exchanging tube, and a decompressing means may be provided in a header. In short, it is enough that decompressing means or a decompressing passage is provided at a part of a refrigerant passage between a refrigerant inlet and a refrigerant outlet.

<Example>

[0075] The so-called multi-flow type condenser having four passes, the first pass to the fourth four pass, was prepared. The first pass was constituted by 19 heat exchanging tubes, the second pass was constituted by 8 heat exchanging tubes, the third pass (decompressing pass) was constituted by 1 heat exchanging tube, and the fourth pass was constituted by 7 heat exchanging tubes.

[0076] In the refrigeration cycle including the condenser as shown in Fig. 1, the cooling performance (kW), the compressor discharge pressure (kPa) and the coefficient of performance to the compressor rotation speed (rpm) were measured.

<Comparative example>

[0077] The so-called multi-flow type condenser having a first pass constituted by 14 heat exchanging tubes, a second pass constituted by 10 tubes, a third pass constituted by 7 tubes and a fourth pass constituted by 4 tubes was prepared. By using the condenser, the same examination as in the aforementioned example was performed.

[0078] The measured results of the aforementioned example and comparative example are shown in the graph shown in Fig. 12. In the graph, "W" denotes the example, and "S" denotes the comparative example. Furthermore, the round mark denotes the cooling performance, the square mark denotes a coefficient of performance and "x" mark denotes the compressor discharge pressure.

[0079] As will apparent from the graph, it is understood that, in each of the cooling performance, the compressor discharge pressure and the coefficient of performance, the refrigeration cycle of the example is superior to that of the comparative example.

[0080] As mentioned above, according to the present invention, in a decompressing-tube system such as an orifice-tube system, since the flow rate of refrigerant is appropriately adjusted by the decompressing means or the decompressing passage in response to fluctuations of thermal load in the condensing portion, the response characteristics to load fluctuations is excellent, and sufficient refrigeration performance can be obtained.

[0081] The terms and expressions which have been employed herein are used as terms of description and not of limitation, and there is no intent, in the use of such terms and expressions, of excluding any of the equivalents of the features shown and described or portions thereof, but it is recognized that various modifications are possible within the scope of the invention claimed.

Industrial Applicability

[0082] As described above, the refrigeration system and the condenser for the decompressing tube system according to the present invention can especially be applied to a refrigeration cycle for car air-conditioners since the response characteristic to fluctuations of thermal load is excellent and sufficient refrigeration performance can be obtained.

Claims

1. A refrigeration system having a refrigeration cycle, said refrigeration system comprising:

   a compressor for compressing a refrigerant;
   a condenser for condensing the refrigerant compressed by said compressor;
   a decompressing tube for decompressing the refrigerant condensed by said condenser;
   an evaporator for evaporating the refrigerant decompressed by said decompressing tube;

and
an accumulator for separating a gaseous refrigerant from the refrigerant evaporated by said evaporator,

wherein said condenser includes a refrigerant inlet for introducing the refrigerant compressed by said compressor, a refrigerant outlet for discharging the refrigerant condensed by said condenser, a refrigerant passage for leading the refrigerant introduced from said refrigerant inlet to said refrigerant outlet while condensing the refrigerant, and decompressing means provided at a part of said refrigerant passage to decompress the refrigerant passing through said decompressing means.

2. The refrigeration system as recited in claim 1, wherein said decompressing tube is an orifice-tube.

3. The refrigeration system as recited in claim 1, at least a part of the refrigerant condensed at an upstream side of said decompressing means in said refrigerant passage is decompressed by said decompressing means into a low-pressure gaseous refrigerant, and the low-pressure gaseous refrigerant is re-condensed at a downstream side of said decompressing means in said refrigerant passage.

4. The refrigeration system as recited in claim 1, at least a part of the refrigerant condensed at an upstream side of said decompressing means in said refrigerant passage is decompressed by said decompressing means into a low-pressure gaseous refrigerant, and the low-pressure gaseous refrigerant is re-condensed at a downstream side of said decompressing means in said refrigerant passage.

5. A refrigeration system having a refrigeration cycle in which a refrigerant is compressed into a compressed refrigerant, the compressed refrigerant is condensed into a condensed refrigerant, the condensed refrigerant is decompressed by giving passage resistance into a decompressed refrigerant, the decompressed refrigerant is evaporated into an evaporated refrigerant, and then a gaseous refrigerant is separated from the evaporated refrigerant and re-compressed,

wherein a decompressing passage for decompressing the refrigerant is provided at a part of a refrigerant passage in which the compressed refrigerant is condensed.

6. A refrigeration system having a refrigeration cycle in which a refrigerant is compressed into a compressed refrigerant, the compressed refrigerant is condensed into a condensed refrigerant, the condensed refrigerant is decompressed by giving passage resistance into a decompressed refrigerant, the decompressed refrigerant is evaporated into an evaporated refrigerant, and then a gaseous refrigerant is separated from the evaporated refrigerant and re-compressed,

wherein a passage for giving passage resistance to the refrigerant is provided at a part of a refrigerant passage in which the compressed refrigerant is condensed.

7. A refrigeration system having a refrigeration cycle in which a refrigerant is compressed into a compressed refrigerant, the compressed refrigerant is condensed into a condensed refrigerant, the condensed refrigerant is decompressed by giving passage resistance into a decompressed refrigerant, the decompressed refrigerant is evaporated into an evaporated refrigerant, and then a gaseous refrigerant is separated from the evaporated refrigerant and re-compressed,

wherein a passage whose cross-sectional area is smaller than that of each passage located before and after said passage is provided at a part of a refrigerant passage in which the compressed refrigerant is condensed.

8. A refrigeration system having a refrigeration cycle, said refrigeration system comprising:

a compressor for compressing a refrigerant;
a condenser for condensing the refrigerant compressed by said compressor;
a decompressing tube for decompressing the refrigerant condensed by said condenser;
an evaporator for evaporating the refrigerant decompressed by said decompressing tube; and
an accumulator for separating a gaseous refrigerant from the refrigerant evaporated by said evaporator,

wherein said condenser includes:

a pair of headers disposed in parallel with each other at a certain distance;
a plurality of heat exchanging tubes disposed between said pair of headers with opposite ends thereof connected with said headers; and
a partition provided in said header to group said plurality of heat exchanging tubes into a plurality of passes;

whereby said plurality of passes constitute a refrigerant passage through which the refrigerant passes in turn, said plurality of passes including a first pass and a final pass; and further includes:

decompressing means which is disposed at a part of said refrigerant passage between said first pass and said final pass to decompress the refrigerant passing through said decompressing means.

9. The refrigeration system as recited in claim 8, wherein said decompressing tube is an orifice-tube.

10. The refrigeration system as recited in claim 8, wherein said plurality of passes include said first pass, said final pass and one or a plurality of intermediate passes located between said first pass and said final pass, and wherein said one or a plurality of intermediate passes constitute a decompressing pass constituting said decompressing means.

11. The refrigeration system as recited in claim 9, wherein said plurality of passes include said first pass, said final pass and one or a plurality of intermediate passes located between said first pass and said final pass, and wherein said one or a plurality of intermediate passes constitute a decompressing pass constituting said decompressing means.

12. The refrigeration system as recited in claim 11, wherein said intermediate pass located immediately before said final pass constitutes said decompressing means.

13. The refrigeration system as recited in claim 11, wherein a total passage cross-sectional area of said decompressing pass is smaller than that of each pass located immediately before and after said decompressing pass.

14. The refrigeration system as recited in claim 12, wherein a total passage cross-sectional area of said decompressing pass is smaller than that of each pass located immediately before and after said decompressing pass.

15. The refrigeration system as recited in claim 10, wherein the number of heat exchanging tubes constituting said decompressing pass is smaller than that of each pass located immediately before and after said decompressing pass.

16. The refrigeration system as recited in claim 11, wherein the number of heat exchanging tubes constituting said decompressing pass is smaller than that of each pass located immediately before and after said decompressing pass.

17. The refrigeration system as recited in claim 12, wherein the number of heat exchanging tubes constituting said decompressing pass is smaller than that of each pass located immediately before and after said decompressing pass.

18. A condenser for use in a decompressing-tube system constituting a refrigeration cycle which includes a compressor, a decompressing tube, an evaporator and an accumulator, said condenser comprising:

a refrigerant inlet for introducing a refrigerant;
a refrigerant outlet for discharging the refrigerant;
a refrigerant passage for leading the refrigerant introduced from said refrigerant inlet to said refrigerant outlet while condensing the refrigerant; and
decompressing means which is provided at a part of said refrigerant passage to decompress the refrigerant passing through said decompressing means.

19. A condenser for use in a decompressing-tube system constituting a refrigeration cycle which includes a compressor, a decompressing tube, an evaporator and an accumulator, said condenser comprising:

a refrigerant inlet for introducing a refrigerant;
a refrigerant outlet for discharging the refrigerant;
a refrigerant passage for leading the refrigerant introduced from said refrigerant inlet to said refrigerant outlet while condensing the refrigerant; and
means which is provided at a part of said refrigerant passage to give passage resistance to the refrigerant.

20. A condenser for use in a decompressing-tube system constituting a refrigeration cycle which includes a compressor, a decompressing tube, an evaporator and an accumulator, said condenser comprising:

a refrigerant inlet for introducing a refrigerant;
a refrigerant outlet for discharging the refrigerant;
a refrigerant passage for leading the refrigerant introduced from said refrigerant inlet to said refrigerant outlet while condensing the refrigerant; and
diminishing means which is provided at a part of said refrigerant passage to diminish a cross-sectional area of the passage.

21. A condenser for use in a decompressing-tube system constituting a refrigeration cycle which includes a compressor, a decompressing tube, an evaporator and an accumulator, said condenser comprising:

a pair of headers disposed in parallel with each other at a certain distance;
a plurality of heat exchanging tubes disposed between said pair of headers with opposite

ends thereof connected with said headers;
a partition provided in said header;

wherein said plurality of heat exchanging tubes are grouped by said partition into a plurality of passes constituting a refrigerant passage through which the refrigerant passes in turn, said plurality of passes including a first pass and a final pass; and, further comprising:

decompressing means which is provided at a part of said refrigerant passage between said first pass and said final pass to decompress the refrigerant passing through said decompressing means.

22. The refrigeration system as recited in claim 21, wherein said plurality of passes include said first pass, said final pass and one or a plurality of intermediate passes located between said first pass and said final pass, and wherein said one or a plurality of intermediate passes constitute a decompressing pass constituting said decompressing means.

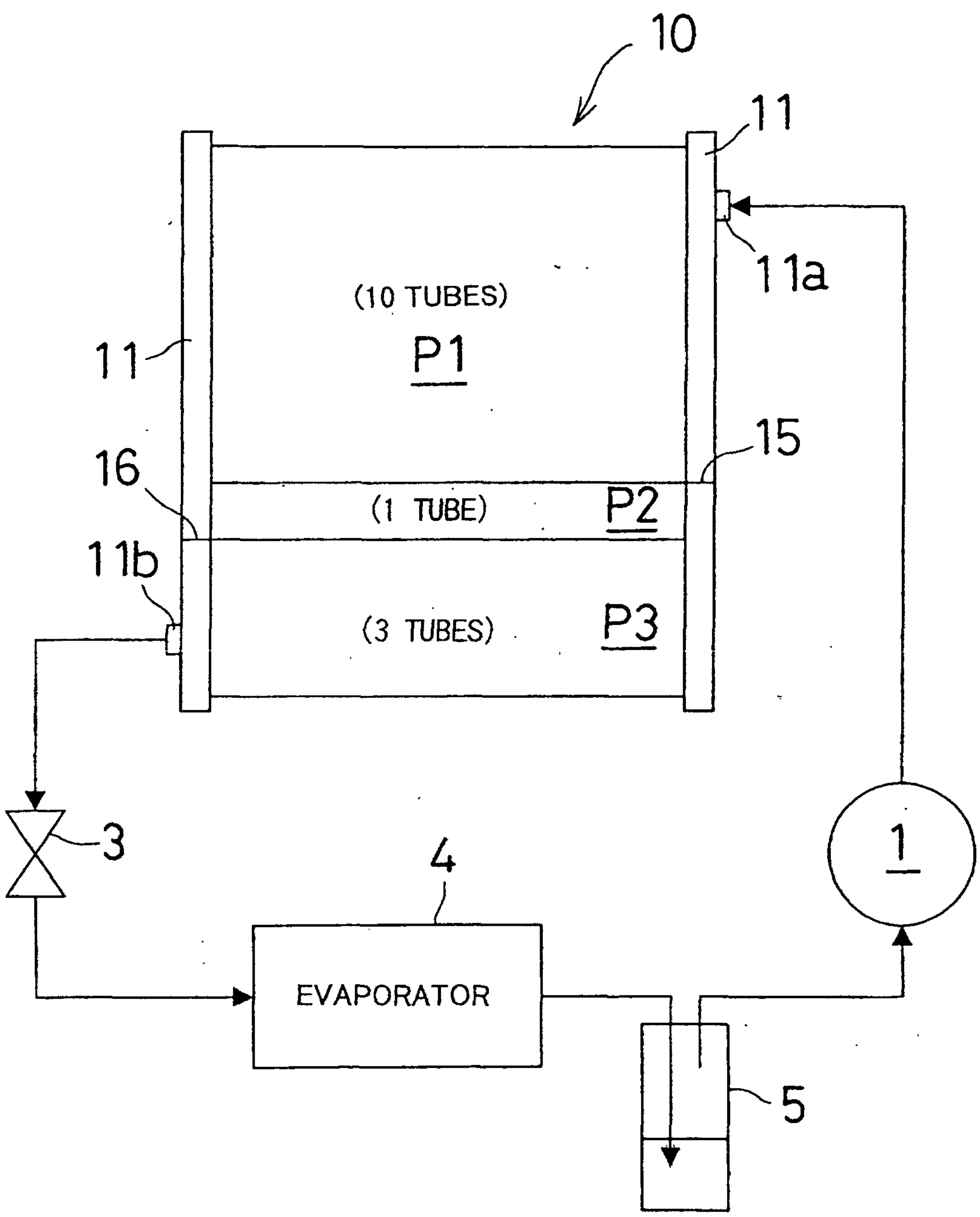
10
11
11a
(10 TUBES)
P1
11
15
16
(1 TUBE)
P2
11b
(3 TUBES)
P3
3
4
1
EVAPORATOR
5

FIG.1

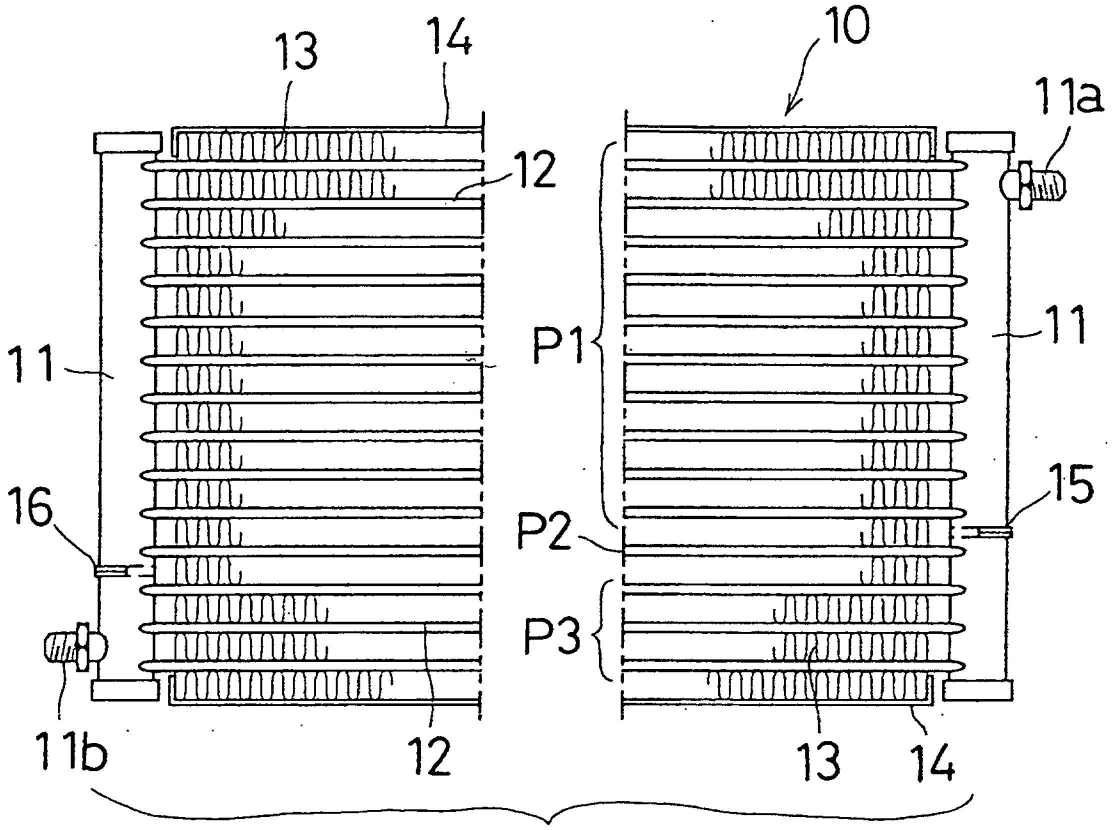

FIG.2

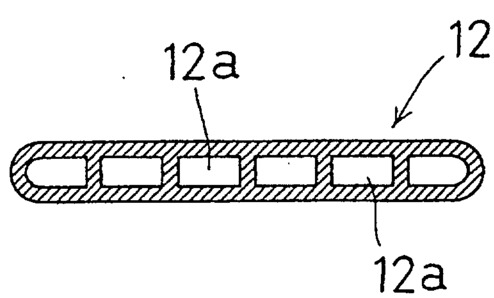

FIG.3

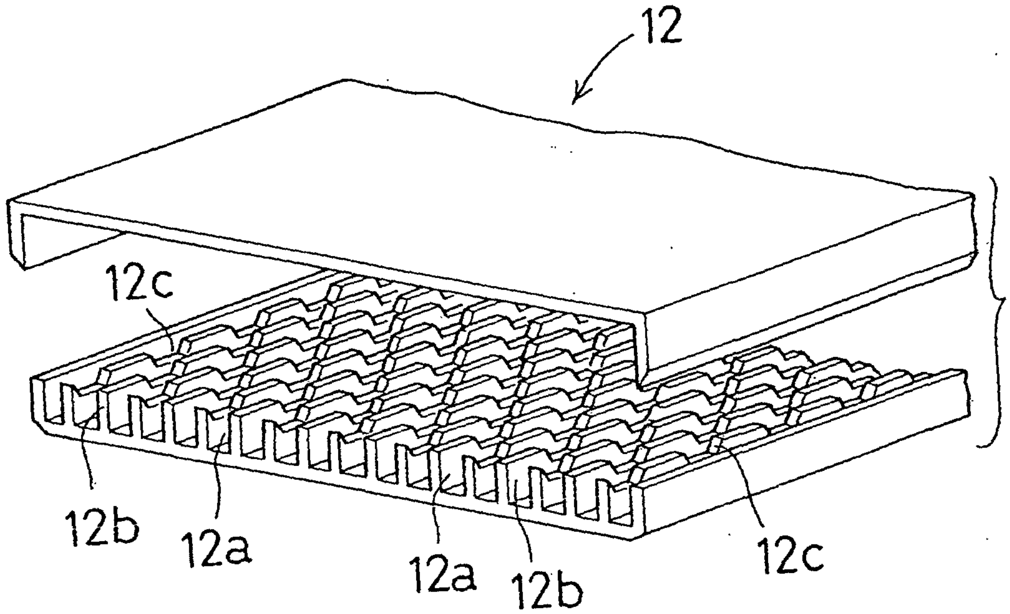

FIG.4

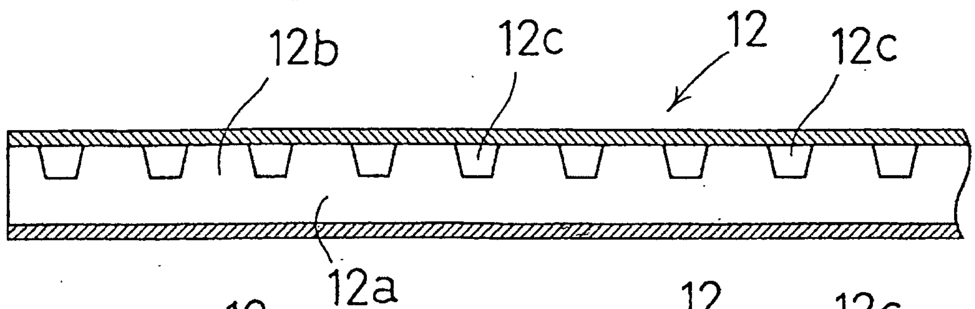

FIG.5A

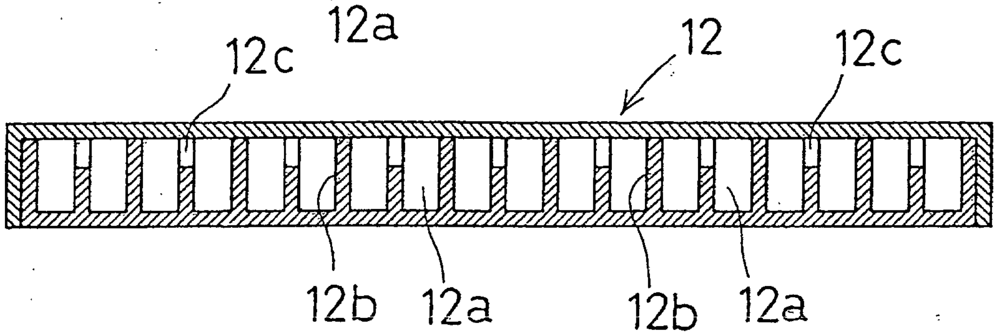

FIG.5B

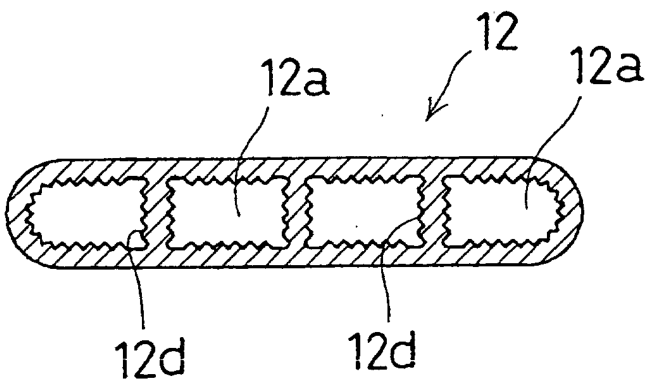

FIG.6

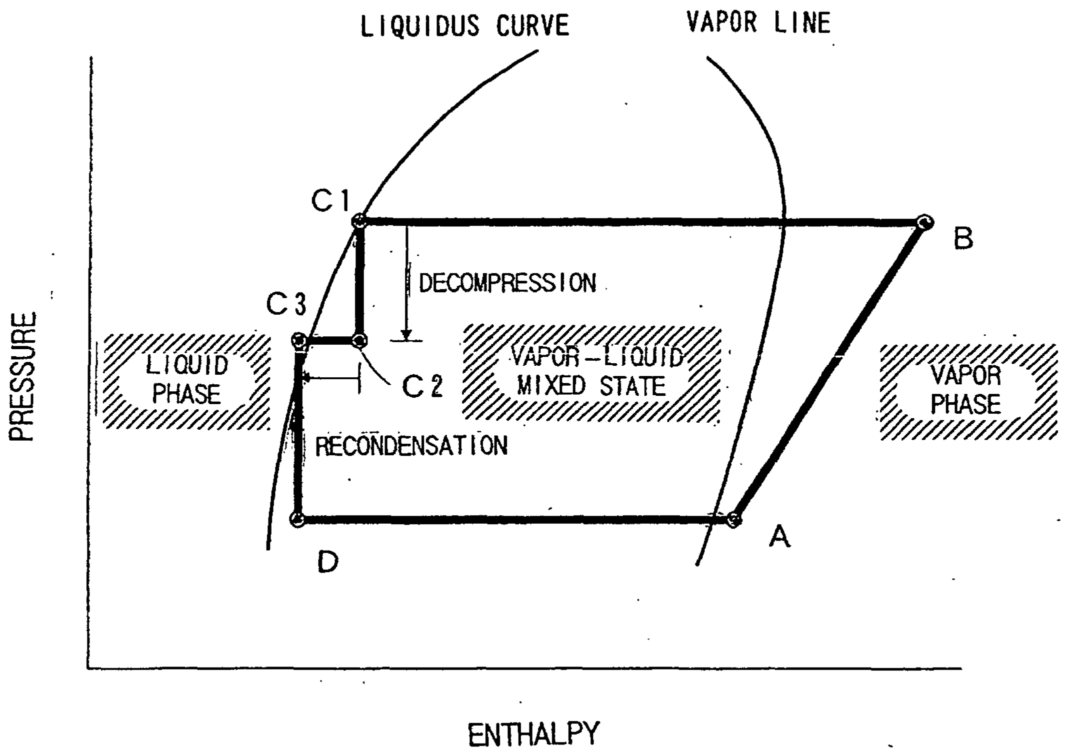
LIQUIDUS CURVE
VAPOR LINE
C1
B
DECOMPRESSION
C3
LIQUID PHASE
C2
VAPOR-LIQUID MIXED STATE
VAPOR PHASE
RECONDENSATION
A
D
PRESSURE
ENTHALPY

FIG.7

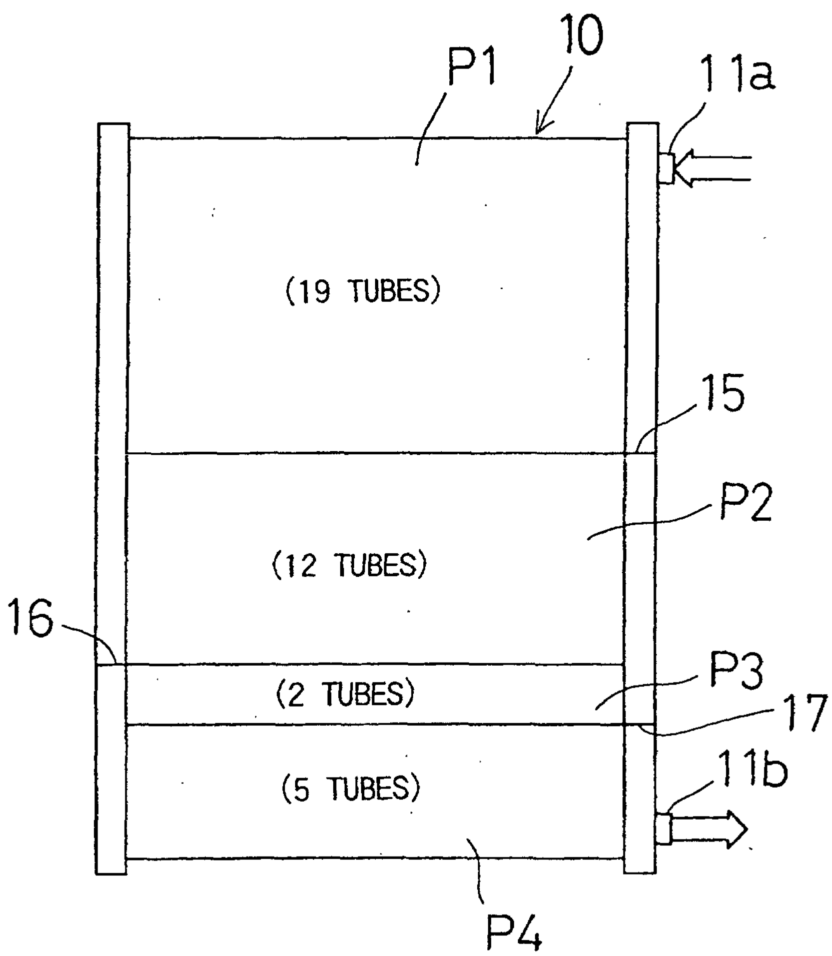

FIG.8

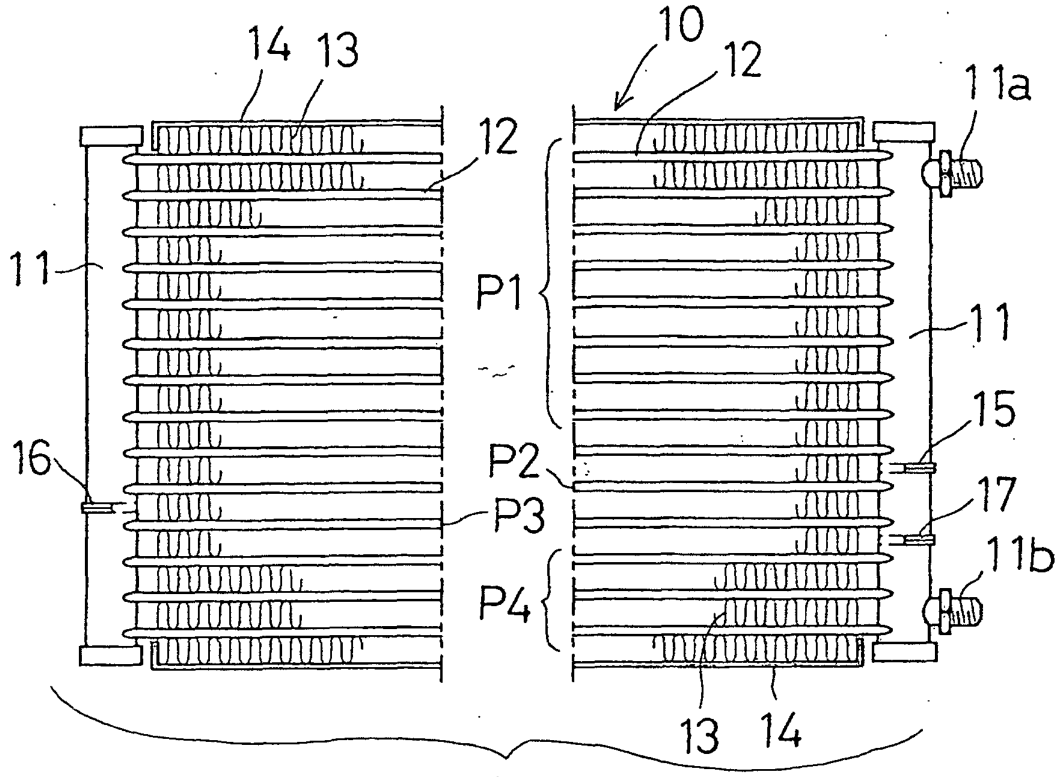
14
13
12
11
16
10
12
11a
P1
11
15
P2
17
P3
11b
P4
13
14

FIG.9

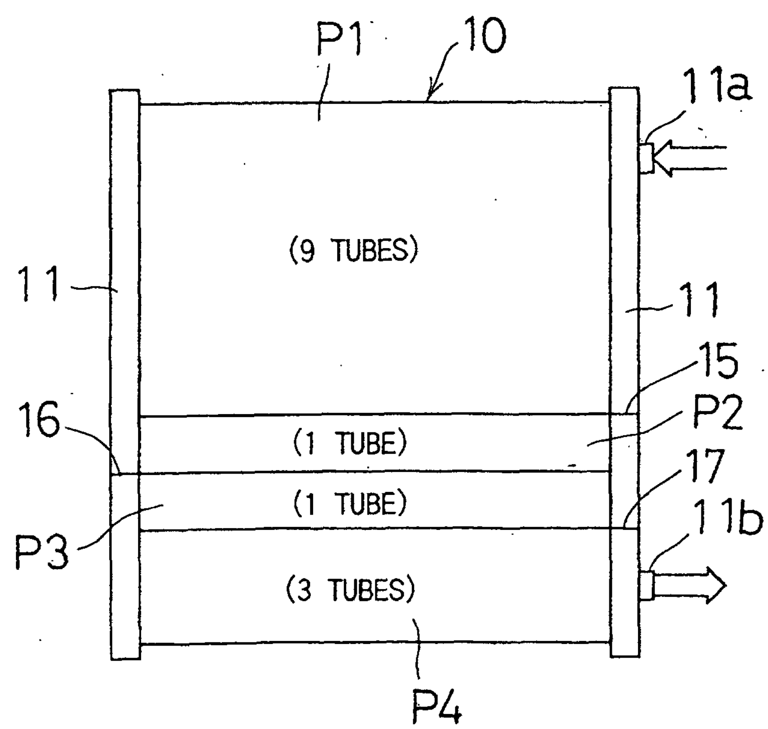
P1
10
11a
(9 TUBES)
11
11
15
16
P2
(1 TUBE)
17
(1 TUBE)
11b
P3
(3 TUBES)
P4

FIG.10

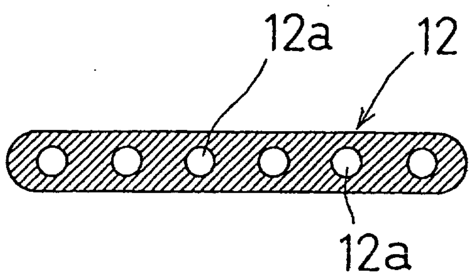

FIG.11

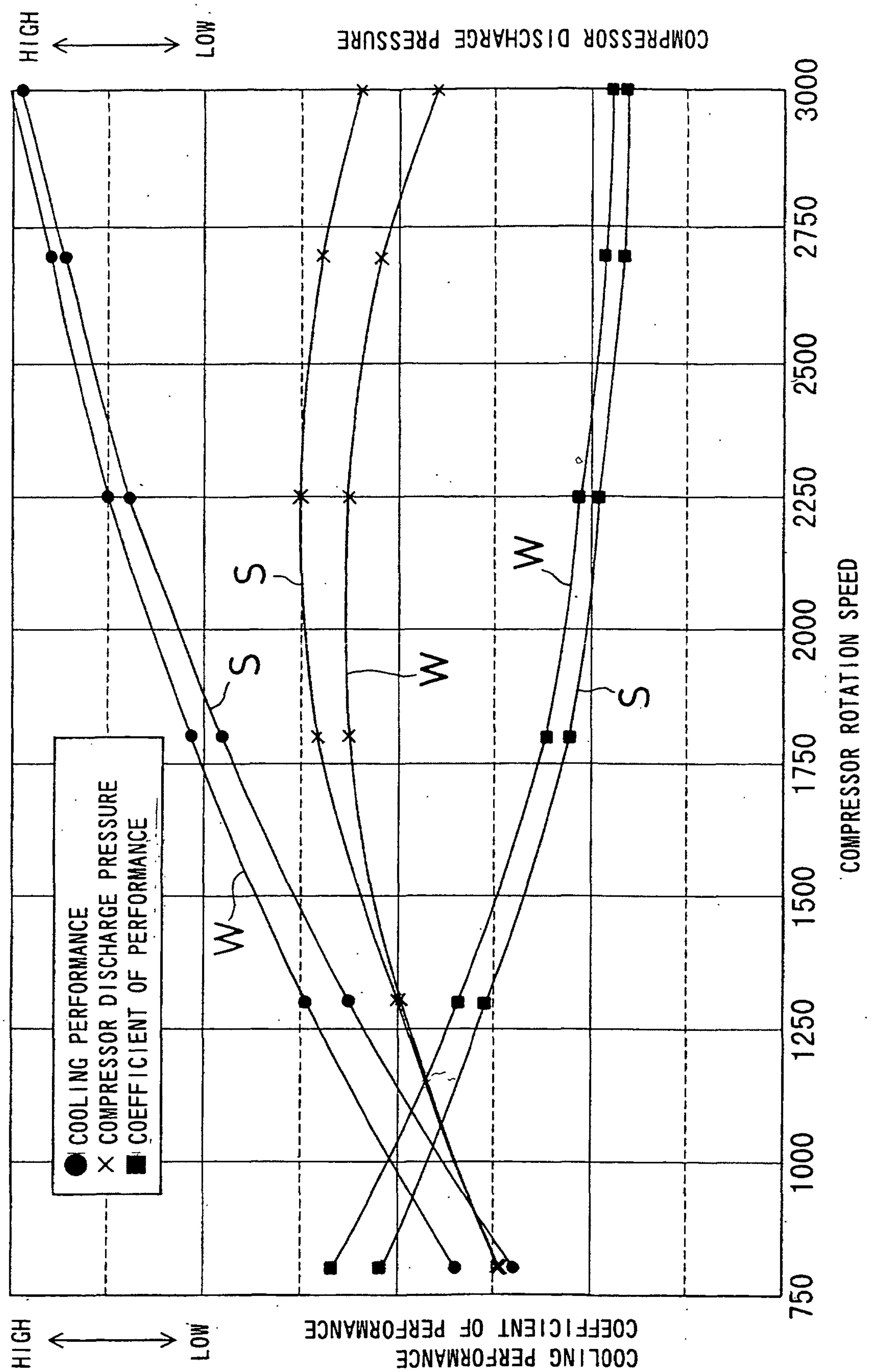
COOLING PERFORMANCE
COMPRESSOR DISCHARGE PRESSURE
COEFFICIENT OF PERFORMANCE
HIGH
LOW
COMPRESSOR DISCHARGE PRESSURE
COOLING PERFORMANCE
COEFFICIENT OF PERFORMANCE
S
W
750
1000
1250
1500
1750
2000
2250
2500
2750
3000
COMPRESSOR ROTATION SPEED

FIG.12

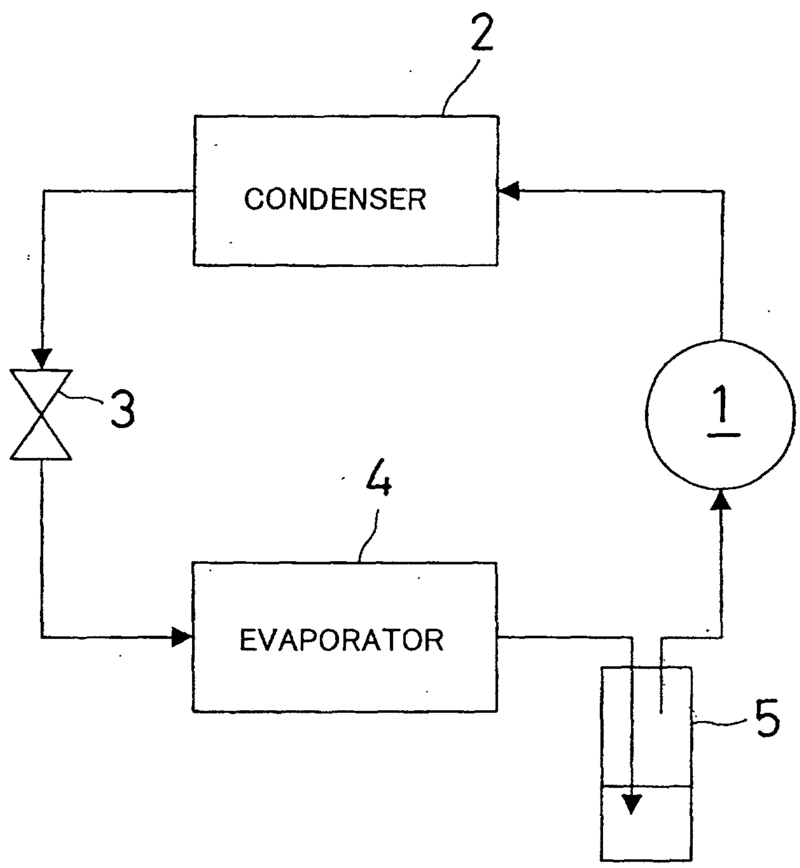

FIG.13

INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP02/09195

A. CLASSIFICATION OF SUBJECT MATTER
Int.Cl[7] F25B39/04, B60H1/32, F25B1/00, F25B41/06

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

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
Int.Cl[7] F25B39/04, B60H1/32, F25B1/00, F25B41/06

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

Jitsuyo Shinan Koho	1922-1996	Toroku Jitsuyo Shinan Koho	1994-2002
Kokai Jitsuyo Shinan Koho	1971-2002	Jitsuyo Shinan Toroku Koho	1996-2002

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

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
	US 006250103 B1 (Showa Denko Kabushiki Kaisha), 26 June, 2001 (26.06.01),	
X	Full text; Figs. 1 to 24	1-4,7-21
Y	Full text; Figs. 1 to 24	5,6
	& EP 001043552 A1 & US 2001/0035025 A1	
	& CZ 020001281 A & JP 2000-356436 A	
	& JP 2001-227844 A & JP 2001-235255 A	
Y	JP 2001-227844 A (Showa Denko Kabushiki Kaisha), 24 August, 2001 (24.08.01), Full text; Figs. 1 to 11 & EP 001043552 A1 & US 006250103 B1 & US 2001/0035025 A1 & CZ 020001281 A	1-21

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

* Special categories of cited documents:
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"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention
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"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art
"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
28 October, 2002 (28.10.02)	12 November, 2002 (12.11.02)

Name and mailing address of the ISA/	Authorized officer
Japanese Patent Office	
Facsimile No.	Telephone No.

Form PCT/ISA/210 (second sheet) (July 1998)