IPOS COPY 9802555-4

世界知的所有権機関
国際事務局

PCT

特許協力条約に基づいて公開された国際出願

(51) 国際特許分類6 F28D 15/02	A1	(11) 国際公開番号	WO97/15799
		(43) 国際公開日	1997年5月1日(01.05.97)

(21) 国際出願番号 PCT/JP96/03129

(22) 国際出願日 1996年10月24日(24.10.96)

(30) 優先権データ
特願平7/275265 1995年10月24日(24.10.95) JP
特願平8/174751 1996年7月4日(04.07.96) JP

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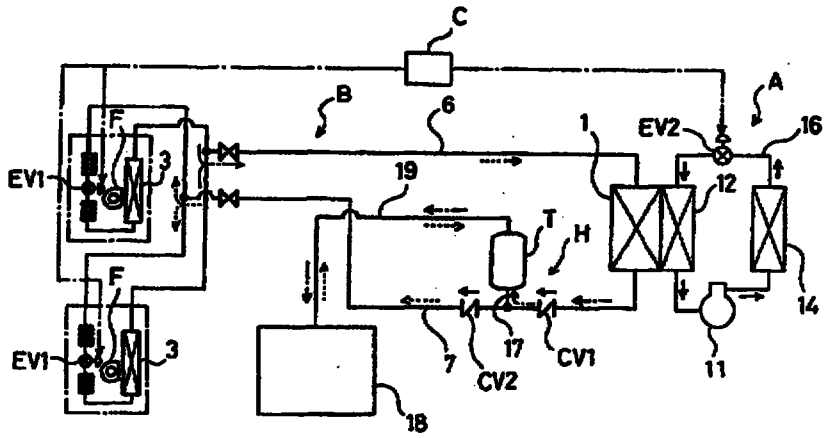
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(81) 指定国 AL, AM, AU, BB, BG, BR, CA, CN, CZ, EE, FI, GE, HU, IS, KG, KR, LK, LR, LT, LV, MD, MG, MK, MN, MX, NO, NZ, PL, RO, SG, SI, SK, TR, TT, UA, US, UZ, VN, ARIPO特許 (KE, LS, MW, SD, SZ, UG), ユーラシア特許 (AM, AZ, BY, KG, KZ, MD, RU, TJ, TM), 欧州特許 (AT, BE, CH, DE, DK, ES, FI, FR, GB, GR, IE, IT, LU, MC, NL, PT, SE), OAPI特許 (BF, BJ, CF, CG, CI, CM, GA, GN, ML, MR, NE, SN, TD, TG).

添付公開書類
国際調査報告書

(54)Title: HEAT TRANSPORT SYSTEM

(54)発明の名称 熱搬送装置



(57) Abstract

A heat exchanger (1) on the secondary heat source, which exchanges heat with a heat exchanger (12) on the primary heat source in a primary cooling circuit (A), is connected with an indoor heat exchanger (3) through a gas pipe (6) and a liquid pipe (7). A tank (T) storing a liquid cooling medium is connected at its lower end to the liquid pipe (7) and at its upper and to a pressure adjustment mechanism (18). Check valves (CV1 and CV2) are disposed on both sides of the connecting portion of the tank (T) with respect to the liquid pipe (7). The internal pressure of the tank (T) is changed over alternately between a high pressure state and a low pressure state by the pressure adjustment mechanism (18) so that the liquid cooling medium is supplied to the indoor heat exchanger (3) at the time of the high pressure operation, and the liquid cooling medium is recovered from the heat exchanger (1) on the secondary side to the tank (T) and is circulated by a secondary cooling circuit (B) at the time of the low pressure operation.

（５７）要約

１次側冷媒回路（Ａ）の１次側熱源熱交換器（12）と熱交換可能な２次側熱源熱交換器（１）と室内熱交換器（３）とをガス配管（６）及び液配管（７）によって連結する。液配管（７）に、液冷媒を貯留したタンク（Ｔ）の下端を連結する。タンク（Ｔ）の上端に加減圧機構（18）を連結する。液配管（７）に対するタンク（Ｔ）の連結部の両側に逆止弁（CV1，CV2）を設ける。加減圧機構（18）によりタンク（Ｔ）内を高圧状態と低圧状態との間で交互に切換え、高圧作用時にタンク（Ｔ）から室内熱交換器（３）へ液冷媒を供給し、低圧作用時に２次側熱源熱交換器（１）からタンク（Ｔ）へ液冷媒を回収して２次側冷媒回路（Ｂ）で冷媒を循環させる。

情報としての用途のみ
ＰＣＴに基づいて公開される国際出願をパンフレット第一頁にＰＣＴ加盟国を同定するために使用されるコード

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明　細　書

熱搬送装置

[技術分野]

本発明は、熱搬送装置に係り、例えば、空気調和装置の冷媒回路などとして利用可能な熱搬送装置であって、特に、ポンプ等の駆動源を必要とすることなしに熱搬送媒体を循環させて熱搬送を行うようにした熱搬送装置に関する。

[背景技術]

従来より、空気調和装置の冷媒回路には、例えば、特開昭６２－２３８９５１号公報に開示されているように、２系統の冷媒回路を備えたものが知られている。この種の冷媒回路は、圧縮機、第１熱源側熱交換器、減圧機構及び第１利用側熱交換器が冷媒配管によって順に接続されて成る１次側冷媒回路と、ポンプ、第２熱源側熱交換器及び第２利用側熱交換器が冷媒配管によって順に接続されて成る２次側冷媒回路とを備えている。

そして、１次側冷媒回路の第１利用側熱交換器と２次側冷媒回路の第２熱源側熱交換器との間で熱交換を行うと共に、第２利用側熱交換器は、空調する室内に配置されている。

この冷媒回路において、室内の冷房運転時には、第１利用側熱交換器で冷媒が蒸発し、第２熱源側熱交換器で冷媒が凝縮する。この凝縮冷媒は、第２利用側熱交換器において室内空気と熱交換して蒸発する。これにより室内を冷房する。

一方、室内の暖房運転時には、第１利用側熱交換器で冷媒が凝縮し、第２熱源側熱交換器で冷媒が蒸発する。この蒸発冷媒は、第２利用側熱交換器において室内空気と熱交換して凝縮する。これにより室内を暖房する。

このようにして、１次側冷媒回路の配管長の短縮化を図り、冷凍能力の向上を図っている。

ところが、このような構成では、２次側冷媒回路において冷媒を循環させるための個別の駆動源としてのポンプが必要であった。この結果、消費電力の増大等を招く

ことになる。また、この駆動源を要することによって故障が発生する要因箇所が増え、装置全体としての信頼性が劣るという不具合があった。

これらの課題を解消するための冷媒回路としては、２次側冷媒回路に駆動源を備えない、いわゆる無動力の熱搬送方式の熱搬送装置がある。この熱搬送装置には、特開昭６３－１８００２２号公報に開示されているものがある。この熱搬送装置における２次側冷媒回路は、加熱器と凝縮器と密閉容器とが冷媒配管によって順に接続されて構成され、密閉容器が加熱器よりも高い位置に配置されている。更に、加熱器と密閉容器とは、開閉弁を備えた均圧管によって接続されている。

このような構成により、室内の暖房運転時には、先ず、開閉弁を閉状態にし、加熱器で加熱されたガス冷媒を凝縮器で凝縮させて液化した後、この液冷媒を密閉容器に回収する。その後、開閉弁を開口して均圧管により加熱器と密閉容器とを均圧状態にし、加熱器よりも高い位置にある密閉容器から液冷媒を加熱器に戻している。

この動作を繰り返し、２次側冷媒回路にポンプ等の駆動源を設けることなしに冷媒の循環を可能としている。

－解決課題－

しかしながら、このような熱搬送装置では、凝縮器から密閉容器にガス冷媒が流入した場合、この密閉容器内の圧力が上昇し、良好な冷媒の循環動作が行えなくなる虞れがある。このため、凝縮器からガス冷媒が流出しないように、該凝縮器において冷媒を過冷却状態にしておく必要があった。

また、上記熱搬送装置は、密閉容器内の構造を改良することにより、密閉容器内の圧力上昇を抑制するようにしているが、十分な信頼性が得られているとは言えないものであった。

また、このように密閉容器に液冷媒を確実に導入させるためには、凝縮器を密閉容器よりも高い位置に配置しておく必要があり、各機器の配設位置の制約が多く、大規模なシステムや長配管システムに対して適用することは難しかった。

本発明は、この点に鑑みて成されたものであって、駆動源を必要としない無動力

の熱搬送方式の熱搬送装置において、機器の配設位置の制約を小さくすると共に、高い信頼性及び汎用性を得ることを目的とする。

[発明の開示]

上記の目的を達成するために、本発明は、利用側冷媒回路の冷媒に圧力を与え、この圧力を利用して該利用側冷媒回路で冷媒を循環させる。また、利用側熱交換手段の所定動作が行われるように冷媒循環方向を規制する。

具体的に、本発明が講じた第１の解決手段は、図１に示すように、先ず、熱源側熱交換手段（１）と利用側熱交換手段（３）とがガス配管（６）及び液配管（７）により冷媒循環が可能に接続されて成り、上記熱源側熱交換手段（１）が熱源手段（Ａ）との間で熱交換を行う冷媒回路（Ｂ）を備えている。

そして、上記液配管（７）に連通し、液冷媒を貯留するタンク手段（Ｔ）を備えている。更に、該タンク手段（Ｔ）の内部圧力を上昇させる加圧動作と、内部圧力を下降させる減圧動作とを交互に行う圧力調節手段（18）を備えている。

加えて、該圧力調節手段（18）の加圧動作時には上記タンク手段（Ｔ）から蒸発器となる熱交換手段への液冷媒の供給のみを許容する一方、減圧動作時には凝縮器となる熱交換手段からタンク手段（Ｔ）への液冷媒の回収のみを許容し、冷媒回路（Ｂ）の冷媒を循環させて利用側熱交換手段（３）に吸熱又は放熱を行わせる冷媒制御手段（Ｈ）を備えている。

この第１の解決手段では、圧力調節手段（18）の加圧動作時に、タンク手段（Ｔ）から蒸発器となる熱交換手段へ液冷媒が供給される。一方、圧力調節手段（18）の減圧動作時には、凝縮器となる熱交換手段からタンク手段（Ｔ）へ液冷媒が回収される。これにより、熱源側熱交換手段（１）と利用側熱交換手段（３）との間で所定方向の冷媒循環が行われて、利用側熱交換手段（３）の吸熱若しくは放熱が行われる。

このように、圧力調節手段（18）からタンク手段（Ｔ）に作用する圧力により冷媒を循環させる。また、凝縮器となる熱交換手段からタンク手段（Ｔ）への液冷媒の回収は、タンク手段（Ｔ）内に発生する低圧によって行われる。

したがって、この第１の解決手段によれば、液配管（７）に接続されたタンク手

段（T）の内部を高圧状態と低圧状態とに交互に切換え、この圧力を利用して熱源側熱交換手段（1）と利用側熱交換手段（3）との間で冷媒を循環させるようにしたために、冷媒を循環するための冷媒循環用ポンプ等の特別な搬送手段が必要なくなる。この結果、消費電力の低減、故障が発生する要因箇所の削減、装置全体としての信頼性の確保を図ることができる。

また、凝縮器となる熱交換手段の液冷媒をタンク手段（T）の低圧によって吸引するので、タンク手段（T）を熱交換手段よりも下方に配置するといった従来のような機器の配設位置の制約がなくなり、実用性の向上を得ることができる。

また、冷媒回路（B）における冷媒循環の動作が安定的に行われるので、この回路全体を大型にしても冷媒循環が良好に行われることになり、システムの拡大化が可能になる。

本発明が講じた第2の解決手段は、図1に示すように、上記第1の解決手段において、先ず、利用側熱交換手段（3）は吸熱を行う蒸発器としたものである。冷媒制御手段（H）は、圧力調節手段（18）の加圧動作時、タンク手段（T）から利用側熱交換手段（3）への液冷媒の供給を許容すると共に、タンク手段（T）から熱源側熱交換手段（1）への液冷媒の供給を阻止する一方、圧力調節手段（18）の減圧動作時、熱源側熱交換手段（1）からタンク手段（T）への液冷媒の回収を許容すると共に、利用側熱交換手段（3）からタンク手段（T）への液冷媒の回収を阻止する。

この第2の解決手段では、利用側熱交換手段（3）の吸熱運転時、タンク手段（T）から利用側熱交換手段（3）へ液冷媒が供給され、この液冷媒が利用側熱交換手段（3）で蒸発する。そして、このガス冷媒は熱源側熱交換手段（1）で凝縮した後、タンク手段（T）へ回収されることになる。このため、利用側熱交換手段（3）で蒸発する冷媒により吸熱動作が得られる。

したがって、この第2の解決手段によれば、圧力調節手段（18）の加圧動作時にはタンク手段（T）から利用側熱交換手段（3）への液冷媒の供給のみを許容し、圧力調節手段（18）の減圧動作時には熱源側熱交換手段（1）からタンク手段（T）への液冷媒の回収のみを許容することにより、利用側熱交換手段（3）の吸熱運転を行わせるようにしたために、利用側熱交換手段（3）の吸熱運転を確実に行うことがで

き、装置の信頼性の向上を図ることができる。

本発明が講じた第３の解決手段は、図７に示すように、上記第１の解決手段において、先ず、利用側熱交換手段（３）は放熱を行う凝縮器としたものである。冷媒制御手段（Ｈ）は、圧力調節手段（18）の加圧動作時、タンク手段（Ｔ）から熱源側熱交換手段（１）への液冷媒の供給を許容すると共に、タンク手段（Ｔ）から利用側熱交換手段（３）への液冷媒の供給を阻止する一方、圧力調節手段（18）の減圧動作時、利用側熱交換手段（３）からタンク手段（Ｔ）への液冷媒の回収を許容すると共に、熱源側熱交換手段（１）からタンク手段（Ｔ）への液冷媒の回収を阻止する。

この第３の解決手段では、利用側熱交換手段（３）の放熱運転時、タンク手段（Ｔ）から熱源側熱交換手段（１）へ液冷媒が供給され、熱源側熱交換手段（１）では、この液冷媒が蒸発する。そして、このガス冷媒は利用側熱交換手段（３）で凝縮した後、タンク手段（Ｔ）へ回収されることになる。このため、利用側熱交換手段（３）で凝縮する冷媒により放熱動作が得られる。

したがって、この第３の解決手段によれば、圧力調節手段（18）の加圧動作時にはタンク手段（Ｔ）から熱源側熱交換手段（１）への液冷媒の供給のみを許容し、圧力調節手段（18）の減圧動作時には利用側熱交換手段（３）からタンク手段（Ｔ）への液冷媒の回収のみを許容することにより、利用側熱交換手段（３）の放熱運転を行わせるようにしたために、利用側熱交換手段（３）の放熱運転を確実に行うことができ、装置の信頼性の向上を図ることができる。

本発明が講じた第４の解決手段は、図４及び図５に示すように、上記第１、第２又は第３の解決手段において、圧力調節手段（18）は、加圧動作時、タンク手段（Ｔ）の冷媒に熱を与えて該タンク手段（Ｔ）の内圧を上昇させる一方、減圧動作時、タンク手段（Ｔ）の冷媒から熱を奪って該タンク手段（Ｔ）の内圧を下降させる構成としている。

この第４の解決手段では、このようにタンク手段（Ｔ）の冷媒を直接的に加熱及び冷却してタンク手段（Ｔ）の内圧を変化させて冷媒の搬送を行う。

したがって、この第４の解決手段によれば、タンク手段（Ｔ）の冷媒を圧力調節

手段（18）によって直接的に加熱及び冷却してタンク手段（T）の内圧を変化させるようにしたために、比較的熱ロスの小さい機構でもってタンク手段（T）の内圧を変化させることができ、冷媒搬送の高効率化を図ることができる。

本発明が講じた第５の解決手段は、図４に示すように、上記第４の解決手段において、圧力調節手段（18）は、タンク手段（T）に隣接され、タンク手段（T）の冷媒に熱を与える加熱動作と、タンク手段（T）の冷媒から熱を奪う冷却動作とを切換えて行う熱交換手段（１８b）で構成したものである。

また、本発明が講じた第６の解決手段は、図５に示すように、上記第４の解決手段において、圧力調節手段（18）は、圧縮機（D1）と第１熱交換器（D3）と減圧機構（D4）と第２熱交換器（D5）と圧縮機（D1）の吐出側に対する第１熱交換器（D3）及び第２熱交換器（D5）の接続状態を交互に切換える切換え手段（D2）とを有する冷媒回路（D）を備えたものである。そして、上記第１熱交換器（D3）が、タンク手段（T）との間で熱交換し、切換え手段（D2）の切換え動作に伴ってタンク手段（T）の冷媒を加熱及び冷却する。

この第５及び第６の解決手段では、圧力調節手段の構成を具体的に得ることができることになる。

したがって、この第５及び第６の解決手段によれば、圧力調節手段の構成を具体化したことにより、装置自体の実用性の向上を図ることができ、また、タンク手段（T）の内圧の調整を正確に行え、運転動作の信頼性の向上を図ることもできる。

本発明が講じた第７の解決手段は、図２、図３及び図６に示すように、上記第１、第２又は第３の解決手段において、圧力調節手段（18）は、タンク手段（T）に圧力管（19）を介して連結された圧力発生手段（18a, 18c, 19c）を備え、加圧動作時、圧力発生手段（18a, 18c, 19c）からタンク手段（T）の内部に高圧を作用させる一方、減圧動作時、圧力発生手段（18a, 18c, 19c）からタンク手段（T）の内部に低圧を作用させる構成としている。

したがって、この第７の解決手段によれば、タンク手段（T）に作用させる圧力の発生源である圧力発生手段（18a, 18c, 19c）を圧力管（19）によりタンク手段（T）

に連結したために、この圧力発生源をタンク手段（T）に近接して配置する必要がなくなり、配設位置の自由度の向上を図ることができる。

本発明が講じた第８の解決手段は、図２に示すように、上記第７の解決手段において、先ず、圧力発生手段は、液冷媒が貯留可能な貯留容器（１８a）としたものである。圧力調節手段（18）は、加圧動作時、貯留容器（１８a）の液冷媒に熱を与えて該液冷媒を蒸発させ、該貯留容器（１８a）の内圧を上昇させる一方、減圧動作時、貯留容器（１８a）のガス冷媒から熱を奪って該ガス冷媒を凝縮させ、該貯留容器（１８a）の内圧を下降させる。

また、本発明が講じた第９の解決手段は、図３に示すように、上記第７の解決手段において、先ず、圧力発生手段は、圧縮機（１８c）としたものである。圧力管（19）の圧縮機（１８c）に対する接続状態が、切換え手段（Ｉ）によって圧縮機（１８c）の吐出側と吸入側とに切換わる。更に、圧力調節手段（18）は、切換え手段（Ｉ）の切換え動作により、加圧動作時に圧力管（19）を圧縮機（１８c）の吐出側に接続する一方、減圧動作時に圧力管（19）を圧縮機（１８c）の吸入側に接続する。

また、本発明が講じた第１０の解決手段は、図６に示すように、上記第７の解決手段において、先ず、圧力発生手段は、冷媒が貯留可能な熱交換器（１９c）としたものである。圧力調節手段（18）は、加圧動作時に熱交換器（１９c）の冷媒に熱を与えて該熱交換器（１９c）の内圧を上昇させる一方、減圧動作時に熱交換器（１９c）の冷媒から熱を奪って該熱交換器（１９c）の内圧を下降させる。

また、本発明が講じた第１１の解決手段は、図６に示すように、上記第１０の解決手段において、圧力調節手段（18）は、圧縮機（D1）と第１熱交換器（D3）と減圧機構（D4）と第２熱交換器（D5）と圧縮機（D1）の吐出側に対する第１熱交換器（D3）及び第２熱交換器（D5）の接続状態を交互に切換える切換え手段（D2）とを有する冷媒回路（Ｄ）を備えたものである。そして、上記第１熱交換器（D3）が、熱交換器（１９c）との間で熱交換し、切換え手段（D2）の切換え動作に伴って熱交換器（１９c）の冷媒を加熱及び冷却する。

この第８～第１１の解決手段では、タンク手段（Ｔ）に対して加圧動作及び減圧動作を行うための圧力発生源となる圧力発生手段（18a, 18c, 19c）の構成を具体的に

得ることができる。

したがって、この第８～第１１の解決手段によれば、圧力発生手段（18a, 18c, 19c）の構成を具体化することができるので、装置自体の実用性の向上を図ることができる。

本発明が講じた第１２の解決手段は、図８に示すように、上記第２の解決手段において、冷媒制御手段（H）は、液配管（7）におけるタンク手段（T）の接続位置と熱源側熱交換手段（1）との間に設けられ、圧力調節手段（18）の加圧動作時に閉鎖し、減圧動作時に開口する第１の電磁弁（SV-A）と、液配管（7）におけるタンク手段（T）の接続位置と利用側熱交換手段（3）との間に設けられ、圧力調節手段（18）の加圧動作時に開口し、減圧動作時に閉鎖する第２の電磁弁（SV-B）とによって構成されたものである。

また、本発明が講じた第１３の解決手段は、図８に示すように、上記第３の解決手段において、冷媒制御手段（H）は、液配管（7）におけるタンク手段（T）の接続位置と熱源側熱交換手段（1）との間に設けられ、圧力調節手段（18）の加圧動作時に開口し、減圧動作時に閉鎖する第１の電磁弁（SV-A）と、液配管（7）におけるタンク手段（T）の接続位置と利用側熱交換手段（3）との間に設けられ、圧力調節手段（18）の加圧動作時に閉鎖し、減圧動作時に開口する第２の電磁弁（SV-B）とによって構成されたものである。

また、本発明が講じた第１４の解決手段は、図１に示すように、上記第２の解決手段において、冷媒制御手段（H）は、液配管（7）におけるタンク手段（T）の接続位置と熱源側熱交換手段（1）との間に設けられ、熱源側熱交換手段（1）からタンク手段（T）への液冷媒の流通のみを許容する第１の逆止弁（CV1）と、液配管（7）におけるタンク手段（T）の接続位置と利用側熱交換手段（3）との間に設けられ、タンク手段（T）から利用側熱交換手段（3）への液冷媒の流通のみを許容する第２の逆止弁（CV2）とによって構成されたものである。

また、本発明が講じた第１５の解決手段は、図７に示すように、上記第３の解決手段において、冷媒制御手段（H）は、液配管（7）におけるタンク手段（T）の接続位置と熱源側熱交換手段（1）との間に設けられ、タンク手段（T）から熱源側熱

交換手段（１）への液冷媒の流通のみを許容する第１の逆止弁（ＣＶ３）と、液配管（７）におけるタンク手段（Ｔ）の接続位置と利用側熱交換手段（３）との間に設けられ、利用側熱交換手段（３）からタンク手段（Ｔ）への液冷媒の流通のみを許容する第２の逆止弁（ＣＶ４）とによって構成されたものである。

この第１２～第１５の解決手段では、冷媒制御手段（Ｈ）の構成を具体的に得ることができる。

したがって、この第１２～第１５の解決手段によれば、冷媒流通制御手段（Ｈ）の構成を具体的に得ることができるので、利用側熱交換手段（３）に吸熱若しくは放熱運転を行わせるための冷媒循環方向の設定を正確に行うことができ、これによっても運転動作の信頼性の向上及び実用性の向上を図ることができる。

本発明が講じた第１６の解決手段は、複数のタンク手段（T1, T2）を設けることにより、利用側熱交換手段（３）の放熱運転若しくは吸熱運転が連続的に行えるようにした。

具体的には、図１０及び図１２などに示すように、先ず、熱源側熱交換手段（１）と利用側熱交換手段（３）とがガス配管（６）及び液配管（７）により冷媒循環が可能に接続されて成り、上記熱源側熱交換手段（１）が熱源手段（Ａ）との間で熱交換を行う冷媒回路（Ｂ）を備えている。

そして、上記液配管（７）に互いに並列に接続され、液冷媒を貯留する１以上の第１タンク手段（T1）及び１以上の第２タンク手段（T2）を備えている。

更に、該第１タンク手段（T1）の内部圧力を上昇させると共に第２タンク手段（T2）の内部圧力を下降させる第１圧力状態と、第１タンク手段（T1）の内部圧力を下降させると共に第２タンク手段（T2）の内部圧力を上昇させる第２圧力状態とに交互に切換える圧力調節手段（18）を備えている。

加えて、該圧力調節手段（18）の第１圧力状態時には、第１タンク手段（T1）から蒸発器となる熱交換手段に液冷媒を供給すると共に、凝縮器となる熱交換手段から第２タンク手段（T2）に液冷媒を回収する一方、第２圧力状態時には、第２タンク手段（T2）から蒸発器となる熱交換手段に液冷媒を供給すると共に、凝縮器となる熱交換手段から第１タンク手段（T1）に液冷媒を回収し、冷媒回路（Ｂ）の冷媒を循環さ

せて利用側熱交換手段（３）に吸熱又は放熱を連続して行わせる冷媒制御手段（Ｈ）を備えている。

この第１６の解決手段では、圧力調節手段（18）の第１圧力状態と第２圧力状態とを交互に切換えながら、冷媒制御手段（Ｈ）による冷媒の流通を阻止することで、一方の熱交換手段に液冷媒を供給するタンク手段と、他方の熱交換手段から冷媒を回収するタンク手段とが交互に切換えられることになる。このため、利用側熱交換手段（３）の吸熱運転若しくは放熱運転が連続的に行われる。

したがって、この第１６の解決手段によれば、第１タンク手段（T1）に高圧を作用させ且つ第２タンク手段（T2）に低圧を作用させる動作と、第２タンク手段（T2）に高圧を作用させ且つ第１タンク手段（T1）に低圧を作用させる動作とを交互に行うようにしたために、利用側熱交換手段（３）の吸熱運転若しくは放熱運転を連続的に行うことができるので、装置全体の性能及び実用性の向上を図ることができる。

また、本解決手段においても、上述した第１の解決手段と同様に、熱源側熱交換手段（１）と利用側熱交換手段（３）との間で冷媒を循環するための特別な搬送手段が必要ないため、消費電力の低減、故障が発生する要因箇所の削減、装置全体としての信頼性の確保を図ることができる。

また、機器の配設位置の制約が小さくすることができ、汎用性の向上を図ることができる。

本発明が講じた第１７の解決手段は、図１２に示すように、上記第１６の解決手段において、先ず、利用側熱交換手段（３）は吸熱を行う蒸発器としたものである。そして、冷媒制御手段（Ｈ）は、圧力調節手段（18）の第１圧力状態時、第１タンク手段（T1）から利用側熱交換手段（３）へ液冷媒を供給すると共に、熱源側熱交換手段（１）から第２タンク手段（T2）へ液冷媒を回収する一方、圧力調節手段（18）の第２圧力状態時、第２タンク手段（T2）から利用側熱交換手段（３）へ液冷媒を供給すると共に、熱源側熱交換手段（１）から第１タンク手段（T1）へ液冷媒を回収するように液配管（７）における冷媒流通状態を切換える。

この第１７の解決手段では、第１タンク手段（T1）から利用側熱交換手段（３）に液冷媒を供給しながら熱源側熱交換手段（１）から第２タンク手段（T2）に液冷媒

を回収する状態と、第２タンク手段（T2）から利用側熱交換手段（３）に液冷媒を供給しながら熱源側熱交換手段（１）から第１タンク手段（T1）に液冷媒を回収する状態とが交互に行われる。この結果、利用側熱交換手段（３）の吸熱運転が連続的に行われる。

また、本発明が講じた第１８の解決手段は、図１３に示すように、上記第１６の解決手段において、先ず、利用側熱交換手段（３）は放熱を行う凝縮器としたものである。そして、冷媒制御手段（H）は、圧力調節手段（18）の第１圧力状態時、第１タンク手段（T1）から熱源側熱交換手段（１）へ液冷媒を供給すると共に、利用側熱交換手段（３）から第２タンク手段（T2）へ液冷媒を回収する一方、圧力調節手段（18）の第２圧力状態時、第２タンク手段（T2）から熱源側熱交換手段（１）へ液冷媒を供給すると共に、利用側熱交換手段（３）から第１タンク手段（T1）へ液冷媒を回収するように液配管（７）における冷媒流通状態を切換える。

この第１８の解決手段では、第１タンク手段（T1）から熱源側熱交換手段（１）に液冷媒を供給しながら利用側熱交換手段（３）から第２タンク手段（T2）に液冷媒を回収する状態と、第２タンク手段（T2）から熱源側熱交換手段（１）に液冷媒を供給しながら利用側熱交換手段（３）から第１タンク手段（T1）に液冷媒を回収する状態とが交互に行われる。この結果、利用側熱交換手段（３）の放熱運転が連続的に行われる。

したがって、この第１７の解決手段によれば、各タンク手段（T1, T2）に対する圧力作用状態を交互に切換えることで、利用側熱交換手段（３）の吸熱運転を連続的に行うことができる。また、第１８の解決手段によれば、同様にして各タンク手段（T1, T2）に対する圧力作用状態を交互に切換えることで、利用側熱交換手段（３）の放熱運転を連続的に行うことができる。このため、例えば、利用側熱交換手段（３）を室内に設置し、室内を冷房或いは暖房して空気調和を行う場合、室内の空気調和状態を常に良好に維持することができる。

また、本発明が講じた第１９の解決手段は、図１０～図１２に示すように、上記第１６、第１７又は第１８の解決手段において、圧力調節手段（18）は、各タンク手段（T1, T2）に圧力管（19d, 19e）を介して連結された圧力発生手段（18A, 18B, D1, E

1, E2）を備え、該圧力発生手段（18A, 18B, D1, E1, E2）により、第１圧力状態時、第１タンク手段（T1）の内部に高圧を作用させ、第２タンク手段（T2）の内部に低圧を作用させる一方、第２圧力状態時、第２タンク手段（T2）の内部に高圧を作用させ、第１タンク手段（T1）の内部に低圧を作用させる構成としたものである。

したがって、この第１９の解決手段によれば、利用側熱交換手段（３）の吸熱運転若しくは放熱運転を連続的に行わせるものにおいても、上述した第７の解決手段と同様に、各タンク手段（T1, T2）に作用させる圧力の発生源を各タンク手段（T1, T2）に近接して配置する必要がなくなり、配設位置の自由度の向上を図ることができる。

本発明が講じた第２０の解決手段は、図１０に示すように、上記第１９の解決手段において、先ず、圧力発生手段を、第１タンク手段（T1）に接続されて液冷媒が貯留可能な第１の貯留容器（１８A）及び第２タンク手段（T2）に接続されて液冷媒が貯留可能な第２の貯留容器（１８B）で構成したものである。

そして、圧力調節手段（18）は、第１圧力状態時、第１の貯留容器（１８A）の液冷媒に熱を与えて該液冷媒を蒸発させ、該貯留容器（１８A）の内圧を上昇させると共に、第２の貯留容器（１８B）のガス冷媒から熱を奪って該ガス冷媒を凝縮させ、該貯留容器（１８B）の内圧を下降させる一方、第２圧力状態時、第２の貯留容器（１８B）の液冷媒に熱を与えて該液冷媒を蒸発させ、該貯留容器（１８B）の内圧を上昇させると共に、第１の貯留容器（１８A）のガス冷媒から熱を奪って凝該ガス冷媒を縮させ、該貯留容器（１８A）の内圧を下降させる。

また、本発明が講じた第２１の解決手段は、図１１に示すように、上記第１９の解決手段において、先ず、圧力発生手段を圧縮機（D1）で構成したものである。そして、第１タンク手段（T1）及び第２タンク手段（T2）の圧縮機（D1）に対する接続状態の切換えが、切換え手段（Ｉ）によって圧力管（19d, 19e）を圧縮機（D1）の吐出側と吸入側とに切換えて行う。

加えて、圧力調節手段（18）は、第１圧力状態時、圧縮機（D1）の吐出側を第１タンク手段（T1）に接続し、圧縮機（D1）の吸入側を第２タンク手段（T2）に接続する一方、第２圧力状態時、圧縮機（D1）の吐出側を第２タンク手段（T2）に接続し、圧縮機（D1）の吸入側を第１タンク手段（T1）に接続する。

また、本発明が講じた第２２の解決手段は、図１２に示すように、上記第１９の解決手段において、先ず、圧力発生手段を、第１タンク手段（T1）に接続されて冷媒が貯留可能な第１の熱交換器（E1）及び第２タンク手段（T2）に接続されて冷媒が貯留可能な第２の熱交換器（E2）で構成したものである。

そして、圧力調節手段（18）は、第１圧力状態時、第１の熱交換器（E1）の冷媒に熱を与えて該熱交換器（E1）の内圧を上昇させると共に、第２の熱交換器（E2）の冷媒から熱を奪って該熱交換器（E2）の内圧を下降させる一方、第２圧力状態時、第２の熱交換器（E2）の冷媒に熱を与えて該熱交換器（E2）の内圧を上昇させると共に、第１の熱交換器（E1）の冷媒から熱を奪って該熱交換器（E1）の内圧を下降させる。

また、本発明が講じた第２３の解決手段は、図１２に示すように、上記第２２の解決手段において、先ず、圧力調節手段（18）は、圧縮機（D1）と第１熱交換器（D3）と減圧機構（D4）と第２熱交換器（D5）と圧縮機（D1）の吐出側に対する第１熱交換器（D3）及び第２熱交換器（D5）の接続状態を交互に切換える切換え手段（Ｉ）とを有する冷媒回路（Ｄ）を備えている。

そして、第１熱交換器（D3）が第１タンク手段（T1）に接続された第１の熱交換器（E1）との間で熱交換し、第２熱交換器（D5）が第２タンク手段（T2）に接続された第２の熱交換器（E2）との間で熱交換し、切換え手段（Ｉ）の切換え動作に伴って第１圧力状態と第２圧力状態とに切換わる。

また、本発明が講じた第２４の解決手段は、図１６に示すように、上記第１９の解決手段において、先ず、圧力発生手段を、加熱用熱交換器（D3）によって加熱される加圧用熱交換器（E2）と、冷却用熱交換器（D5）によって冷却される減圧用熱交換器（E1）とによって構成したものである。

そして、圧力調節手段（18）は、第１圧力状態時、加圧用熱交換器（E2）を第１タンク手段（T1）に接続すると共に、減圧用熱交換器（E1）を第２タンク手段（T2）に接続する一方、第２圧力状態時、加圧用熱交換器（E2）を第２タンク手段（T2）に接続すると共に、減圧用熱交換器（E1）を第１タンク手段（T1）に接続する。

また、本発明が講じた第２５の解決手段は、図１６に示すように、上記第２４の解決手段において、圧力調節手段（18）は、圧縮機（D1）と加熱用熱交換器（D3）と減圧機構（D4）と冷却用熱交換器（D5）とが冷媒配管によって順に接続されて成る冷

媒回路（D）を備えたものである。

したがって、この第２０～第２５の解決手段によれば、上述した第１９の解決手段の効果を発揮させることができる圧力発生手段の具体構成を得ることができる。この結果、実用性の更なる向上を図ることができる。

本発明が講じた第２６の解決手段は、図２２に示すように、上記第１の解決手段において、圧力調節手段（18）は、タンク手段（T）の内部圧力を上昇させて該タンク手段（T）の液冷媒を液配管（７）に押出す加圧動作を行う加圧手段（50）と、タンク手段（T）の内部圧力を下降させて液配管（７）から該タンク手段（T）へ液冷媒を回収する減圧動作を行う減圧手段（60）とを備えている。

そして、上記減圧手段（60）は、タンク手段（T）に接続され且つ冷媒を凝縮することによってタンク手段（T）の内部圧力を下降させる循環用凝縮器（61）を備え、該循環用凝縮器（61）の凝縮圧力が、凝縮器となる熱交換手段の凝縮圧力より低く設定されている。

この第２６の解決手段では、循環用凝縮器（61）での冷媒の凝縮により、タンク手段（T）の内部が低圧になる。そして、この圧力は凝縮器となる熱交換手段の凝縮圧力より低くなっているので、この凝縮器内の液冷媒はタンク手段（T）に吸引されることになる。

したがって、この第２６の解決手段によれば、タンク手段（T）に接続された減圧手段（60）の循環用凝縮器（61）での冷媒の凝縮によって、凝縮器からタンク手段（T）に液冷媒を回収するための低圧を発生させるようにしたために、この場合にも、タンク手段（T）を凝縮器よりも低い位置に配置するといった制約を受けることがなく、各機器を設置することができる。

また、減圧手段（60）とタンク手段（T）との連通状態と非連通状態とを切換えるのみでタンク手段（T）の内圧を変化させることが可能となるので、減圧手段（60）及びタンク手段（60）の閉回路に電磁弁などを採用するようにすれば、これを開閉するのみで冷媒の循環動作を行わせることができ、高い信頼性と故障が発生する要因箇所の低減を図ることができる。

本発明が講じた第２７の解決手段は、図２３に示すように、上記第１の解決手段において、先ず、圧力調節手段（18）は、タンク手段（Ｔ）の内部圧力を上昇させて該タンク手段（Ｔ）の液冷媒を液配管（７）に押出す加圧動作を行う加圧手段（50）と、タンク手段（Ｔ）の内部圧力を下降させて液配管（７）から該タンク手段（Ｔ）へ液冷媒を回収する減圧動作を行う減圧手段（60）とを備えている。

そして、上記加圧手段（50）は、タンク手段（Ｔ）に接続され且つ冷媒を蒸発することによってタンク手段（Ｔ）の内部圧力を上昇させる循環用蒸発器（51）を備え、該循環用蒸発器（51）の蒸発圧力が、蒸発器となる熱交換手段の蒸発圧力より高く設定されている。

この第２７の解決手段では、循環用蒸発器（51）での冷媒の蒸発により、タンク手段（Ｔ）の内部が高圧になる。そして、この圧力は蒸発器となる熱交換手段の蒸発圧力より高くなっているので、この蒸発器に対してタンク手段（Ｔ）から液冷媒が供給されることになる。

したがって、この第２７の解決手段によれば、タンク手段（Ｔ）に接続された加圧手段（50）の循環用蒸発器（51）での冷媒の蒸発によって、タンク手段（Ｔ）から蒸発器に液冷媒を供給するための高圧を発生させるようにしたために、タンク手段（Ｔ）と蒸発器との配設位置に制約を受けるといったことがなくなる。

また、本発明にあっても加圧手段（50）とタンク手段（Ｔ）との連通状態と非連通状態とを切換えるのみでタンク手段（Ｔ）の内圧を変化させることが可能となり、高い信頼性と故障が発生する要因箇所の低減を図ることができる。

本発明が講じた第２８の解決手段は、図２６に示すように、上記第２７の解決手段において、先ず、循環用蒸発器（51）の上方には補助タンク手段（ST）が設けられている。そして、減圧手段（60）による減圧動作時、補助タンク手段（ST）を減圧手段（60）及び液配管（７）にそれぞれ連通させて液配管（７）の液冷媒を補助タンク手段（ST）に回収する一方、加圧手段（50）による加圧動作時、補助タンク手段（ST）を加圧手段（50）に連通させて補助タンク手段（ST）の液冷媒を循環用蒸発器（51）に落下供給させる切換え手段（Ｉ）が設けられている。

この第２８の解決手段では、比較的小型の補助タンク手段（ST）を循環用蒸発器

（51）の上方に配置しておけば循環用蒸発器（51）への液冷媒の供給が可能となり、循環用蒸発器（51）の内部に液冷媒が無くなって冷媒の循環が不能になるとったことがなくなる。

したがって、この第２８の解決手段によれば、循環用蒸発器（51）の上方に補助タンク手段（ST）を設け、循環用蒸発器（51）に供給するための液冷媒を補助タンク手段（ST）に一時的に貯留させるようにしたために、タンク手段（T）と循環用蒸発器（51）との配設高さ位置に制約を受けることなく循環用蒸発器（51）に十分な液冷媒を供給することができる。

本発明が講じた第２９の解決手段は、図２７に示すように、上記第２７の解決手段において、先ず、循環用蒸発器（51）の上方には、１以上の第１補助タンク手段（ＳＴ1）及び１以上の第２補助タンク手段（ＳＴ2）設けられている。そして、第１補助タンク手段（ＳＴ1）を減圧手段（60）及び液配管（７）にそれぞれ連通させて液配管（７）の液冷媒を該第１補助タンク手段（ＳＴ1）に回収させると共に、第２補助タンク手段（ＳＴ2）を加圧手段（50）に連通させて該第２補助タンク手段（ＳＴ2）の液冷媒を循環用蒸発器（51）に落下供給させる第１切換え状態と、第２補助タンク手段（ＳＴ2）を減圧手段（60）及び液配管（７）にそれぞれ連通させて液配管（７）の液冷媒を該第２補助タンク手段（ＳＴ2）に回収させると共に、第１補助タンク手段（ST１）を加圧手段（50）に連通させて該第１補助タンク手段（ＳＴ1）の液冷媒を循環用蒸発器（51）に落下供給させる第２切換え状態とに切換える切換え手段（Ｉ）が設けられている。

この第２９の解決手段では、一方の補助タンク手段に対して液冷媒が回収され、他方の補助タンク手段からは循環用蒸発器（51）に液冷媒が供給される状態が同時に行われる。このため、個々の補助タンク手段（ST1，ST2）に対し液冷媒の回収動作と供給動作とを繰り返す頻度を低減できることになる。

したがって、この第２９の解決手段によれば、複数の補助タンク手段（ST1，ST2）を設け、一方の補助タンク手段に液冷媒を回収させ、他方の補助タンク手段から循環用蒸発器に液冷媒を供給させるようにしたために、補助タンク手段（ST1，ST2）の動作をタンク手段（Ｔ）に対する加圧減圧動作に同期させて切換える必要が無くなる。

このため、個々の補助タンク手段（ST1，ST2）に対し液冷媒回収動作と供給動作とを繰り返す頻度を低減できることになり、その長寿命化を図ることができる。

本発明が講じた第３０の解決手段は、図２４に示すように、上記第１６の解決手段において、先ず、圧力調節手段（18）は、第１タンク手段（T1）及び第２タンク手段（T2）に対し、一方のタンク手段の内部圧力を上昇させて該タンク手段の液冷媒を液配管（７）に押出す加圧動作を行う加圧手段（50）と、他方のタンク手段の内部圧力を下降させて液配管（７）から該タンク手段へ液冷媒を回収する減圧動作を行う減圧手段（60）とを備えている。

そして、上記減圧手段（60）は、各タンク手段（T1，T2）に接続され且つ冷媒を凝縮することによって各タンク手段（T1，T2）の内部圧力を下降させる循環用凝縮器（61）を備え、該循環用凝縮器（61）の凝縮圧力が、凝縮器となる熱交換手段の凝縮圧力より低く設定されている。

更に、上記圧力調節手段（18）は、第１圧力状態時に、第１タンク手段（T1）を加圧手段（50）により加圧すると共に第２タンク手段（T2）を減圧手段（60）により減圧する一方、第２圧力状態時に、第２タンク手段（T2）を加圧手段（50）により加圧すると共に第１タンク手段（T1）を減圧手段（60）により減圧する。

この第３０の解決手段では、利用側熱交換手段（３）の吸熱運転若しくは放熱運転を連続的に行うことができる冷媒回路において、各タンク手段（T1，T2）を凝縮器よりも低い位置に配置することがなく、この凝縮器からタンク手段（T1，T2）へ液冷媒が回収される。

したがって、この第３０の解決手段によれば、利用側熱交換手段（３）の吸熱運転若しくは放熱運転を連続的に行うことができる冷媒回路において、各タンク手段（T1，T2）を凝縮器よりも低い位置に配置するといった制約を受けることがなく、この凝縮器からタンク手段（T1，T2）へ液冷媒を回収することができ、冷媒を循環させることができる。

本発明が講じた第３１の解決手段は、図２５に示すように、上記第１６の解決手段において、先ず、圧力調節手段（18）は、第１タンク手段（T1）及び第２タンク手

段（T2）に対し、一方のタンク手段の内部圧力を上昇させて該タンク手段の液冷媒を液配管（７）に押出す加圧動作を行う加圧手段（50）と、他方のタンク手段の内部圧力を下降させて液配管（７）から該タンク手段へ液冷媒を回収する減圧動作を行う減圧手段（60）とを備えている。

そして、上記加圧手段（50）は、各タンク手段（T1，T2）に接続され且つ冷媒を蒸発することによって各タンク手段（T1，T2）の内部圧力を上昇させる循環用蒸発器（51）を備え、該循環用蒸発器（51）の蒸発圧力が、蒸発器となる熱交換手段の蒸発圧力より高く設定されている。

更に、上記圧力調節手段（18）は、第１圧力状態時に、第１タンク手段（T1）を加圧手段（50）により加圧すると共に第２タンク手段（T2）を減圧手段（60）により減圧する一方、第２圧力状態時に、第２タンク手段（T2）を加圧手段（50）により加圧すると共に第１タンク手段（T1）を減圧手段（60）により減圧する。

この第３１の解決手段では、利用側熱交換手段（３）の吸熱運転若しくは放熱運転を連続的に行うことができる冷媒回路において、各タンク手段（T1，T2）の蒸発器に対する配設位置に制約を受けることがなく、タンク手段（T1，T2）から蒸発器へ液冷媒が供給される。

したがって、この第３１の解決手段によれば、利用側熱交換手段（３）の吸熱運転若しくは放熱運転を連続的に行うことができる冷媒回路において、各タンク手段（T1，T2）の蒸発器に対する配設位置に制約を受けることがなく、タンク手段（T1，T2）から蒸発器への液冷媒の供給を可能にすることができる。

本発明が講じた第３２の解決手段は、図３０、図３３及び図３６などに示すように、上記第３１の解決手段において、先ず、循環用蒸発器（51）の上方には、１以上の第１補助タンク手段（ＳＴ１）及び１以上の第２補助タンク手段（ＳＴ２）が設けられている。

そして、該第１補助タンク手段（ＳＴ１）を減圧手段（60）及び液配管（７）にそれぞれ連通させて液配管（７）の液冷媒を該第１補助タンク手段（ＳＴ１）に回収させると共に、第２補助タンク手段（ＳＴ２）を加圧手段（50）に連通させて該第２補助タンク手段（ＳＴ２）の液冷媒を循環用蒸発器（51）に落下供給させる第１切換え状態と、

第２補助タンク手段（ＳＴ2）を減圧手段（60）及び液配管（７）にそれぞれ連通させて液配管（７）の液冷媒を該第２補助タンク手段（ＳＴ2）に回収させると共に、第１補助タンク手段（ＳＴ1）を加圧手段（50）に連通させて該第１補助タンク手段（ＳＴ1）の液冷媒を循環用蒸発器（51）に落下供給させる第２切換え状態とに切換える手段（Ｉ）が設けられている。

したがって、この第３２の解決手段によれば、複数の補助タンク（ST1, ST2）を設けたことにより、上述した第２９の解決手段の効果と同様の効果が発揮され、利用側熱交換手段（３）の吸熱運転若しくは放熱運転を連続的に行うことができる冷媒回路において、個々の補助タンク手段（ST1, ST2）に対し液冷媒回収動作と供給動作とを繰り返す頻度を低減できて、その長寿命化を図ることができる。

本発明が講じた第３３の解決手段は、図２２に示すように、上記第２６又は第３０の解決手段において、先ず、熱源手段（Ａ）は、熱源側熱交換手段（１）との間で熱交換を行う第１の熱交換手段（12）と、循環用凝縮器（61）との間で熱交換を行う第２の熱交換手段（72）とを備えている。

そして、利用側熱交換手段（３）の吸熱運転時における第１の熱交換手段（12）と第２の熱交換手段（72）との蒸発温度は同一である一方、上記第２の熱交換手段（72）を流れる冷媒の流量に対する循環用凝縮器（61）の容量の比は、第１の熱交換手段（12）を流れる冷媒の流量に対する熱源側熱交換手段（１）の容量の比よりも大きく設定されている。

この第３３の解決手段では、循環用凝縮器（61）の凝縮圧力を、凝縮器となる熱源側熱交換手段（１）の凝縮圧力より低く設定するための構成が具体的に得られることになる。

また、本発明が講じた第３４の解決手段は、図２３に示すように、上記第２７又は第３１の解決手段において、先ず、熱源手段（Ａ）は、熱源側熱交換手段（１）との間で熱交換を行う第１の熱交換手段（12）と、循環用蒸発器（51）との間で熱交換を行う第２の熱交換手段（71）とを備えている。

そして、利用側熱交換手段（３）の放熱運転時における第１の熱交換手段（12）と第２の熱交換手段（71）の凝縮温度は同一である一方、上記第２の熱交換手段（71）

を流れる冷媒の流量に対する循環用蒸発器（51）の容量の比は、第１の熱交換手段（12）を流れる冷媒の流量に対する熱源側熱交換手段（１）の容量の比よりも大きく設定されている。

この第３４の解決手段では、循環用蒸発器（51）の蒸発圧力を、蒸発器となる熱源側熱交換手段（1 ）の蒸発圧力より高く設定するための構成が具体的に得られることになる。

したがって、この第３３の解決手段によれば、循環用凝縮器（61）の凝縮圧力を、凝縮器となる熱源側熱交換手段の凝縮圧力より低く設定するための具体構成が、また、第３４の解決手段によれば、循環用蒸発器（51）の蒸発圧力を、蒸発器となる熱源側熱交換手段の蒸発圧力より高く設定するための具体構成がそれぞれ得られることになる。このため、タンク手段（Ｔ）に所定の圧力を確実に作用させることができ、装置の信頼性の向上を図ることができる。

本発明が講じた第３５の解決手段は、図２３に示すように、上記第２６又は第３０の解決手段において、先ず、減圧手段（60）は、タンク手段（Ｔ）の上端部と循環用凝縮器（61）のガス側とを接続するガス回収管（62）と、タンク手段（Ｔ）の下端部と循環用凝縮器（１）の液側とを接続する液供給管（63）とを備えている。　そして、上記液供給管（63）は、液配管（７）とは独立してタンク手段（Ｔ）の下端部に接続されている。

したがって、この第３５の解決手段によれば、減圧手段（60）の液供給管（63）を液配管（７）とは独立してタンク手段（Ｔ）の下端部に接続し、循環用凝縮器（61）及び凝縮器となる熱交換器をそれぞれ個別にタンク手段（Ｔ）に接続したために、各配管（63,7）にその配管径に応じた口径の逆止弁をそれぞれ配設することができ、特に、液供給管（63）においては、圧損を低く設定することが可能な逆止弁の適用が可能になり、この減圧手段（60）における冷媒の循環を円滑に行うことができる。

本発明が講じた第３６の解決手段は、図３０及び図３３に示すように、上記第１又は第１６の解決手段において、先ず、圧力調節手段（18）は、タンク手段（Ｔ）の内部圧力を上昇させて該タンク手段（Ｔ）の液冷媒を液配管（７）に押出す加圧動作

を行う加圧手段（50）と、タンク手段（T）の内部圧力を下降させて液配管（7）から該タンク手段（T）へ液冷媒を回収する減圧動作を行う減圧手段（60）とを備えている。

そして、上記減圧手段（60）は、タンク手段（T）に接続され且つ冷媒を凝縮することによってタンク手段（T）の内部圧力を下降させる循環用凝縮器（61）を備え、上記加圧手段（50）は、タンク手段（T）に接続され且つ冷媒を蒸発することによってタンク手段（T）の内部圧力を上昇させる循環用蒸発器（51）を備えている。

更に、熱源手段（A）は、圧縮機（11）、熱源側熱交換手段（1）の間で熱交換を行う第1の熱交換手段（12）と、循環用凝縮器（61）の間で熱交換を行う第2の熱交換手段（72）と、循環用蒸発器（51）の間で熱交換を行う第3の熱交換手段（71）とを備え、利用側熱交換手段（3）の放熱運転時、圧縮機（11）から吐出したガス冷媒を、第3の熱交換手段（71）において循環用蒸発器（51）との間で熱交換を行って顕熱変化させた後、第1の熱交換手段（12）において熱源側熱交換手段（1）との間で熱交換を行って凝縮させ、更に、第2の熱交換手段（72）において循環用凝縮器（61）との間で熱交換を行って蒸発させる。

また、本発明が講じた第37の解決手段は、図36に示すように、上記第1又は第16の解決手段において、先ず、圧力調節手段（18）は、タンク手段（T）の内部圧力を上昇させて該タンク手段（T）の液冷媒を液配管（7）に押出す加圧動作を行う加圧手段（50）と、タンク手段（T）の内部圧力を下降させて液配管（7）から該タンク手段（T）へ液冷媒を回収する減圧動作を行う減圧手段（60）とを備えている。

そして、上記減圧手段（60）は、タンク手段（T）に接続され且つ冷媒を凝縮することによってタンク手段（T）の内部圧力を下降させる循環用凝縮器（61）を備え、上記加圧手段（50）は、タンク手段（T）に接続され且つ冷媒を蒸発することによってタンク手段（T）の内部圧力を上昇させる循環用蒸発器（51）を備えている。

更に、熱源手段（A）は、圧縮機（11）、熱源側熱交換手段（1）の間で熱交換を行う第1の熱交換手段（12）と、循環用凝縮器（61）の間で熱交換を行う第2の熱交換手段（72）と、循環用蒸発器（51）の間で熱交換を行う第3の熱交換手段（71）とを備え、利用側熱交換手段（3）の放熱運転時、圧縮機（11）から吐出したガス冷媒を、第3の熱交換手段（71）及び第1の熱交換手段（12）に分流し、第3の熱交換

手段（71）において循環用蒸発器（51）との間で熱交換を行って凝縮させると共に、第１の熱交換手段（12）において熱源側熱交換手段（１）との間で熱交換を行って凝縮させ、その後、凝縮冷媒を、第２の熱交換手段（72）において循環用凝縮器（61）との間で熱交換を行って蒸発させる。

したがって、この第３６及び第３７の解決手段によれば、加圧手段（50）と減圧手段（60）とによってタンク手段（T）を加減圧して冷媒を循環させる回路に対して適用可能な熱源手段（A）となる冷媒回路を得ることができ、装置全体の構成の具体化を図ることができる。

［ 図面の簡単な説明 ］

図１は、第１実施形態における冷媒回路の全体構成を示す図である。

図２は、加減圧機構を示す図である。

図３は、加減圧機構の第１の変形例を示す図である。

図４は、加減圧機構の第２の変形例を示す図である。

図５は、加減圧機構の第３の変形例を示す図である。

図６は、加減圧機構の第４の変形例を示す図である。

図７は、第２実施形態における図１相当図である。

図８は、冷媒流通制御手段の変形例を示す図である。

図９は、第３実施形態における図１相当図である。

図１０は、第４実施形態における図２相当図である。

図１１は、第４実施形態の変形例を示す図３相当図である。

図１２は、第５実施形態における２次側冷媒回路を示す図である。

図１３は、第６実施形態における２次側冷媒回路を示す図である。

図１４は、第７実施形態における２次側冷媒回路を示す図である。

図１５は、第８実施形態における図１相当図である。

図１６は、第９実施形態における図１相当図である。

図１７は、第９実施形態における冷房運転動作を示す図である。

図１８は、第９実施形態における暖房運転動作を示す図である。

図１９は、第１０実施形態における図１相当図である。

図２０は、第１０実施形態における冷房運転動作を示す図である。

図２１は、第１０実施形態における暖房運転状態を示す図である。

図２２は、第１１実施形態における図１相当図である。

図２３は、第１２実施形態における図１相当図である。

図２４は、第１３実施形態における２次側冷媒回路を示す図である。

図２５は、第１４実施形態における２次側冷媒回路を示す図である。

図２６は、第１５実施形態における２次側冷媒回路を示す図である。

図２７は、第１６実施形態における２次側冷媒回路を示す図である。

図２８は、第１７実施形態における図１相当図である。

図２９は、第１８実施形態における２次側冷媒回路を示す図である。

図３０は、第１９実施形態における図１相当図である。

図３１は、第１９実施形態における冷房運転動作を示す図である。

図３２は、第１９実施形態における暖房運転状態を示す図である。

図３３は、第２０実施形態における図１相当図である。

図３４は、第２０実施形態における冷房運転動作を示す図である。

図３５は、第２０実施形態における暖房運転状態を示す図である。

図３６は、第２１実施形態における図１相当図である。

図３７は、第２１実施形態における冷房運転動作を示す図である。

図３８は、第２１実施形態における暖房運転状態を示す図である。

［ 発明を実施するための最良の形態 ］

次に、本発明の実施形態を図面に基いて説明する。本各実施形態は、１次側冷媒回路と２次側冷媒回路との２系統の冷媒回路を備えている。そして、本各実施形態は、１次側冷媒回路と２次側冷媒回路との間で熱交換を行って室内の空気調和を行う空気調和装置の冷媒回路に本発明を適用したものである。

－第１実施形態－

先ず、第１実施形態について図１を用いて説明する。

本実施形態は、冷房専用の空気調和装置を構成し、１次側冷媒回路（Ａ）が熱源

手段を構成している。そして、図１は、本実施形態に係る熱搬送装置全体の冷媒回路を示している。

先ず、室内の空気との間で熱交換を行って室内の冷房を行う２次側冷媒回路（Ｂ）について説明する。

この２次側冷媒回路（Ｂ）は、空調用室内に配置された利用側熱交換手段としての室内熱交換器（3，3）と、１次側冷媒回路（Ａ）との間で熱の授受を行う熱源側熱交換手段としての２次側熱源熱交換器（１）とがガス配管（６）及び液配管（７）によって接続され、冷媒の循環が可能な閉回路に構成されている。各室内熱交換器（3，3）は、互いに並列に接続され、液配管（７）には、各室内熱交換器（3，3）に対応して室内電動膨張弁（EV1，EV1）が設けられている。

本実施形態の特徴として、液配管（７）には液冷媒が貯留されたタンク（Ｔ）が接続されている。このタンク（Ｔ）は、その下端部が接続管（17）によって液配管（７）に接続されている。

この液配管（７）における接続管（17）の接続位置と２次側熱源熱交換器（１）との間には、該２次側熱源熱交換器（１）からタンク（Ｔ）への液冷媒の流通のみを許容する第１逆止弁（ＣＶ１）が設けられている。また、この液配管（７）における接続管（17）の接続位置と室内熱交換器（3，3）との間には、タンク（Ｔ）から室内熱交換器（3，3）への液冷媒の流通のみを許容する第２逆止弁（ＣＶ２）が設けられている。これら逆止弁（CV1，CV2）により冷媒制御手段（Ｈ）が構成されている。

上記タンク（Ｔ）の上端部には、圧力調節手段としての加減圧機構（18）が圧力管である加減圧管（19）によって接続されている。この加減圧機構（18）は、例えば、図２に示すように、液冷媒を貯留する圧力発生手段としての貯留容器（１８ａ）と、該貯留容器（１８ａ）を加熱又は冷却する熱交換手段としての熱交換ユニット（１８ｂ）とで構成されている。つまり、この熱交換ユニット（１８ｂ）によって貯留容器（１８ａ）を加熱すると該貯留容器（１８ａ）内で蒸発する冷媒によって内圧が上昇する一方、冷却すると該貯留容器（１８ａ）内で凝縮する冷媒によって内圧が下降する。

次に、この２次側冷媒回路（Ｂ）との間で熱交換を行う１次側冷媒回路（Ａ）について説明する。

この１次側冷媒回路（Ａ）は、圧縮機（11）、室外熱交換器（14）、室外電動膨張弁（ＥＶ2）、１次側熱源熱交換器（12）が冷媒配管（16）によって順に接続されて構成されている。そして、圧縮機（11）の吐出側が室外熱交換器（14）に、また、吸入側が１次側熱源熱交換器（12）にそれぞれ接続されている。

また、上記各電動弁（EV1, EV2）はコントーラ（Ｃ）によって開閉制御される。尚、図１における（Ｆ）は室内ファンである。

次に、上述の如く構成された本冷媒回路（A, B）における室内の冷房運転時について説明する。

この冷房運転開始時、１次側冷媒回路（Ａ）においては、圧縮機（11）が駆動し、図１に実線で示す矢印のように、圧縮機（11）から吐出した高温高圧のガス冷媒が、室外熱交換器（14）において外気との間で熱交換を行って凝縮した後、室外電動膨張弁（ＥＶ2）において減圧され、１次側熱源熱交換器（12）において２次側熱源熱交換器（１）との間で熱交換を行い、該２次側熱源熱交換器（１）の冷媒から熱を奪って蒸発し、圧縮機（11）に吸入される。この循環動作を繰り返す。

一方、２次側冷媒回路（Ｂ）にあっては、加減圧機構（18）の貯留容器（１８a）が熱交換ユニット（１８b）によって加熱され、該貯留容器（１８a）内で冷媒が蒸発し、その内圧が上昇する。この圧力は、図１に破線の矢印で示すように、加減圧管（19）によってタンク（Ｔ）内に作用し、該タンク（Ｔ）内の液冷媒の液面を押し下げながら、該液冷媒を接続管（17）を経て液配管（７）に押し出す。この押し出された液冷媒は、液配管（７）を室内熱交換器（3, 3）に向って流れ、室内電動膨張弁（EV1, EV1）によって減圧された後、各室内熱交換器（3, 3）において室内空気との間で熱交換を行い、蒸発して室内空気を冷却する。このガス冷媒は、ガス配管（６）を介して２次側熱源熱交換器（１）に流れ、１次側熱源熱交換器（12）との間で熱交換を行って凝縮する。

このような動作の後、加減圧機構（18）の貯留容器（１８a）を熱交換ユニット（１８b）によって冷却すると、該貯留容器（１８a）内で冷媒が凝縮し、その内圧が下降する。この圧力は、図１に一点鎖線の矢印で示すように、加減圧管（19）によってタンク（Ｔ）内に作用し、該タンク（Ｔ）の内圧が低下する。これにより、２次側熱

源熱交換器（１）で凝縮した液冷媒は、液配管（７）を経てタンク（Ｔ）内に回収される。

以上のように、加減圧機構（18）のタンク（Ｔ）に対する加圧及び減圧が繰り返されることにより、加圧時にはタンク（Ｔ）から液冷媒が押し出され、減圧時にはタンク（Ｔ）に液冷媒が回収される。この結果、２次側冷媒回路（Ｂ）において冷媒が循環し、室内を冷房する。

したがって、本実施形態の熱搬送装置では、２次側冷媒回路（Ｂ）にポンプ等の機械的な駆動源を設けることなく、該２次側冷媒回路（Ｂ）において熱搬送を行うことができる。このため、消費電力の低減、故障が発生する要因箇所の削減、装置全体としての信頼性の確保を図ることができる。

また、タンク（Ｔ）内に発生する吸引力によって２次側熱源熱交換器（１）の液冷媒を回収するようにしているので、タンク（Ｔ）を２次側熱源熱交換器（１）よりも低い位置に設置する必要がなくなり、機器の配設位置の制約が小さくでき、汎用性の向上を図ることができる。

また、冷媒圧力を利用することで、２次側冷媒回路（Ｂ）での冷媒循環が安定的に行われるので、この２次側冷媒回路（Ｂ）を大型にしても冷媒循環が良好に行われることになり、システムの拡大化が可能になり、また、高い信頼性が得られる。

－加減圧機構の変形例－

次に、上述した２次側冷媒回路（Ｂ）に適用可能な加減圧機構（18）の変形例について説明する。

図３は第１の変形例を示し、加減圧機構（18）は、加減圧用の圧縮機（１８ｃ）を備えている。詳しくは、加減圧管（19）が第１及び第２の２本の分岐管（19a，19b）に分岐され、第１分岐管（１９ａ）が圧縮機（１８ｃ）の吐出側に、第２分岐管（１９ｂ）が圧縮機（１８ｃ）の吸入側にそれぞれ接続されている。各分岐管（19a，19b）には第１及び第２の電磁弁（SV1，SV2）が設けられている。

これら各分岐管（19a，19b）及び電磁弁（SV1，SV2）が切換え手段（Ｉ）を構成している。尚、この加減圧用の圧縮機（１８ｃ）は１次側冷媒配管（Ａ）の圧縮機（11）

と兼用してもよい。

上記タンク（T）から液配管（7）に液冷媒を押し出す際には第1電磁弁（SV1）を開口すると共に、第2電磁弁（SV2）を閉鎖してタンク（T）内に高圧を作用させる。一方、タンク（T）に液配管（7）から液冷媒を回収する際には第2電磁弁（SV2）を開口すると共に、第1電磁弁（SV1）を閉鎖してタンク（T）内を低圧状態にする。

このタンク（T）に対する加圧及び減圧を繰り返し、上記実施形態と同様に、タンク（T）から液冷媒が押し出される状態と回収される状態とが交互に行われ、2次側冷媒回路（B）において冷媒を循環させながら室内を冷房する。

図4は加減圧機構（18）の第2の変形例を示し、該加減圧機構（18）は、タンク（T）を直接的に加熱及び冷却することによって該タンク（T）の内圧を変化させるものである。

つまり、タンク（T）に隣接して上記実施形態と同様の熱交換ユニット（18b）が配設されている。この熱交換ユニット（18b）によってタンク（T）を加熱すると該タンク（T）で蒸発する冷媒によってタンク内圧が上昇する一方、冷却すると凝縮する冷媒によってタンク内圧が下降する。

このタンク（T）に対する加熱及び冷却を繰り返し、2次側冷媒回路（B）において冷媒を循環させながら室内を冷房する。

図5は加減圧機構（18）の第3の変形例を示し、該加減圧機構（18）は、タンク（T）を直接的に加熱及び冷却する加減圧用の冷媒回路（D）を備えている。

この冷媒回路（D）は、圧縮機（D1）、四路切換弁（D2）、第1熱交換器（D3）、膨張弁（D4）、第2熱交換器（D5）が冷媒配管（D6）によって接続されて構成されている。該第1熱交換器（D3）のガス側は、四路切換弁（D2）を介して圧縮機（D1）の吸入側と吐出側とに切換え可能に接続されている。この第1熱交換器（D3）は、タンク（T）に隣接されて該タンク（T）との間で熱交換する。

上記タンク（T）から液配管（7）に液冷媒を押し出す際には、四路切換弁（D2）が実線側に切換えられ、圧縮機（D1）の吐出ガス冷媒が第1熱交換器（D3）の流れ、

タンク（T）内の冷媒に熱を与えて凝縮した後、膨張弁（D4）で減圧され、第２熱交換器（D5）で蒸発して圧縮機（D1）に戻る。この第１熱交換器（D3）の冷媒から熱を受けたタンク（T）の冷媒は蒸発し、これによって該タンク（T）内が高圧になり、この圧力により該タンク（T）から液配管（７）に液冷媒が押し出される。

逆に、タンク（T）に液配管（７）から液冷媒を回収する際には、四路切換弁（D2）が破線側に切換えられ、圧縮機（D1）の吐出ガス冷媒が第２熱交換器（D5）で凝縮し、膨張弁（D4）で減圧された後、第１熱交換器（D3）に流れ、タンク（T）内の冷媒から熱を奪って蒸発した後、圧縮機（D1）に戻る。この第１熱交換器（D3）の冷媒により熱を奪われたタンク（T）の冷媒は凝縮し、これによってタンク（T）内が低圧になり、この圧力により該タンク（T）に液配管（７）から液冷媒が回収される。

このタンク（T）に対する加熱及び冷却を繰り返し、２次側冷媒回路（B）において冷媒を循環させながら室内を冷房する。

図６は加減圧機構（18）の第４の変形例を示し、該加減圧機構（18）は、上記の加減圧用の冷媒回路（D）を備えている。タンク（T）に接続される加減圧管（19）には熱交換器（19c）が接続され、この熱交換器（19c）と冷媒回路（D）の第１熱交換器（D3）との間で熱交換が行われる。

上記四路切換弁（D2）の切換え状態を交互に切換えると、熱交換器（19c）が高圧状態と低圧状態とに交互に切換わり、これによって、熱交換器（19c）の圧力がタンク（T）に作用して、液冷媒のタンク（T）からの液冷媒の押出しと、タンク（T）への回収とが交互に行われる。

このタンク（T）に対する加圧及び減圧を繰り返し、２次側冷媒回路（B）において冷媒を循環させながら室内を冷房する。

－第２実施形態－

次に、本発明の第２実施形態について説明する。尚、ここでは上述した第１実施形態との相違点について説明する。

本実施形態の冷媒回路は、暖房専用の空気調和装置を構成し、１次側冷媒回路

（A）の構成及び液配管（7）に設けられる逆止弁が上述した第1実施形態のものと異なっている。

図7に示すように、1次側冷媒回路（A）は、圧縮機（11）、1次側熱源熱交換器（12）、室外電動膨張弁（EV2）、室外熱交換器（14）が冷媒配管（16）によって順に接続されて構成されている。該圧縮機（11）の吐出側は1次側熱源熱交換器（12）に、また、吸入側は室外熱交換器（14）にそれぞれ接続されている。

一方、2次側冷媒回路（B）には、第3逆止弁（CV3）と第4逆止弁（CV4）が設けられている。該第3逆止弁（CV3）は、液配管（7）における接続管（17）の接続位置と2次側熱源熱交換器（1）との間に設けられ、タンク（T）から2次側熱源熱交換器（1）への液冷媒の流通のみを許容する。上記第4逆止弁（CV4）は、液配管（7）における接続管（17）の接続位置と室内熱交換器（3, 3）との間に設けられ、室内熱交換器（3, 3）からタンク（T）への液冷媒の流通のみを許容する。

尚、本実施形態の加減圧機構（18）は、上述した第1実施形態のものと同様である。

次に、上述の如く構成された本冷媒回路（A, B）における室内の暖房運転時について説明する。

この暖房運転開始時、1次側冷媒回路（A）においては、圧縮機（11）が駆動し、図7に実線で示す矢印のように、圧縮機（11）から吐出した高温高圧のガス冷媒が、1次側熱源熱交換器（12）において2次側熱源熱交換器（1）との間で熱交換を行い、該2次側熱源熱交換器（1）の冷媒に熱を与えて凝縮した後、室外電動膨張弁（EV2）において減圧され、室外熱交換器（14）において外気との間で熱交換を行って蒸発し、圧縮機（11）に戻る。この循環動作を繰り返す。

一方、2次側冷媒回路（B）にあっては、加減圧機構（18）の貯留容器（18a）が熱交換ユニット（18b）によって加熱され、該貯留容器（18a）の冷媒が蒸発し、その内圧が上昇する（図2参照）。この圧力は、図7に破線の矢印で示すように、加減圧管（19）によってタンク（T）に作用し、該タンク（T）内の液冷媒の液面を押し下げながら、該液冷媒を接続管（17）を経て液配管（7）に押し出す。この押し出された液冷媒は、液配管（7）を2次側熱源熱交換器（1）に向って流れ、該2次側

熱源熱交換器（1）において1次側熱源熱交換器（12）の冷媒と熱交換を行い、蒸発した後、ガス配管（6）を経て室内熱交換器（3, 3）において室内空気と熱交換し、凝縮して室内空気を加熱する。

この動作の後、加減圧機構（18）の貯留容器（18a）を熱交換ユニット（18b）によって冷却すると、該貯留容器（18a）の冷媒が凝縮し、その内圧が下降する。この圧力は、図7に一点鎖線の矢印で示すように、加減圧管（19）によってタンク（T）に作用し、該タンク（T）の内圧が低下する。これにより、室内熱交換器（3）で凝縮した液冷媒が液配管（7）を経てタンク（T）内に回収される。

上記加減圧機構（18）のタンク（T）に対する加圧及び減圧を繰り返し、2次側冷媒回路（B）の冷媒が循環し、室内を暖房する。このように、本実施形態の熱搬送装置にあっても、2次側冷媒回路（B）にポンプ等の駆動源を設けることなく、該2次側冷媒回路（B）において熱搬送を行うことができる。

また、本第2実施形態の暖房専用の冷媒回路における加減圧機構（18）に対しても上述した各変形例で示した構成を適用することができる。

尚、上記第1実施形態及び第2実施形態における液配管（7）の逆止弁（CV1～CV4）に代えて、図8に示すように、開閉自在な電磁弁（SV-A, SV-B）を設け、た構成とし、加減圧機構（18）から作用する圧力状態に応じて各電磁弁（SV-A, SV-B）の開閉状態を切換えるようにしてもよい。

－第3実施形態－

次に、本発明の第3実施形態について説明する。尚、ここでは上述した第2実施形態との相違点について説明する。

本実施形態の冷媒回路は、室内の冷房及び暖房を選択的に行えるいわゆるヒートポンプの空気調和装置を構成するものである。

具体的に、図9に示すように、2次側冷媒回路（B）は、第3電磁弁（SV3）と第4電磁弁（SV4）が設けられている。該第3電磁弁（SV3）は、液配管（7）における第4逆止弁（CV4）と室内熱交換器（3, 3）との間に設けられ、室内の暖房運転

時に開口し、冷房運転時に閉鎖する。上記第４電磁弁（ＳＶ4）は、液配管（７）における第３逆止弁（ＣＶ3）と２次側熱源熱交換器（１）との間に設けられ、室内の暖房運転時に開口し、冷房運転時に閉鎖する。

上記液配管（７）における第３電磁弁（ＳＶ3）と室内熱交換器（3，3）との間には供給側冷房液配管（34）の一端が接続され、該供給側冷房液配管（34）の他端は、液配管（７）における第３逆止弁（ＣＶ3）と第４電磁弁（ＳＶ4）との間に接続されている。この供給側冷房液配管（34）には冷房運転時に開口し、暖房運転時に閉鎖する第５電磁弁（ＳＶ5）が設けられている。

上記液配管（７）における第４逆止弁（ＣＶ4）と第３電磁弁（ＳＶ3）との間には回収側冷房液配管（35）の一端が接続され、該回収側冷房液配管（35）の他端は、液配管（７）における第４電磁弁（ＳＶ4）と２次側熱源熱交換器（１）との間に接続されている。この回収側冷房液配管（35）には冷房運転時に開口し、暖房運転時に閉鎖する第６電磁弁（ＳＶ6）が設けられている。

一方、１次側冷媒回路（Ａ）は、１次側熱源熱交換器（12）によって２次側熱源熱交換器（１）を加熱及び冷却する。具体的に、圧縮機（11）、四路切換弁（22）、室外熱交換器（14）、室外電動膨張弁（ＥＶ2）、１次側熱源熱交換器（12）が冷媒配管（16）によって接続されている。上記１次側熱源熱交換器（12）のガス側は、四路切換弁（22）を介して圧縮機（11）の吸入側と吐出側とに切換わる。

以下、室内の冷房及び暖房運転時について説明する。

冷房運転時には、先ず、１次側冷媒回路（Ａ）の四路切換弁（22）を実線側に切換え、２次側冷媒回路（Ｂ）の第５電磁弁（ＳＶ5）及び第６電磁弁（ＳＶ6）を開口し、第３電磁弁（ＳＶ3）及び第４電磁弁（ＳＶ4）を閉鎖する。この状態で、１次側冷媒回路（Ａ）にあっては、上述した第１実施形態の場合と同様に、図９に実線の矢印で示す如く、圧縮機（11）から吐出した高温高圧のガス冷媒が、室外熱交換器（14）において外気と熱交換を行って凝縮した後、室外電動膨張弁（ＥＶ2）において減圧され、１次側熱源熱交換器（12）において２次側熱源熱交換器（１）との間で熱交換を行い、該２次側熱源熱交換器（１）の冷媒から熱を奪って蒸発し、圧縮機（11）に戻る。こ

の循環動作を繰り返す。

一方、２次側冷媒回路（Ｂ）にあっては、図９に実線の矢印で示すように、加減圧機構（18）から加減圧管（19）を経てタンク（Ｔ）に高圧が作用すると、該タンク（Ｔ）内の液冷媒の液面が押し下げられ、該液冷媒が接続管（17）を経て液配管（７）に押し出される。この押し出された液冷媒は、液配管（７）から供給側冷房液配管（34）を経て室内熱交換器（3，3）に向って流れ、室内電動膨張弁（EV1，EV1）によって減圧された後、室内熱交換器（3，3）において室内空気と熱交換を行い、蒸発して室内空気を冷却する。その後、このガス冷媒は、ガス配管（６）を介して２次側熱源熱交換器（１）に流れ、１次側熱源熱交換器（12）との間で熱交換を行って凝縮する。

この動作の後、加減圧機構（18）からタンク（Ｔ）に低圧を作用させ、該タンク（Ｔ）の内圧が低下すると、図９に破線の矢印で示すように、２次側熱源熱交換器（１）の液冷媒が、液配管（７）から回収側冷房液配管（35）を経てタンク（Ｔ）に回収される。

この加減圧機構（18）のタンク（Ｔ）に対する加圧及び減圧を繰り返し、２次側冷媒回路（Ｂ）の冷媒が循環し、室内を冷房する。

次に、室内の暖房運転時について説明する。

この暖房運転時には、先ず、１次側冷媒回路（Ａ）の四路切換弁（22）を破線側に切換え、２次側冷媒回路（Ｂ）の第３電磁弁（ＳＶ3）及び第４電磁弁（ＳＶ4）を開口し、第５電磁弁（ＳＶ5）及び第６電磁弁（ＳＶ6）を閉鎖する。この状態で、１次側冷媒回路（Ａ）にあっては、上述した第２実施形態の場合と同様に、図９に一点鎖線の矢印で示す如く、圧縮機（11）から吐出した高温高圧のガス冷媒が、１次側熱源熱交換器（12）において２次側熱源熱交換器（１）との間で熱交換を行い、該２次側熱源熱交換器（１）の冷媒に熱を与えて凝縮した後、室外電動膨張弁（ＥＶ2）において減圧され、室外熱交換器（14）において外気と熱交換を行って蒸発し、圧縮機（11）に戻る。この循環動作を繰り返す。

一方、２次側冷媒回路（Ｂ）にあっては、図９に一点鎖線の矢印で示すように、加減圧機構（18）から加減圧管（19）を経てタンク（Ｔ）に高圧が作用すると、該タ

ンク（T）内の液冷媒の液面が押し下げられ、該液冷媒を接続管（17）を経て液配管（7）に押し出す。この押し出された液冷媒は、液配管（7）を２次側熱源熱交換器（1）に向って流れ、該２次側熱源熱交換器（1）において１次側熱源熱交換器（12）の冷媒と熱交換を行い、蒸発した後、ガス配管（6）を経て室内熱交換器（3，3）において室内空気と熱交換を行い、凝縮して室内空気を加熱する。

この動作の後、加減圧機構（18）からタンク（T）に低圧を作用させ、該タンク（T）の内圧が低下すると、図9に二点鎖線の矢印で示すように、室内熱交換器（3）で凝縮した液冷媒が液配管（7）を経てタンク（T）に回収される。

この加減圧機構（18）のタンク（T）に対する加圧及び減圧を繰り返し、２次側冷媒回路（B）の冷媒が循環し、室内を暖房する。

－第４実施形態－

次に、本発明の第４実施形態について説明する。尚、ここでは上述した第１実施形態との相違点について説明する。

本実施形態の冷媒回路は、室内の空調運転を連続的に行うように２次側冷媒回路（B）を構成したものであって、上述した第１～第３の何れの実施形態の２次側冷媒回路（B）にも適用可能である。

具体的には、図１０に示すように、液配管（7）の一部が第１及び第２の分岐配管（7a，7b）に分岐され、それぞれに接続管（17a，17b）により第１及び第２のタンク（T1，T2）が接続されている。つまり、各タンク（T1，T2）は液配管（7）に対して並列に接続されている。

上記各タンク（T1，T2）の上端部には、それぞれ個別の第１及び第２の加減圧機構（18A，18B）が加減圧管（19d，19e）により接続されている。この各加減圧機構（18A，18B）は、一方の加減圧機構（１８A）が接続するタンク（T1）に対して高圧を作用させている際には、他方の加減圧機構（１８B）は接続するタンク（T2）に対して低圧を作用させ、このような高圧及び低圧の作用状態が交互に切換えられる。

また、各分岐配管（7a，7b）における接続管（17a，17b）の接続位置の両側には、加減圧機構（18A，18B）から作用する圧力状態に応じて切換え制御される電磁弁（SV7～SV10）が設けられている。

次に、空調運転動作について説明する。

例えば、室内の暖房運転時にあっては、第１タンク（T1）から液冷媒を押し出し、第２タンク（T2）に液冷媒を回収する際には、第１分岐配管（7a）では、熱源熱交換器（１）側に位置する電磁弁（ＳＶ7）を開口し、室内熱交換器（３）側に位置する電磁弁（ＳＶ8）を閉鎖する。一方、第２分岐配管（7b）では、熱源熱交換器（１）側に位置する電磁弁（ＳＶ9）を閉鎖し、室内熱交換器（３）側に位置する電磁弁（SV10）を開口する。この状態で、第１加減圧機構（１８A）から第１タンク（T1）に高圧を、第２加減圧機構（１８B）から第２タンク（T2）に低圧をそれぞれ作用させ、２次側冷媒回路（Ｂ）の冷媒循環が行われる。

このような運転状態が所定時間継続され、第１タンク（T1）内の液冷媒の殆どが排出されると、第２タンク（T2）から液冷媒を排出し、第１タンク（T1）に液冷媒を回収する運転状態に切換えられる。この運転時には、第１分岐配管（7a）では、熱源熱交換器（１）側に位置する電磁弁（ＳＶ7）を閉鎖し、室内熱交換器（３）側に位置する電磁弁（ＳＶ8）を開口する。一方、第２分岐配管（7b）では、熱源熱交換器（１）側に位置する電磁弁（ＳＶ9）を開口し、室内熱交換器（３）側に位置する電磁弁（SV10）を閉鎖する。この状態で、第２加減圧機構（１８B）から第２タンク（T2）に高圧を、第１加減圧機構（１８A）から第１タンク（T1）に低圧をそれぞれ作用させ、２次側冷媒回路（Ｂ）の冷媒循環が行われる。

このような各加減圧機構（18A, 18B）から各タンク（T1, T2）への圧力導入状態及び各電磁弁（ＳＶ7～SV10）の開閉状態が交互に切換えられることにより、室内の空調運転が連続的に行われる。尚、室内の冷房運転時には、上記各電磁弁（ＳＶ7～SV10）の開閉動作が逆になる。

－加減圧機構の変形例－

次に、上述した第４実施形態の２次側冷媒回路（Ｂ）に適用可能な加減圧機構（18）の変形例を図１１を用いて説明する。

この変形例では、加減圧用の圧縮機（D1）を備えている。詳しくは、各タンク（T1, T2）から延びる加減圧管（19d, 19e）に四路切換弁（D2）を介して圧縮機（D1）

を接続し、この四路切換弁（D2）の切換え動作によって、第１タンク（T1）が圧縮機（D1）の吐出側に、第２タンク（T2）が圧縮機（D1）の吸入側にそれぞれ接続した状態（図１１に破線で示す切換え状態）と、第１タンク（T1）が圧縮機（D1）の吸入側に、第２タンク（T2）が圧縮機（D1）の吐出側にそれぞれ接続した状態（図１１に実線で示す切換え状態）とが切換わる。

この四路切換弁（D2）の切換え動作により、上述した第４実施形態と同様に、各タンク（T1, T2）への圧力導入状態が交互に切換えられることになり、これによって室内の空調運転が連続的に行われる。

尚、本変形例では、上述した第４実施形態での各電磁弁（ＳＶ7～SV10）に代えて逆止弁（ＣＶ7～CV10）を使用している。つまり、この図１１に示すものは暖房用の回路であって、冷房専用の回路にあっては、これら逆止弁として冷媒流通許容方向が逆のものが設けられる。

－第５実施形態－

次に、上述した第４実施形態の如く、複数のタンク（T1, T2）を設け、連続した空調運転を行う空気調和装置の具体的な２次側冷媒回路（Ｂ）について説明する。

図１２に示すものは、冷房専用の空気調和装置を構成している。本２次側冷媒回路（Ｂ）は、各タンク（T1, T2）から延びる加減圧管（19d, 19e）に第１及び第２の駆動用熱交換器（E1, E2）が接続されると共に、加減圧用の冷媒回路（Ｄ）を備えている。この冷媒回路（Ｄ）と駆動用熱交換器（E1, E2）との間で熱交換を行うことによって各タンク（T1, T2）に対して冷媒循環用の圧力を作用させる。

この冷媒回路（Ｄ）について説明すると、圧縮機（D1）、第１駆動用熱交換器（E1）との間で熱交換が可能な第１熱交換器（D3）、膨張弁（D4）、第２駆動用熱交換器（E2）との間で熱交換が可能な第２熱交換器（D5）が冷媒配管（D6）によって接続されている。詳しくは、第１熱交換器（D3）のガス側配管（D3-G）は分岐され、圧縮機（D1）の吐出側及び吸入側にそれぞれ接続されている。この分岐配管のうち圧縮機（D1）の吐出側の配管（D3-G1）には第１吐出側電磁弁（SV-O1）が、吸入側の配管（D3-G2）には第１吸入側電磁弁（SV-I1）がそれぞれ設けられている。

同様に、第２熱交換器（D5）のガス側配管（D5-G）も分岐され、圧縮機（D1）の

吐出側及び吸入側にそれぞれ接続され、圧縮機（D1）の吐出側の配管（D5-G1）には第２吐出側電磁弁（SV-02）が、吸入側の配管（D5-G2）には第２吸入側電磁弁（SV-I2）がそれぞれ設けられている。更に、各熱交換器（D3, D5）の液側同士は、液配管（D6-L）により上記膨張弁（D4）を介して互いに連結されている。

各駆動用熱交換器（E1, E2）には、駆動用の冷媒を貯留する貯留器（20, 21）が接続されている。上記液配管（７）の第１分岐配管（7a）における接続管（17a）の接続位置と２次側熱源熱交換器（１）との間には、該２次側熱源熱交換器（１）から第１タンク（T1）への液冷媒の流通のみを許容する逆止弁（CV1-A）が設けられている。上記第１分岐配管（7a）における接続管（17a）の接続位置と室内熱交換器（3, 3）との間には、第１タンク（T1）から室内熱交換器（3, 3）への液冷媒の流通のみを許容する逆止弁（CV2-A）が設けられている。

一方、液配管（７）の第２分岐配管（7b）における接続管（17b）の接続位置と２次側熱源熱交換器（１）との間には、該２次側熱源熱交換器（１）から第２タンク（T2）への液冷媒の流通のみを許容する逆止弁（CV1-B）が設けられている。上記第２分岐配管（7b）における接続管（17b）の接続位置と室内熱交換器（3, 3）との間には、第２タンク（T2）から室内熱交換器（3, 3）への液冷媒の流通のみを許容する逆止弁（CV2-B）が設けられている。

その他の構成は上述した第１実施形態の１次側及び２次側冷媒回路（A, B）と同様である。

次に、室内の冷房運転動作を説明する。

先ず、加減圧用の冷媒回路（D）にあっては、第１吐出側電磁弁（SV-01）及び第２吸入側電磁弁（SV-I2）を開口し、第１吸入側電磁弁（SV-I1）及び第２吐出側電磁弁（SV-02）を閉鎖し、圧縮機（D1）を駆動する。この圧縮機（D1）から吐出した高温高圧のガス冷媒は、図１２に実線の矢印で示すように、ガス側配管（D3-G1）を経て第１熱交換器（D3）に流れ、この第１熱交換器（D3）において第１駆動用熱交換器（E1）との間で熱交換を行い、該第１駆動用熱交換器（E1）の冷媒に熱を与えて凝縮する。その後、液冷媒は、液配管（D6-L）において膨張弁（D4）で減圧され、第２熱交換器（D5）において第２駆動用熱交換器（E2）との間で熱交換を行い、該第２

駆動用熱交換器（E2）の冷媒から熱を奪って蒸発し、ガス側配管（D5-G2）を経て圧縮機（D1）に戻る。この循環動作を繰り返す。

この冷媒循環動作により、第１駆動用熱交換器（E1）では冷媒が蒸発して内部が高圧になる一方、第２駆動用熱交換器（E2）では冷媒が凝縮して内部が低圧になる。このため、第１タンク（T1）には高圧が、第２タンク（T2）には低圧がそれぞれ作用し、上述した第４実施形態のような第１タンク（T1）からの液冷媒の排出動作と、第２タンク（T2）への液冷媒の回収動作とが同時に行われ、２次側冷媒回路（B）の冷媒循環が行われる。

この運転状態が所定時間継続された後、加減圧用の冷媒回路（D）の各電磁弁の開閉状態を切換える。つまり、第１吐出側電磁弁（SV-O1）及び第２吸入側電磁弁（SV-I2）を閉鎖し、第１吸入側電磁弁（SV-I1）及び第２吐出側電磁弁（SV-O2）を開口する。これにより、図１２に破線で示す矢印のように冷媒が流れ、第１駆動用熱交換器（E1）では冷媒が凝縮して内部が低圧になる一方、第２駆動用熱交換器（E2）では冷媒が蒸発して内部が高圧になる。このため、第１タンク（T1）には低圧が、第２タンク（T2）には高圧がそれぞれ作用して、第２タンク（T2）からの液冷媒の排出動作と、第１タンク（T1）への液冷媒の回収動作とが同時に行われ、２次側冷媒回路（B）の冷媒循環が行われる。

この各タンク（T1，T2）への圧力導入状態が交互に切換えられることにより、室内の冷房運転を連続して行わせることができる。また、低圧が作用する側のタンク（T2）から駆動用熱交換器（E2）に回収されたガス冷媒は、該駆動用熱交換器（E2）で凝縮して一時的に貯留器（21）に溜められる。そして、各電磁弁（SV-O1～SV-I2）の開閉状態が切換えられた際には、この冷媒が駆動用熱交換器（E2）において蒸発してタンク（T2）に高圧を作用させる。

－第６実施形態－

次に、複数のタンク（T1，T2）を備えて連続運転を行う空気調和装置であって、暖房専用の空気調和装置の場合について説明する。尚、本実施形態では上述した第５実施形態との相違点について説明する。

図１３に示すように、本２次側冷媒回路（B）は、液配管（７）の各分岐配管

(7a, 7b) に設けられている逆止弁の構成が上述した第５実施形態のものと異なっている。

つまり、液配管（７）の第１分岐配管 (7a) における接続管（１７a）の接続位置と２次側熱源熱交換器（１）との間には、第１タンク (T1) から２次側熱源熱交換器（１）への液冷媒の流通のみを許容する逆止弁 (CV3-A) が設けられている。この第１分岐配管 (7a) における接続管（１７a）の接続位置と室内熱交換器 (3, 3) との間には、室内熱交換器 (3, 3) から第１タンク (T1) への液冷媒の流通のみを許容する逆止弁 (CV4-A) が設けられている。

一方、液配管（７）の第２分岐配管 (7b) における接続管（１７b）の接続位置と２次側熱源熱交換器（１）との間には、第２タンク (T2) から２次側熱源熱交換器（１）への液冷媒の流通のみを許容する逆止弁 (CV3-B) が設けられている。この第２分岐配管 (7b) における接続管（１７b）の接続位置と室内熱交換器 (3, 3) との間には、室内熱交換器 (3, 3) から第２タンク (T2) への液冷媒の流通のみを許容する逆止弁 (CV4-B) が設けられている。

その他の構成は上述した第５実施形態の２次側冷媒回路と同様である。

次に、暖房運転動作について説明する。

上述した第５実施形態のように、加減圧用の冷媒回路（Ｄ）の各電磁弁の切換え動作が行われる。つまり、第１吐出側電磁弁 (SV-O1) 及び第２吸入側電磁弁 (SV-I2) を開口し、第１吸入側電磁弁 (SV-I1) 及び第２吐出側電磁弁 (SV-O2) を閉鎖する状態と、第１吐出側電磁弁 (SV-O1) 及び第２吸入側電磁弁 (SV-I2) を閉鎖し、第１吸入側電磁弁 (SV-I1) 及び第２吐出側電磁弁 (SV-O2) を開口した状態とが交互に繰り返される。これにより、第１タンク (T1) に高圧が、第２タンク (T2) に低圧がそれぞれ作用される運転状態と、第１タンク (T1) に低圧が、第２タンク (T2) に高圧がそれぞれ作用される運転状態とが交互に繰り返される。そして、一方のタンク (T1) から液配管（７）に排出された液冷媒が２次側熱源熱交換器（１）で蒸発した後、室内熱交換器 (3, 3) で凝縮して室内空気を加熱する。その後、この液冷媒が他方のタンク (T2) に回収される。この冷媒循環動作を繰り返し、室内の暖房運転が連続して行われる。

－第７実施形態－

次に、本発明の第７実施形態について説明する。尚、ここでは上述した第６実施形態との相違点について説明する。

本実施形態の２次側冷媒回路（Ｂ）は、ヒートポンプの空気調和装置に適用したものである。

具体的には、図１４に示すように、液配管（７）における室内熱交換器（３）側の各分岐配管（7a, 7b）の分岐点と室内熱交換器（3, 3）との間には、室内の暖房運転時に開口し、冷房運転時に閉鎖する第３電磁弁（ＳＶ３）が設けられている。上記液配管（７）における２次側熱源熱交換器（１）側の各分岐配管（7a, 7b）の分岐点と２次側熱源熱交換器（１）との間には、室内の暖房運転時に開口し、冷房運転時に閉鎖する第４電磁弁（ＳＶ４）が設けられている。

また、液配管（７）における第３電磁弁（ＳＶ３）と室内熱交換器（3, 3）との間には供給側冷房液配管（34）の一端が接続され、該供給側冷房液配管（34）の他端が第１分岐配管（7a）における第３逆止弁（CV3-A）の下流側に接続されている。この供給側冷房液配管（34）には、冷房運転時に開口し、暖房運転時に閉鎖する第５電磁弁（ＳＶ５）が設けられている。

上記液配管（７）における第４電磁弁（ＳＶ４）と２次側熱源熱交換器（１）との間には回収側冷房液配管（35）の一端が接続され、該回収側冷房液配管（35）の他端が第２分岐配管（7b）における第４逆止弁（CV4-B）の上流側に接続されている。この回収側冷房液配管（35）には、冷房運転時に開口し、暖房運転時に閉鎖する第６電磁弁（ＳＶ６）が設けられている。

その他の構成は上記第６実施形態と同様である。また、本実施形態における１次側冷媒回路は上述した第３実施形態のものが使用される。

以下、室内の冷房及び暖房運転時について説明する。

冷房運転時には、上述した第５実施形態の場合と同様に、加減圧用の冷媒回路（Ｄ）の各電磁弁の切換え動作により、図１４に実線の矢印で示す如く、一方のタンク（T1）から分岐配管（7a）に排出された液冷媒が供給側冷房液配管（34）を経て室

内電動膨張弁（EV1, EV1）で減圧された後、室内熱交換器（3, 3）で蒸発して室内空気を冷却する。その後、このガス冷媒は、２次側熱源熱交換器（１）で凝縮された後、回収側冷房液配管（35）を経て他方のタンク（T2）に回収される。この冷媒循環動作を繰り返し、液冷媒を排出するタンクと回収するタンクとが交互に切換わり、室内の冷房運転が連続して行われる。

一方、室内の暖房運転時には、上述した第６実施形態の場合と同様に、加減圧用の冷媒回路（Ｄ）の各電磁弁の切換え動作により、図１４に破線の矢印で示す如く、一方のタンク（T1）から液配管（７）に排出された液冷媒が、液配管（７）を２次側熱源熱交換器（１）に向って流れ、該２次側熱源熱交換器（１）において１次側熱源熱交換器（12）の冷媒と熱交換を行い、蒸発した後、ガス配管（６）を経て室内熱交換器（3, 3）において室内空気との間で熱交換を行い、凝縮して室内空気を加熱する。その後、室内熱交換器（３）で凝縮した液冷媒は液配管（７）を経て他方のタンク（T2）内に回収される。この冷媒循環動作を繰り返し、液冷媒を排出するタンクと回収するタンクとが交互に切換わり、室内の暖房運転が連続して行われる。

－第８実施形態－

次に、本発明の第８実施形態について説明する。尚、ここでは上述した第６実施形態との相違点について説明する。

本実施形態の２次側冷媒回路（Ｂ）も、ヒートポンプの空気調和装置に適用したものであり、１次側冷媒回路（Ａ）の構成は、上述した第３実施形態のものと同様であるので説明を省略する。

図１５に示すように、２次側冷媒回路（Ｂ）は、液配管（７）に四路切換弁（10）を備えている。この四路切換弁（10）は、２次側熱源熱交換器（１）の液側から延びる第１の液配管（7A）と、液配管（７）の各分岐配管（7a, 7b）の両端の第１及び第２の分岐部（X, Y）（尚、 Xは２次側熱源熱交換器（１）側の分岐部、 Yは室内熱交換器（３）側の分岐部）からそれぞれ延びる第２及び第３の液配管（7B, 7C）と、室内熱交換器（3, 3, 3）からタンク（T1, T2）に向って延びる第４の液配管（7D）とがそれぞれ接続されている。

上記四路切換弁（10）は、２次側熱源熱交換器（１）の液側及び室内熱交換器

(3, 3, 3) の液側と、各分岐配管 (7a, 7b) の各分岐部 (X, Y) との接続状態を切換える。つまり、2次側熱源熱交換器 (1) の液側が第1の分岐部 (X) に接続され、室内熱交換器 (3, 3, 3) の液側が第2の分岐部 (Y) に接続される状態 (図15に実線で示す切換え状態) と、2次側熱源熱交換器 (1) の液側が第2の分岐部 (Y) に接続され、室内熱交換器 (3, 3, 3) の液側が第1の分岐部に (X) 接続される状態 (図15に破線で示す切換え状態) とに切換わる。

また、本実施形態の2次側冷媒回路 (B) は室内熱交換器 (3) を3台備えている。その他の構成は、上述した第6実施形態のものと同様である。尚、図15における (28) はアキュムレータである。

以下、室内の冷房及び暖房運転動作について説明する。

先ず、冷房運転時には、1次側冷媒回路 (A) では、四路切換弁 (22) が実線側に切換えられる一方、2次側冷媒回路 (B) では、四路切換弁 (10) が破線側に切換えられる。この状態で、1次側冷媒回路 (A) の圧縮機 (11) 及び2次側冷媒回路 (B) の加減圧用冷媒回路 (D) の圧縮機 (D1) を共に駆動する。

これにより、上述した第5実施形態の場合と同様に、加減圧用の冷媒回路 (D) の各電磁弁の切換え動作により、図15に実線の矢印で示す如く、一方のタンク (T1) から排出された液冷媒が第2液配管 (7B) を経て四路切換弁 (10) 及び第4液配管 (7D) を流れた後、室内電動膨張弁 (EV1, EV1) で減圧され、室内熱交換器 (3, 3) で蒸発して室内空気を冷却する。その後、このガス冷媒は、ガス配管 (6) を経て2次側熱源熱交換器 (1) で凝縮した後、第1液配管 (7A)、四路切換弁 (10) 及び第3液配管 (7C) を経て他方のタンク (T2) に回収される。この冷媒循環動作を繰り返し、液冷媒を排出するタンクと回収するタンクとが交互に切換わり、室内の冷房運転が連続して行われる。

一方、室内の暖房運転時には、1次側冷媒回路 (A) では、四路切換弁 (22) が破線側に切換えられる一方、2次側冷媒回路 (B) では、四路切換弁 (10) が実線側に切換えられる。この状態で、1次側冷媒回路 (A) の圧縮機 (11) 及び2次側冷媒回路 (B) の加減圧用冷媒回路 (D) の圧縮機 (D1) を共に駆動する。

これにより、上述した第６実施形態の場合と同様に、加減圧用の冷媒回路（Ｄ）の各電磁弁の切換え動作により、図１５に破線の矢印で示す如く、一方のタンク（T1）から排出された液冷媒が第２液配管（7B）を経て四路切換弁（10）及び第１液配管（7A）を流れた後、２次側熱源熱交換器（１）において１次側熱源熱交換器（12）の冷媒と熱交換を行い蒸発する。このガス冷媒は、ガス配管（６）を経て室内熱交換器（3，3）に導入され室内空気との間で熱交換を行い、凝縮して室内空気を加熱する。その後、室内熱交換器（３）で凝縮した液冷媒は第４液配管（7D）、四路切換弁（10）及び第３液配管（7C）を経て他方のタンク（T2）内に回収される。この冷媒循環動作を繰り返し、液冷媒を排出するタンクと回収するタンクとが交互に切換わり、室内の暖房運転が連続して行われる。

－第９実施形態－

次に、本発明の第９実施形態について説明する。尚、ここでは上述した第８実施形態との相違点について説明する。本実施形態の２次側冷媒回路（Ｂ）も、ヒートポンプの空気調和装置に適用したものであり、１次側冷媒回路（Ａ）の構成は、上述した第３実施形態のものと同様であるので説明を省略する。また、本実施形態は加減圧機構（18）の構成が上述した第８実施形態のものと異なっているので、この加減圧機構（18）についてのみ説明する。

図１６に示すように、２次側冷媒回路（Ｂ）は、各タンク（T1，T2）から延びる第１及び第２の加減圧管（19d，19e）に第１及び第２の駆動用熱交換器（E1，E2）が接続されると共に、加減圧用の冷媒回路（Ｄ）を備えている。この冷媒回路（Ｄ）と駆動用熱交換器（E1，E2）との間で熱交換を行うことによって各タンク（T1，T2）に対して冷媒循環用の圧力を作用させる。

詳しくは、第１タンク（T1）の上端部に接続されている第１加減圧管（１９d）は、第１駆動用熱交換器（E1）に繋がる第１加減圧分岐管（19d-A）と、第２駆動用熱交換器（E2）に繋がる第２加減圧分岐管（19d-B）とに分岐されている。この第１加減圧分岐管（19d-A）には第１加減圧用電磁弁（SV-1）が、第２加減圧分岐管（19d-B）には第２加減圧用電磁弁（SV-2）がそれぞれ設けられている。

一方、第２タンク（T2）の上端部に接続されている第２加減圧管（１９e）は、上

記第１加減圧分岐管（19d-A）における第１加減圧用電磁弁（SV-1）と第１駆動用熱交換器（E1）との間に繋がる第３加減圧分岐管（19e-A）と、第２加減圧分岐管（19d-B）における第２加減圧用電磁弁（SV-2）と第２駆動用熱交換器（E2）との間に繋がる第４加減圧分岐管（19e-B）とに分岐されている。この第３加減圧分岐管（19e-A）には第３加減圧用電磁弁（SV-3）が、第４加減圧分岐管（19e-B）には第４加減圧用電磁弁（SV-4）がそれぞれ設けられている。

また、冷媒回路（D）について説明すると、圧縮機（D1）、第２駆動用熱交換器（E2）との間で熱交換が可能な第２熱交換器（D3）、膨張弁（D4）、第１駆動用熱交換器（E1）との間で熱交換が可能な第１熱交換器（D5）が冷媒配管（D6）によって順に接続されている。

各駆動用熱交換器（E1, E2）には、駆動用の冷媒を貯留する貯留器（20, 21）が逆止弁（CV-1, CV-2）及び電磁弁（SV-5～SV-8）を介して接続されている。詳しくは、第１駆動用熱交換器（E1）は、その下端部が各貯留器（20, 21）の下端部に対して、第１駆動用熱交換器（E1）から各貯留器（20, 21）への液冷媒の流通のみを許容する逆止弁（CV1, CV1）を介して接続されている。第２駆動用熱交換器（E2）は、その下端部が各貯留器（20, 21）の下端部に対して、各貯留器（20, 21）から第２駆動用熱交換器（E2）への液冷媒の流通のみを許容する逆止弁（CV2, CV2）を介して接続されている。

第１の貯留器（20）の上端部は、第５電磁弁（SV-5）を介して第１加減圧分岐管（19d-A）に、また、第６電磁弁（SV-6）を介して第２加減圧分岐管（19d-B）にそれぞれ接続されており、第２の貯留器（21）の上端部は、第７電磁弁（SV-7）を介して第１加減圧分岐管（19d-A）に、また、第８電磁弁（SV-8）を介して第２加減圧分岐管（19d-B）にそれぞれ接続されている。

その他の構成は上述した第８実施形態の１次側及び２次側冷媒回路（A, B）と同様である。

以下、室内の冷房及び暖房運転動作について説明する。

先ず、冷房運転時には、１次側冷媒回路（A）では、四路切換弁（22）が実線側に切換えられる一方、２次側冷媒回路（B）では、四路切換弁（10）が破線側に切換

えられる。また、加減圧機構（18）にあっては、第２電磁弁（SV-2）、第３電磁弁（SV-3）、第６電磁弁（SV-6）及び第７電磁弁（SV-7）を開口する一方、第１電磁弁（SV-1）、第４電磁弁（SV-4）、第５電磁弁（SV-5）及び第８電磁弁（SV-8）を閉鎖する。

この状態で、１次側冷媒回路（Ａ）の圧縮機（11）及び２次側冷媒回路（Ｂ）の加減圧用冷媒回路（Ｄ）の圧縮機（D1）を共に駆動する。そして、加減圧用冷媒回路（Ｄ）では、圧縮機（D1）から吐出した冷媒が、図１７に実線の矢印で示すように、第１熱交換器（D3）に流れ、この第１熱交換器（D3）において第２駆動用熱交換器（E2）との間で熱交換を行い、該第２駆動用熱交換器（E2）の冷媒に熱を与えて凝縮する。その後、膨張弁（D4）で減圧され、第２熱交換器（D5）において第１駆動用熱交換器（E1）との間で熱交換を行い、該第１駆動用熱交換器（E1）の冷媒から熱を奪って蒸発し、圧縮機（D1）に戻る。この循環動作を繰り返す。

この冷媒循環動作により、第２駆動用熱交換器（E2）では冷媒が蒸発して内部が高圧になる一方、第１駆動用熱交換器（E1）では冷媒が凝縮して内部が低圧になる。このため、第１タンク（T1）には第２加減圧分岐管（19d-B）より高圧が、第２タンク（T2）には第３加減圧分岐管（19e-A）より低圧がそれぞれ作用し、上記第４実施形態のような第１タンク（T1）からの液冷媒の排出動作と、第２タンク（T2）への液冷媒の回収動作とが同時に行われ、２次側冷媒回路（Ｂ）の冷媒循環が行われる。

つまり、図１７の破線で示す矢印のように、第１タンク（T1）から排出された液冷媒が第２液配管（7B）を経て四路切換弁（10）及び第４液配管（7D）を流れた後、室内電動膨張弁（EV1, EV1, …）で減圧され、室内熱交換器（3, 3, …）で蒸発して室内空気を冷却する。その後、このガス冷媒は、ガス配管（６）を経て２次側熱源熱交換器（１）で凝縮された後、第１液配管（7A）、四路切換弁（10）及び第３液配管（7C）を経て第２タンク（T2）に回収される。この冷媒循環動作を繰り返す。

このとき、第２タンク（T2）から第３加減圧分岐管（19e-A）を経て第１駆動用熱交換器（E1）に回収されたガス冷媒は、凝縮して第２の貯留器（21）に溜る。第２駆動用熱交換器（E2）は、第１の貯留器（20）に均圧され、該貯留器（20）内の液冷媒が第２駆動用熱交換器（E2）に供給される。

このような冷媒循環動作が所定時間行われた後、各電磁弁の切換え動作が行われ

る。つまり、第１電磁弁（SV-1）、第４電磁弁（SV-4）、第５電磁弁（SV-5）及び第８電磁弁（SV-8）を開口する一方、第２電磁弁（SV-2）、第３電磁弁（SV-3）、第６電磁弁（SV-6）及び第７電磁弁（SV-7）を閉鎖する。

このように各電磁弁が切換えられると、第２タンク（T2）には第４加減圧分岐管（19e-B）より高圧が、第１タンク（T1）には第１加減圧分岐管（19d-A）より低圧がそれぞれ作用し、第２タンク（T2）からの液冷媒の排出動作と、第１タンク（T1）への液冷媒の回収動作とが同時に行われ、２次側冷媒回路（B）の冷媒循環が行われる。

つまり、図１７の一点鎖線で示す矢印のように、第２タンク（T2）から排出された液冷媒が第２液配管（7B）を経て四路切換弁（10）及び第４液配管（7D）を流れた後、室内電動膨張弁（EV1, EV1, …）で減圧され、室内熱交換器（3, 3, …）で蒸発して室内空気を冷却する。その後、このガス冷媒は、ガス配管（６）を経て２次側熱源熱交換器（１）で凝縮された後、第１液配管（7A）、四路切換弁（10）及び第３液配管（7C）を経て第１タンク（T1）に回収される。この冷媒循環動作を繰り返す。

このとき、第１タンク（T1）から第１加減圧分岐管（19d-A）を経て第１駆動用熱交換器（E1）に回収されたガス冷媒は、凝縮して第１の貯留器（20）に溜る。第２駆動用熱交換器（E2）は、第２の貯留器（21）に均圧し、該貯留器（21）の液冷媒が第２駆動用熱交換器（E2）に供給される。

このようにして液冷媒を排出するタンクと回収するタンクとが交互に切換わり、室内の冷房運転が連続して行われる。

一方、室内の暖房運転時には、１次側冷媒回路（A）では、四路切換弁（22）が破線側に切換えられる一方、２次側冷媒回路（B）では、四路切換弁（10）が実線側に切換えられる。そして、加減圧機構（18）にあっては、上述した冷房運転時の場合と同様に、第２電磁弁（SV-2）、第３電磁弁（SV-3）、第６電磁弁（SV-6）及び第７電磁弁（SV-7）を開口する一方、第１電磁弁（SV-1）、第４電磁弁（SV-4）、第５電磁弁（SV-5）及び第８電磁弁（SV-8）を閉鎖する。この結果、第１タンク（T1）に高圧が、第２タンク（T2）に低圧がそれぞれ作用する状態になる。

また、第１電磁弁（SV-1）、第４電磁弁（SV-4）、第５電磁弁（SV-5）及び第８

電磁弁（SV-8）を開口する一方、第２電磁弁（SV-2）、第３電磁弁（SV-3）、第６電磁弁（SV-6）及び第７電磁弁（SV-7）を閉鎖する。この結果、第２タンク（T2）に高圧が、第１タンク（T1）に低圧がそれぞれ作用する状態になる。この２つの状態が交互に切換わる。

これにより、図１８に破線及び一点鎖線で示す矢印のように、一方のタンク（T1）から排出された液冷媒が第２液配管（7B）を経て四路切換弁（10）及び第１液配管（7A）を流れた後、２次側熱源熱交換器（１）において１次側熱源熱交換器（12）の冷媒と熱交換を行い蒸発する。このガス冷媒は、ガス配管（６）を経て室内熱交換器（3, 3）に導入され室内空気との間で熱交換を行い、凝縮して室内空気を加熱する。その後、室内熱交換器（３）で凝縮した液冷媒は第４液配管（7D）、四路切換弁（10）及び第３液配管（7C）を経て他方のタンク（T2）内に回収される。この冷媒循環動作を繰り返し、液冷媒を排出するタンクと回収するタンクとが交互に切換わり、室内の暖房運転が連続して行われる。

－第１０実施形態－

次に、本発明の第１０実施形態について図１９～図２１を用いて説明する。本実施形態は、上述した第９実施形態の２次側冷媒回路（Ｂ）と略同様の回路に組合せる１次側冷媒回路（Ａ）の変形例である。また、本実施形態は、冷房及び暖房を行うヒートポンプ回路としたものである。

先ず、１次側冷媒回路（Ａ）について説明する。

この１次側冷媒回路（Ａ）は、圧縮機（11）、四路切換弁（22）、室外熱交換器（14）、室外電動膨張弁（ＥＶＷ）、１次側熱源熱交換器（12）が冷媒配管（16）によって接続されて構成されている。この１次側熱源熱交換器（12）のガス側は、四路切換弁（22）を介して圧縮機（11）の吸入側と吐出側とに切換わる。

室外熱交換器（14）と室外電動膨張弁（ＥＶＷ）との間には、該室外電動膨張弁（ＥＶＷ）から室外熱交換器（14）への冷媒の流通のみを許容する逆止弁（CV-1）が設けられている。この１次側冷媒回路（Ａ）は、２次側冷媒回路（Ｂ）との間で熱交換を行うための加熱用熱交換器（D3）及び冷却用熱交換器（D5）を備えている。

加熱用熱交換器（D3）の一端（図１９における下端）は、逆止弁（CV-2）を備え

た第１冷房液ライン（CL-1）により上記室外熱交換器（14）と逆止弁（CV-1）との間の冷媒配管（16）に接続され、他端（図１９における上端）は、第２冷房液ライン（CL-2）により上記室外電動膨張弁（ＥＶＷ）と逆止弁（CV-1）との間の冷媒配管（16）に接続されている。

この第２冷房液ライン（CL-2）には、冷房時に開口し、暖房時に閉鎖する第１電磁弁（SV-1）が設けられている。上記第２冷房液ライン（CL-2）における加熱用熱交換器（D3）と第１電磁弁（SV-1）との間には暖房ガスライン（ＷＧＬ）の一端が接続され、該暖房ガスライン（ＷＧＬ）の他端は圧縮機（11）の吐出側に接続されている。この暖房ガスライン（ＷＧＬ）には、暖房時に開口し、冷房時に閉鎖する第２電磁弁（SV-2）が設けられている。

第１冷房液ライン（CL-1）と１次側熱源熱交換器（12）との間には暖房液ライン（ＷＬＬ）が接続され、この暖房液ライン（ＷＬＬ）には、暖房時に開口し、冷房時に閉鎖する第３電磁弁（SV-3）が設けられている。更に、冷却用熱交換器（D5）の一端（図１９における下端）は、吸入ガスライン（ＩＧＬ）により圧縮機（11）の吸入側に接続され、他端（図１９における上端）は、冷却用液ライン（ＣＬＬ）により上記第１冷房液ライン（CL-1）に接続されている。この冷却用液ライン（ＣＬＬ）には電動弁（D4）が設けられている。

一方、２次側冷媒回路（Ｂ）は、上記１次側熱源熱交換器（12）との間で熱交換する２次側熱源熱交換器（１）と室内熱交換器（3，3，3）とがガス配管（６）及び液配管（７）によって接続されている。液配管（７）の一部は、第１及び第２の分岐配管（7a，7b）に分岐され、それぞれに液冷媒を貯留した第１及び第２のタンク（T1，T2）が接続管（17a，17b）により接続されている。液配管（７）の第１分岐配管（7a）における接続管（１７ａ）の接続位置と２次側熱源熱交換器（１）との間には、第１タンク（T1）から２次側熱源熱交換器（１）への液冷媒の流通のみを許容する逆止弁（CV3-A）が設けられている。この第１分岐配管（7a）における接続管（１７ａ）の接続位置と室内熱交換器（3，3）との間には、室内熱交換器（3，3）から第１タンク（T1）への液冷媒の流通のみを許容する逆止弁（CV4-A）が設けられている。

一方、液配管（７）の第２分岐配管（7b）における接続管（１７ｂ）の接続位置と

２次側熱源熱交換器（１）との間には、第２タンク（T2）から２次側熱源熱交換器（１）への液冷媒の流通のみを許容する逆止弁（CV3-B）が設けられている。この第２分岐配管（7b）における接続管（１７b）の接続位置と室内熱交換器（3，3）との間には、室内熱交換器（3，3）から第２タンク（T2）への液冷媒の流通のみを許容する逆止弁（CV4-B）が設けられている。

各接続管（17a，17b）の２箇所の接続部（X，Y）のうち、２次側熱源熱交換器（１）側の接続部（Ｘ）と２次側熱源熱交換器（１）との間には第４電磁弁（SV-4）が、室内熱交換器（3，3，3）側の接続部（Ｙ）と室内熱交換器（3，3，3）との間には第５電磁弁（SV-5）がそれぞれ設けられている。

上記２次側熱源熱交換器（１）と第４電磁弁（SV-4）との間には、液冷媒回収管（ＬＲ１）の一端が接続されている。該液冷媒回収管（ＬＲ１）は、他端が逆止弁（CV4-B）の上流側に接続されると共に、第６電磁弁（SV-6）が設けられている。上記室内熱交換器（3，3，3）と第５電磁弁（SV-5）との間には、液冷媒供給管（ＬＳ１）の一端が接続されている。該液冷媒供給管（ＬＳ１）は、他端が逆止弁（CV3-A）の下流側に接続されると共に、第７電磁弁（SV-7）が設けられている。

上記各タンク（T1，T2）の上端部には、加減圧機構（18）が第１及び第２の加減圧管（19d，19e）により接続されている。この加減圧機構（18）は、一方のタンク（T1）に対して高圧を作用させている際には、他方のタンク（T2）に対して低圧を作用させ、この状態を交互に切換えるものである。詳しくは、第１及び第２の加減圧管（19d，19e）に第１及び第２の駆動用熱交換器（E1，E2）が接続されている。この駆動用熱交換器（E1，E2）は上記加熱用及び冷却用熱交換器（D3，D5）との間で熱交換を行うことによって各タンク（T1，T2）に対して冷媒循環用の圧力を作用させる。

つまり、第１タンク（T1）の上端部に接続されている第１加減圧管（１９d）は、第１駆動用熱交換器（E1）に繋がる第１分岐管（19d-A）と、第２駆動用熱交換器（E2）に繋がる第２分岐管（19d-B）とに分岐されている。第１分岐管（19d-A）には第８電磁弁（SV-8）が設けられ、第２分岐管（19d-B）には、第１タンク（T1）から第２駆動用熱交換器（E2）への冷媒の流通のみを許容する逆止弁（CV-5）と第９電磁弁（SV-9）とが設けられている。

一方、第２タンク（T2）の上端部に接続されている第２加減圧管（１９e）は、分

岐され、一方の第１分岐管（19e-A）は、第１０電磁弁（SV-10）を介して上記第１分岐管（19d-A）における第８電磁弁（SV-8）と第１駆動用熱交換器（E1）との間に接続されている。他方の第２分岐管（19e-B）は、逆止弁（CV-6）を介して上記第２分岐管（19d-B）における第９電磁弁（SV-9）と第２駆動用熱交換器（E2）との間にそれぞれ接続されている。

また、２次側冷媒回路（B）は、一対の貯留器（20, 21）を備え、第１の貯留器（20）の上端に接続する配管は分岐され、一方が第２分岐管（19d-B）における逆止弁（CV-5）と第９電磁弁（SV-9）との間に、他方が第１１電磁弁（SV-11）を介して第１分岐管（19d-A）における第８電磁弁（SV-8）と第１駆動用熱交換器（E1）との間にそれぞれ接続されている。この第１の貯留器（20）の下端に接続する配管は、分岐されて第１駆動用熱交換器（E1）及び第２駆動用熱交換器（E2）に接続され、それぞれ分岐管には、貯留器（20）から第１駆動用熱交換器（E1）への冷媒流のみを許容する逆止弁（CV-8）と、第２駆動用熱交換器（E2）から貯留器（20）への冷媒流のみを許容する逆止弁（CV-7）とが設けられている。

第２の貯留器（21）の上端に接続する配管は分岐され、一方が第１２電磁弁（SV-12）を介して第２分岐管（19d-B）における第９電磁弁（SV-9）と第２駆動用熱交換器（E2）との間に、他方が第１３電磁弁（SV-13）を介して第１分岐管（19d-A）における第８電磁弁（SV-8）と第１駆動用熱交換器（E1）との間にそれぞれ接続されている。この第２の貯留器（21）の下端に接続する配管も、分岐されて第１駆動用熱交換器（E1）及び第２駆動用熱交換器（E2）に接続され、それぞれ分岐管には、貯留器（21）から第１駆動用熱交換器（E1）への冷媒流のみを許容する逆止弁（CV-8）と、第２駆動用熱交換器（E2）から貯留器（21）への冷媒流のみを許容する逆止弁（CV-7）とが設けられている。

上記ガス配管（６）と第２分岐管（19d-B）との間には第１バイパス管（BPL-1）が接続されている。該第１バイパス管（BPL-1）は、第１４電磁弁（SV-14）を備え、ガス冷媒の一部をガス配管（６）にバイパスさせる。上記ガス配管（６）と第２加減圧管（19e）との間には第２バイパス管（BPL-2）が接続されている。該第２バイパス管（BPL-2）は、第１５電磁弁（SV-15）を備え、ガス冷媒の一部をガス配管（６）にバイパスさせる。上記液配管（７）と各駆動用熱交換器（E1, E2）との間には第３

バイパス管（BPL-3）が接続されている。該第３バイパス管（BPL-3）は、逆止弁（CV-9）を備え、各駆動用熱交換器（E1，E2）に液冷媒をバイパスさせる。

次に、冷房運転動作について説明する。

１次側冷媒回路（A ）では、四路切換弁（22）が実線側に切換えられ、第１電磁弁（SV-1）を開口し、第２及び第３電磁弁（SV-2，SV-3）を閉鎖する。そして、図２０に実線の矢印で示すように、圧縮機（11）から吐出した冷媒が、室外熱交換器（14）で凝縮した後、その一部が加熱用熱交換器（D3）に、他部が減圧弁（D4）により減圧されて冷却用熱交換器（D5）にそれぞれ供給される。この加熱用熱交換器（D3）に供給された冷媒は、第１駆動用熱交換器（E1）に熱を与えて過冷却状態となり、その後、加熱用熱交換器（12）において２次側熱源熱交換器（１）の冷媒から熱を奪って蒸発した後、圧縮機（11）に戻る。一方、冷却用熱交換器（D5）に供給された冷媒は、第２駆動用熱交換器（E2）から熱を奪って蒸発した後、圧縮機（11）に戻る。

この冷媒循環動作により、第１駆動用熱交換器（E1）では冷媒が蒸発して内部が高圧になる一方、第２駆動用熱交換器（E2）では冷媒が凝縮して内部が低圧になる。この状態で、第６，第７，第８，第１１，第１２電磁弁（SV-6,SV-7,SV-8,SV-11,SV-12）を開口し、第４，第５，第９，第１０，第１３電磁弁（SV-4,SV-5,SV-9,SV-10,SV-13）を閉鎖する。

この結果、第１タンク（T1）には高圧が、第２タンク（T2）には低圧がそれぞれ作用し、第１タンク（T1）からの液冷媒の排出動作と、第２タンク（T2）への液冷媒の回収動作とが同時に行われ、図２０の破線で示す矢印のように、２次側冷媒回路（B）の冷媒循環が行われる。また、第２タンク（T2）から第２駆動用熱交換器（E2）に回収されたガス冷媒は凝縮して第２の貯留器（21）に溜り、第１の貯留器（20）は第１駆動用熱交換器（E1）に均圧して該第１駆動用熱交換器（E1）に液冷媒を供給する。

この冷媒循環動作が所定時間行われた後、各電磁弁を切換え、第６，第７，第９，第１０，第１３電磁弁（SV-6,SV-7,SV-9,SV-10,SV-13）を開口し、第４，第５，第８，第１１，第１２電磁弁（SV-4,SV-5,SV-8,SV-11,SV-12）を閉鎖する。

この結果、第２タンク（T2）に高圧が、第１タンク（T1）に低圧がそれぞれ導入

されて、第２タンク（T2）からの液冷媒の排出動作と、第１タンク（T1）への液冷媒の回収動作とが同時に行われ、図２０の一点鎖線で示す矢印のように、２次側冷媒回路（B）の冷媒循環が行われる。また、第１タンク（T1）から第２駆動用熱交換器（E2）に回収されたガス冷媒は凝縮して第１の貯留器（20）に溜り、第２の貯留器（21）は第１駆動用熱交換器（E1）に均圧して該第１駆動用熱交換器（E1）に液冷媒を供給する。

このようにして液冷媒を排出するタンクと回収するタンクとが交互に切換えられ、室内の冷房運転が連続して行われる。

また、第２駆動用熱交換器（E2）において凝縮した液冷媒は、貯留器（20, 21）に回収されるので、第２駆動用熱交換器（E2）の熱交換面積を大きく確保することができ、冷却用熱交換器（D5）との間の熱交換量が増大されて装置全体としての性能が向上する。

一方、室内の暖房運転時には、１次側冷媒回路（A）では、四路切換弁（22）が破線側に切換えられ、第１電磁弁（SV-1）を閉鎖し、第２及び第３電磁弁（SV-2, SV-3）を開口する。図２１に実線の矢印で示すように、圧縮機（11）から吐出した冷媒の一部は１次側熱源熱交換器（12）に、他の冷媒は加熱用熱交換器（D3）にそれぞれ供給される。

１次側熱源熱交換器（12）に供給された冷媒は、２次側熱源熱交換器（1）との間で熱交換を行って凝縮した後、その一部が室外熱交換器（14）に、他部が冷却用熱交換器（D5）にそれぞれ供給される。そして、室外熱交換器（14）に供給された冷媒は、外気と熱交換を行って蒸発した後、圧縮機（11）に回収され、冷却用熱交換器（D5）に供給された冷媒は、第２駆動用熱交換器（E2）から熱を奪って蒸発した後、圧縮機（11）に戻る。

一方、加熱用熱交換器（D3）に供給された冷媒は、第１駆動用熱交換器（E1）に熱を与えて凝縮し、その後、１次側熱源熱交換器（12）において２次側熱源熱交換器（1）の冷媒に熱を与えて過冷却状態となった後、室外熱交換器（14）及び冷却用熱交換器（D5）で蒸発し、圧縮機（11）に戻る。

この冷媒循環動作により、第１駆動用熱交換器（E1）では冷媒が蒸発して内部が

高圧になる一方、第２駆動用熱交換器（E2）では冷媒が凝縮して内部が低圧になる。この状態で、第４，第５，第８，第１１，第１２電磁弁（SV-4, SV-5, SV-8, SV-11, SV-12）を開口し、第６，第７，第９，第１０，第１３電磁弁（SV-6, SV-7, SV-9, SV-10, SV-13）を閉鎖する。この結果、第１タンク（T1）には高圧が、第２タンク（T2）には低圧がそれぞれ作用し、第１タンク（T1）からの液冷媒の排出動作と、第２タンク（T2）への液冷媒の回収動作とが同時に行われ、図２１の破線で示す矢印のように、２次側冷媒回路（Ｂ）の冷媒循環が行われる。

この冷媒循環動作が所定時間行われた後、各電磁弁を切換え、第４，第５，第９，第１０，第１３電磁弁（SV-4, SV-5, SV-9, SV-10, SV-13）を開口し、れ、第６，第７，第８，第１１，第１２電磁弁（SV-6, SV-7, SV-8, SV-9, SV-11, SV-12）を閉鎖する。この結果、第２タンク（T2）に高圧が、第１タンク（T1）に低圧がそれぞれ作用し、第２タンク（T2）からの液冷媒の排出動作と、第１タンク（T1）への液冷媒の回収動作とが同時に行われ、図２１の一点鎖線で示す矢印のように、２次側冷媒回路（Ｂ）の冷媒循環が行われる。

この液冷媒を排出するタンクと回収するタンクとが交互に切換えられ、室内の暖房運転が連続して行われる。この暖房運転時においても、貯留器（20, 21）に液冷媒を回収することで、第２駆動用熱交換器（E2）と冷却用熱交換器（D5）との間の熱交換量が増大し、装置全体としての性能が向上する。

したがって、本実施形態においても、室内の冷房運転時、室外熱交換器（14）で凝縮した液冷媒を加熱用熱交換器（D3）において過冷却状態まで冷却することができるので、１次側熱源熱交換器（12）と２次側熱源熱交換器（１）との間の熱交換量を大きく確保することができて、装置全体としての性能の向上を図ることができる。

－第１１実施形態－

次に、本発明の第１１実施形態について図２２を用いて説明する。

本実施形態は、加減圧機構（18）として加圧回路（50）と減圧回路（60）とを備え、冷房専用の空気調和機に適用したものである。

先ず、２次側冷媒回路（Ｂ）について説明する。

この２次側冷媒回路（Ｂ）は、室内熱交換器（３）と２次側熱源熱交換器（１）とがガス配管（６）及び液配管（７）によって接続されている。

液配管（７）にはタンク（Ｔ）が接続されている。この液配管（７）におけるタンク（Ｔ）と２次側熱源熱交換器（１）との間には、該２次側熱源熱交換器（１）からタンク（Ｔ）への液冷媒の流通のみを許容する第１逆止弁（ＣＶ１）が設けられている。液配管（７）におけるタンク（Ｔ）と室内熱交換器（３）との間には、タンク（Ｔ）から室内熱交換器（３）への液冷媒の流通のみを許容する第２逆止弁（ＣＶ２）が設けられている。更に、液配管（７）における第２逆止弁（ＣＶ２）と室内熱交換器（３）との間には室内電動膨張弁（ＥＶ１）が設けられている。

上記タンク（Ｔ）には、加圧回路（50）及び減圧回路（60）が接続されている。先ず、加圧回路（50）について説明する。

この加圧回路（50）は循環用蒸発器（51）を備えている。この循環用蒸発器（51）はタンク（Ｔ）の設置位置よりも低い位置に設置されている。上記循環用蒸発器（51）は、ガス供給管（52）によりタンク（Ｔ）の上部に、また、液回収管（53）によりタンク（Ｔ）の下部にそれぞれ接続されている。

ガス供給管（52）には、タンク（Ｔ）内に高圧を作用させる際に開放する第１電磁弁（ＳＶ１）が設けられている。液回収管（53）には、タンク（Ｔ）から循環用蒸発器（51）への冷媒の流通のみを許容する第３逆止弁（ＣＶ３）が設けられている。

次に、減圧回路（60）について説明する。

この減圧回路（60）は循環用凝縮器（61）を備えている。この循環用凝縮器（61）はタンク（Ｔ）の設置位置よりも高い位置に設置されている。上記循環用凝縮器（61）は、ガス回収管（62）によりタンク（Ｔ）の上部に、また、液供給管（63）によりタンク（Ｔ）の下部にそれぞれ接続されている。

ガス回収管（62）には、タンク（Ｔ）内に低圧を作用させる際に開放する第２電磁弁（ＳＶ２）が設けられている。液供給管（63）には、循環用凝縮器（61）からタンク（Ｔ）への冷媒の流通のみを許容する第４逆止弁（ＣＶ４）が設けられている。

次に、この２次側冷媒回路（Ｂ）との間で熱交換を行う１次側冷媒回路（Ａ）に

ついて説明する。

この１次側冷媒回路（Ａ）は、圧縮機（11）、外気と熱交換を行う室外熱交換器（14）、上記循環用蒸発器（51）との間で熱交換可能な加熱熱交換器（71）、上記循環用凝縮器（61）との間で熱交換可能な冷却熱交換器（72）、２次側熱源熱交換器（１）との間で熱交換可能な１次側熱源熱交換器（12）が冷媒配管（16）によって接続されて構成されている。

詳しくは、圧縮機（11）の吐出側に室外熱交換器（14）及び加熱熱交換器（71）が順に接続され、この加熱熱交換器（71）の液側は第１分岐管（１６ａ）と第２分岐管（１６ｂ）とに分岐されている。該第１分岐管（１６ａ）は冷却熱交換器（72）に、第２分岐管（１６ｂ）は１次側熱源熱交換器（12）にそれぞれ接続されている。上記第１分岐管（１６ａ）には第１室外電動膨張弁（EV-A）が、第２分岐管（１６ｂ）には第２室外電動膨張弁（EV-B）がそれぞれ設けられている。上記冷却熱交換器（72）及び１次側熱源熱交換器（12）のガス側は合流して圧縮機（11）の吸入側に接続されている。

本実施形態では、循環用凝縮器（61）における凝縮温度が２次側熱源熱交換器（１）における凝縮温度よりも低く設定されている。具体的に、第１分岐管（１６ａ）と第２分岐管（１６ｂ）とは、配管径が異なり、第１分岐管（１６ａ）の流量が第２分岐管（１６ｂ）の流量よりも所定の比率だけ小さく設定されている。これに対し、冷却熱交換器（72）と循環用凝縮器（61）との熱交換面積は、１次側熱源熱交換器（12）と２次側熱源熱交換器（１）の熱交換面積よりも小さく設定され、その比率は上記所定の比率よりも小さく設定されている。

つまり、例えば、第１分岐管（１６ａ）の流量と第２分岐管（１６ｂ）の流量との比が１：１０である場合、冷却熱交換器（72）と循環用凝縮器（61）との熱交換面積と、１次側熱源熱交換器（12）と２次側熱源熱交換器（１）の熱交換面積の比は２：１０に設定されている。これにより、冷媒流量に対する熱交換器としての能力は、循環用凝縮器（61）の方が２次側熱源熱交換器（１）よりも高く設定されている。この結果、循環用凝縮器（61）の凝縮温度は２次側熱源熱交換器（１）の凝縮温度よりも低くなる。

次に、本実施形態における室内の冷房運転動作について説明する。

冷房運転時、１次側冷媒回路（Ａ）においては、圧縮機（11）を駆動し、図２２に実線で示す矢印のように、圧縮機（11）から吐出した高温高圧のガス冷媒が、室外熱交換器（14）及び加熱熱交換器（71）を順に流れ、外気及び循環用蒸発器（51）内の冷媒と熱交換を行って凝縮する。これにより、循環用蒸発器（51）の冷媒に熱を与える。その後、この液冷媒は、各分岐管（16a，16b）に分流し、各室外電動膨張弁（EV-A，EV-B）によって減圧された後、冷却熱交換器（72）及び１次側熱源熱交換器（12）に流れる。ここで、液冷媒は、循環用凝縮器（61）及び２次側熱源熱交換器（１）の冷媒と熱交換を行って蒸発し、循環用凝縮器（61）及び２次側熱源熱交換器（１）の冷媒から熱を奪う。その後、これら冷却熱交換器（72）及び１次側熱源熱交換器（12）を流れたガス冷媒は合流して圧縮機（11）に戻る。この循環動作を繰り返す。

一方、２次側冷媒回路（Ｂ）にあっては、上述した熱交換動作によって、循環用蒸発器（51）では冷媒の蒸発動作が、循環用凝縮器（61）では冷媒の凝縮動作がそれぞれ行われる。そして、循環用蒸発器（51）では高圧が、循環用凝縮器（61）では低圧がそれぞれ発生する。

この状態において、先ず、第１電磁弁（ＳＶ1）を開口すると共に、第２電磁弁（ＳＶ2）を閉鎖する。これにより、循環用蒸発器（51）の高圧がガス供給管（52）によってタンク（Ｔ）内に作用し、該タンク（Ｔ）の液冷媒の液面が押し下げられ、図２２に破線の矢印で示すように、該液冷媒が液配管（７）に押し出される。この押し出された液冷媒は、液配管（７）を室内熱交換器（３）に向って流れ、室内電動膨張弁（ＥＶ1）によって減圧された後、室内熱交換器（３）において室内空気と熱交換を行い、蒸発して室内空気を冷却する。この蒸発したガス冷媒は、ガス配管（６）を介して２次側熱源熱交換器（１）に流れ、１次側熱源熱交換器（12）との間で熱交換を行って凝縮する。

この動作の後、第１電磁弁（ＳＶ1）を閉鎖すると共に、第２電磁弁（ＳＶ2）を開口する。これにより、循環用凝縮器（61）の低圧がガス回収管（62）によってタンク（Ｔ）内に作用する。この循環用凝縮器（61）の凝縮温度は２次側熱源熱交換器（１）の凝縮温度よりも低く、循環用凝縮器（61）の内圧が２次側熱源熱交換器（１）の内圧よりも低い。このため、タンク（Ｔ）の内圧が２次側熱源熱交換器（１）の内圧よ

りも低くなり、図２２に一点鎖線の矢印で示すように、２次側熱源熱交換器（１）の液冷媒が、液配管（７）を経てタンク（Ｔ）内に回収される。

この際、タンク（Ｔ）内上層部分のガス冷媒が循環用凝縮器（61）に吸引された後、凝縮して液冷媒となり、液供給管（63）によってタンク（Ｔ）内に回収される。この状態から上述した加圧回路（50）による加圧動作に移ると、加圧回路（50）全体が均圧されることによってタンク（Ｔ）の液冷媒の一部は循環用蒸発器（51）に回収され、高圧発生用の冷媒として利用される。

以上のような加圧回路（50）による加圧動作と減圧回路（60）による減圧動作とを交互に繰り返し、加圧動作時にはタンク（Ｔ）から液冷媒が押し出され、減圧動作時にはタンク（Ｔ）に液冷媒が回収され、２次側冷媒回路（Ｂ）において冷媒が循環し、室内を冷房する。

したがって、本実施形態によれば、タンク（Ｔ）内に発生する吸引力によって２次側熱源熱交換器（１）の液冷媒を回収するようにしているので、タンク（Ｔ）を２次側熱源熱交換器（１）よりも低い位置に設置する必要がなくなり、機器の配設位置の制約が小さくできて汎用性の向上を図ることができる。

－第１２実施形態－

次に、本発明の第１２実施形態について説明する。尚、ここでは上述した第１１実施形態との相違点について説明する。

本実施形態の冷媒回路は、暖房専用の空気調和機に適用したものであって、１次側冷媒回路（Ａ）の構成及び液配管（７）に設けられている逆止弁の構成が上述した第１１実施形態のものと異なっている。

つまり、図２３に示すように、１次側冷媒回路（Ａ）における圧縮機（11）の吐出側には、加熱熱交換器（71）及び１次側熱源熱交換器（12）が順に接続され、この１次側熱源熱交換器（12）の液側が第１分岐管（１６ｃ）と第２分岐管（１６ｄ）とに分岐されている。該第１分岐管（１６ｃ）は室外熱交換器（14）に、第２分岐管（１６ｄ）は冷却熱交換器（72）にそれぞれ接続されている。第１分岐管（１６ｃ）には第１室外電動膨張弁（EV-C）が、第２分岐管（１６ｄ）には第２室外電動膨張弁（EV-D）がそれ

ぞれ設けられている。上記室外熱交換器（14）及び冷却熱交換器（72）のガス側は合流して圧縮機（11）の吸入側に接続されている。

一方、２次側冷媒回路（Ｂ）における液配管（７）のタンク（Ｔ）と２次側熱源熱交換器（１）との間には、タンク（Ｔ）から２次側熱源熱交換器（１）への液冷媒の流通のみを許容する第１逆止弁（ＣＶ３）が設けられている。この液配管（７）におけるタンク（Ｔ）と室内熱交換器（３）との間には、室内熱交換器（３）からタンク（Ｔ）への液冷媒の流通のみを許容する第２逆止弁（ＣＶ４）が設けられている。

尚、本実施形態の加圧回路（50）及び減圧回路（60）は上述した第１１実施形態のものと同様である。

このように循環用蒸発器（51）との間で熱交換する加熱熱交換器（71）と、２次側熱源熱交換器（１）との間で熱交換する１次側熱源熱交換器（12）とを直列に接続したことにより、循環用蒸発器（51）における蒸発温度が２次側熱源熱交換器（１）における蒸発温度よりも高くなる。つまり、この蒸発温度の差によって循環用蒸発器（51）の内部圧力が２次側熱源熱交換器（１）の内部圧力よりも高くなる。

次に、本実施形態における室内の暖房運転動作について説明する。

暖房運転時、１次側冷媒回路（Ａ）においては、圧縮機（11）を駆動し、図２３に実線で示す矢印のように、圧縮機（11）から吐出した高温高圧のガス冷媒が、加熱熱交換器（71）及び１次側熱源熱交換器（12）を順に流れ、循環用蒸発器（51）及び２次側熱源熱交換器（１）の冷媒と熱交換を行って凝縮する。これにより、循環用蒸発器（51）及び２次側熱源熱交換器（１）の冷媒に熱を与える。その後、この液冷媒は、各分岐管（16c, 16d）に分流し、各室外電動膨張弁（EV-C, EV-D）によって減圧された後、冷却熱交換器（72）及び室外熱交換器（14）に流れる。この冷却熱交換器（72）では循環用凝縮器（61）との間で、室外熱交換器（14）では外気とそれぞれ熱交換を行って蒸発する。つまり、循環用凝縮器（61）の冷媒から熱を奪う。その後、これら冷却熱交換器（72）及び室外熱交換器（14）を流れたガス冷媒は合流して圧縮機（11）に戻る。この循環動作を繰り返す。

一方、２次側冷媒回路（Ｂ）にあっては、上述した第１１実施形態の場合と同様に、加圧回路（50）及び減圧回路（60）の各電磁弁（SV1, SV2）の切換え動作が行わ

れ、タンク（T）内に高圧を作用させる状態と低圧を作用させる状態とに交互に切換わる。このタンク（T）に高圧が作用する状態では、上述したように、循環用蒸発器（51）の蒸発温度が２次側熱源熱交換器（１）の蒸発温度よりも高いので、循環用蒸発器（51）の内圧が２次側熱源熱交換器（１）の内圧よりも高くなる。このため、タンク（T）が高圧になり、図２３に破線の矢印で示すように、タンク（T）から押し出された液冷媒が、２次側熱源熱交換器（１）において１次側熱源熱交換器（12）との間で熱交換を行い蒸発する。この蒸発したガス冷媒は、ガス配管（６）を介して室内熱交換器（３）に流れ、室内空気と熱交換を行い、凝縮して該室内を加熱する。

上記タンク（T）内に低圧が作用する状態では、図２３に一点鎖線の矢印で示すように、室内熱交換器（３）の液冷媒が液配管（７）を経てタンク（T）内に回収される。この動作においても、減圧回路（60）の減圧動作時には、循環用凝縮器（61）に吸引されたガス冷媒は凝縮して液冷媒となり、液供給管（63）によってタンク（T）に回収される。また、加圧回路（50）の加圧動作時には、タンク（T）内の液冷媒の一部が循環用蒸発器（51）に回収され、高圧発生用の冷媒として利用される。この動作を繰り返し、室内を暖房する。

したがって、本実施形態によれば、タンク（T）内に発生する吸引力によって室内熱交換器（３）の液冷媒を回収するようにしているので、タンク（T）を室内熱交換器（３）よりも低い位置に設置する必要がなくなり、機器の配設位置の制約が小さくできて汎用性の向上を図ることができる。

－第１３実施形態－

次に、本発明の第１３実施形態について説明する。尚、ここでも上述した第１１実施形態との相違点について説明する。

本実施形態の冷媒回路は、冷房専用の空気調和機に適用したものであり、１次側冷媒回路（A）は上述した第１１実施形態のものと同様であるので、ここでは説明を省略する。

そして、本実施形態の２次側冷媒回路（B）の特徴とするところは、上述した第５実施実施形態のものと同様に、２個のタンク（T1, T2）を備え、加圧回路（50）及

び減圧回路（60）に対して各タンク（T1，T2）が互いに並列に接続されている。

詳しくは、図２４に示すように、上記加圧回路（50）のガス供給管（52）は分岐管（52a，52b）に分岐されて各タンク（T1，T2）の上端部にそれぞれ接続され、各分岐管（52a，52b）には電磁弁（SV1-1，SV1-2）が設けられている。加圧回路（50）の液回収管（53）は分岐管（53a，53b）に分岐されて各タンク（T1，T2）の下端部にそれぞれ接続され、各分岐管（53a，53b）には逆止弁（CV3-1，CV3-2）が設けられている。

一方、上記減圧回路（60）のガス回収管（62）は各分岐管（62a，62b）に分岐されて各タンク（T1，T2）の上端部にそれぞれ接続され、各分岐管（62a，62b）には電磁弁（SV2-1，SV2-2）が設けられている。減圧回路（60）の液供給管（63）は分岐管（63a，63b）に分岐されて各タンク（T1，T2）の下端部にそれぞれ接続され、各分岐管（63a，63b）には逆止弁（CV4-1，CV4-2）が設けられている。

上記２次側熱源熱交換器（１）に繋がる液配管（７）は分岐液配管（7a，7b）に分岐されて各タンク（T1，T2）の下端部にそれぞれ接続され、各分岐液配管（7a，7b）には逆止弁（CV1-1，CV1-2）がそれぞれ設けられている。室内熱交換器（３）に繋がる液配管（７）は分岐液配管（7c，7d）に分岐されて各タンク（T1，T2）の下端部にそれぞれ接続され、各分岐液配管（7c，7d）には逆止弁（CV2-1，CV2-2）がそれぞれ設けられている。

本実施形態における冷房運転時の２次側冷媒回路（Ｂ）の動作は、加圧回路（50）の各分岐管（52a，52b）に設けられた電磁弁（SV1-1，SV1-2）のうち一方が開口し、他方が閉鎖する動作が交互に繰り返される。また、減圧回路（60）の各分岐管（62a，62b）に設けられた電磁弁（SV2-1，SV2-2）のうち一方が開口し他方が閉鎖する動作が交互に繰り返される。これにより、一方のタンクが室内熱交換器（３）に向かって液冷媒を押し出し、他方のタンクが２次側熱源熱交換器（１）から液冷媒を回収する動作を繰り返す。

具体的には、加圧回路（50）の分岐管（５２ａ）の電磁弁（SV1-1）を開口し、且つ減圧回路（60）の分岐管（６２ｂ）の電磁弁（SV2-2）を開口した場合には、図２４に実線で示すように、上側のタンク（T1）が室内熱交換器（３）に向かって液冷媒を

押し出し、下側のタンク（T2）が２次側熱源熱交換器（１）から液冷媒を回収する。逆に、加圧回路（50）の分岐管（５２b）の電磁弁（SV1-2）を開口し、且つ減圧回路（60）の分岐管（６２a）の電磁弁（SV2-1）を開口した場合には、図２４に破線の矢印で示すように、下側のタンク（T2）が室内熱交換器（３）に向かって液冷媒を押し出し、上側のタンク（T1）が２次側熱源熱交換器（１）から液冷媒を回収する。この動作を交互に繰り返し、室内の冷房が連続的に行われる。

－第１４実施形態－

次に、本発明の第１４実施形態について説明する。尚、ここでは上述した第１３実施形態との相違点について説明する。

本実施形態の冷媒回路は、暖房専用の空気調和機に適用したものであり、１次側冷媒回路（A）は、上述した第１３実施形態のものと同様であるので、ここでは説明を省略する。また、２次側冷媒回路（B）は、液配管（７）に設けられている逆止弁の構成が上述した第１３実施形態のものと異なっている。

つまり、図２５に示すように、液配管（７）に設けられた逆止弁（CV1-1, CV1-2, CV2-1, CV2-2）として冷媒流通許容方向が異なるものが採用されている。

したがって、暖房運転時において、上述した第１３実施形態の場合と同様に、加圧回路（50）の各分岐管（52a, 52b）に設けられた電磁弁（SV1-1, SV1-2）及び減圧回路（60）の各分岐管（62a, 62b）に設けられた電磁弁（SV2-1, SV2-2）の開閉動作が交互に繰り返される。この結果、一方のタンクが２次側熱源熱交換器（１）に向かって液冷媒を押し出し、他方のタンクが室内熱交換器（３）から液冷媒を回収する動作を交互に繰り返し（図２５の実線の矢印で示す状態と破線の矢印で示す状態とが交互に繰り返される）、室内の暖房が連続的に行われる。

－第１５実施形態－

次に、本発明の第１５実施形態について説明する。尚、ここでは上述した第１１実施形態との相違点について説明する。

本実施形態の冷媒回路は、冷房専用の空気調和機に適用したものであり、１次側冷媒回路（A）は上述した第１１実施形態のものと同様であるので、ここでは説明を

省略する。

本実施形態の２次側冷媒回路（Ｂ）の特徴は、図２６に示すように、上述したタンク（以下メインタンクと呼ぶ）（Ｔ）とは別に、小型のサブタンク（ST）を備え、該サブタンク（ST）に液冷媒を一時的に貯留するようにしたことである。

以下、本実施形態の２次側冷媒回路（Ｂ）の回路構成について説明する。

循環用凝縮器（61）に繋がるガス回収管（62）は、分岐管（62a，62b）に分岐され、一方が第２電磁弁（ＳＶ2）を介してメインタンク（Ｔ）の上端部に、他方が第３電磁弁（ＳＶ3）を介して上記サブタンク（ST）の上端部にそれぞれ接続されている。

循環用蒸発器（51）に繋がるガス供給管（52）は、分岐管（52a，52b）に分岐され、一方が第１電磁弁（ＳＶ1）を介してメインタンク（Ｔ）の上端部に、他方が第４電磁弁（ＳＶ4）を介して上記分岐管（６２ｂ）にそれぞれ接続されている。

上記サブタンク（ST）は循環用蒸発器（51）よりも高い位置に設置され、この循環用蒸発器（51）の下端とサブタンク（ST）の下部とは液回収管（53）によって接続されている。この液回収管（53）には、サブタンク（ST）から循環用蒸発器（51）に向かう冷媒の流通のみを許容する逆止弁（ＣＶ3）が設けられている。

上記液回収管（53）と液配管（７）とは、液配管（７）から液回収管（53）への液冷媒の流通のみを許容する逆止弁（ＣＶ5）を備えた液吸引管（54）によって接続されている。液吸引管（54）は、一端が、液回収管（53）のサブタンク（ST）と逆止弁（ＣＶ3）との間に、他端が、液配管（７）の室内電動膨張弁（ＥＶ1）と逆止弁（ＣＶ2）との間にそれぞれ接続されている。このようにしてサブタンク（ST）に対する圧力作用状態を切換える切換え手段（Ｉ）が構成されている。その他の構成は、上述した第１１実施形態のものと同様である。

次に、本実施形態における室内の冷房運転時について説明する。

１次側冷媒回路（Ａ）では上述した第１１実施形態と同様の動作が行われる。

２次側冷媒回路（Ｂ）にあっては、先ず、第１及び第４電磁弁（SV1，SV4）を開口する一方、第２及び第３電磁弁（SV2，SV3）を閉鎖する。これにより、循環用蒸発器（51）の高圧がメインタンク（Ｔ）に作用すると共に、この循環用蒸発器（51）とサブタンク（ST）とが均圧される。この結果、図２６に実線の矢印で示すように、メ

インタンク（T）から押し出された液冷媒が、室内熱交換器（3）において蒸発した後、2次側熱源熱交換器（1）で凝縮する。また、サブタンク（ST）の液冷媒が液回収管（53）により循環用蒸発器（51）に落下供給される。

その後、電磁弁の開閉動作を切換え、第1及び第4電磁弁（SV1, SV4）を閉鎖する一方、第2及び第3電磁弁（SV2, SV3）を開口する。これにより、循環用凝縮器（61）の低圧がガス回収管（62）の各分岐管（62a, 62b）によってメインタンク（T）及びサブタンク（ST）に作用する。この結果、図26に破線の矢印で示すように、2次側熱源熱交換器（1）の液冷媒が液配管（7）を経てメインタンク（T）に回収される。また、サブタンク（ST）内も低圧状態になっているため、液配管（7）内の液冷媒の一部は、液吸引管（54）を経てサブタンク（ST）に回収される。このサブタンク（ST）に回収された液冷媒は、再び電磁弁が切換えられてサブタンク（ST）と循環用蒸発器（51）とが均圧された際に循環用蒸発器（51）に供給されて駆動用の冷媒となる。この動作を繰り返し、室内を冷房する。

したがって、本実施形態によれば、サブタンク（ST）に液冷媒を貯留し、この液冷媒を循環用蒸発器（51）に供給するようにしたために、上述した各実施形態のように、メインタンク（T）を循環用蒸発器（51）よりも上方に配置する必要がなくなり、このメインタンク（T）及び循環用蒸発器（51）の設置位置の自由度の向上を図ることができる。

尚、このようなサブタンクを備えた冷媒回路を暖房専用の空気調和装置に適用する場合には、液配管（7）に設けられた逆止弁（CV1, CV2）として冷媒流通許容方向の異なるものを採用することになる。

－第16実施形態－

次に、本発明の第16実施形態について説明する。本実施形態の2次側冷媒回路（B）の特徴は、上述した第15実施形態と同様に2個のサブタンク（ST1, ST2）を備えていることである。

この回路構成について説明すると、図27に示すように、各サブタンク（ST1, ST2）が、循環用凝縮器（61）及び循環用蒸発器（51）において並列に接続されている。

つまり、各サブタンク（ST1, ST2）はそれぞれガス回収管（62b-1, 62b-2）によって循環用凝縮器（61）に、ガス供給管（52b-1, 52b-2）によって循環用蒸発器（51）に接続されている。各ガス回収管（62b-1, 62b-2）には、それぞれ電磁弁（SV3-A, SV3-B）が設けられ、各ガス供給管（52b-1, 52b-2）には、それぞれ電磁弁（SV4-A, SV4-B）が設けられている。各サブタンク（ST1, ST2）の下端には、循環用蒸発器（51）に接続する液回収管（53-1, 53-2）に対応して液吸引管（54-1, 54-2）が設けられている。

このようにして各サブタンク（ST1, ST2）に対する圧力作用状態を切換える切換え手段（I）が構成されている。その他の構成は、上述した第１１実施形態のものと同様である。その他の構成は、上述した第１５実施形態のものと略同様である。

次に、本実施形態における室内の冷房運転時について説明する。

１次側冷媒回路（A）では上述した第１１実施形態と同様の動作が行われる。

２次側冷媒回路（B）にあっては、２個のサブタンク（ST1, ST2）のうち一方が循環用凝縮器（61）に、他方が循環用蒸発器（51）にそれぞれ接続された状態が交互に繰り返される。つまり、ガス回収管（６２b）の一方の電磁弁（SV3-A）を開口し、他方の電磁弁（SV3-B）を閉鎖し、且つガス供給管（５２b）の一方の電磁弁（SV4-B）を開口し、他方の電磁弁（SV4-A）を閉鎖する。この結果、図２７に実線の矢印で示すように、一方のサブタンク（ＳＴ１）が、循環用凝縮器（61）に連通して液配管（７）の液冷媒の一部を回収し、且つ他方のサブタンク（ＳＴ２）が、循環用蒸発器（51）に連通して該循環用蒸発器（51）に液冷媒を落下供給する状態となる。

これら各電磁弁を切換えると、図２７に破線の矢印で示すように、他方のサブタンク（ＳＴ２）が液配管（７）の液冷媒の一部を回収し、且つ一方のサブタンク（ＳＴ１）が循環用蒸発器（51）に液冷媒を落下供給する状態となる。この状態が交互に繰り返される。

したがって、一方のサブタンクに液冷媒を回収する動作と、他方のサブタンクから循環用蒸発器（51）に液冷媒を供給する動作とが同時に行われるので、上述した第１５実施形態のような１個のサブタンク（ST）を備えさせた場合に比べて、サブタンク（ST1, ST2）の動作を切換えるための電磁弁の開閉頻度が低減され、その耐久性の

向上が図れることになる。

尚、本実施形態の場合にも暖房専用の空気調和装置に適用する場合には、液配管（７）に設けられた逆止弁（CV1, CV2）として冷媒流通許容方向の異なるものを採用する。

－第１７実施形態－

次に、本発明の第１７実施形態について説明する。尚、ここでは上述した第１１実施形態との相違点について説明する。

本実施形態の冷媒回路はヒートポンプの空気調和装置に適用したものである。

先ず、１次側冷媒回路（Ａ）について説明する。

図２８に示すように、１次側冷媒回路（Ａ）は、１次側熱源熱交換器（12）によって２次側熱源熱交換器（１）を加熱及び冷却する。具体的に、圧縮機（11）、四路切換弁（22）、室外熱交換器（14）、加熱熱交換器（71）、室外電動膨張弁（EV-A, EV-B）、冷却熱交換器（72）、１次側熱源熱交換器（12）が冷媒配管（16）によって接続されている。詳しくは、加熱熱交換器（71）に接続する配管が分岐され、一方が第１室外電動膨張弁（EV-A）を介して冷却熱交換器（72）に、他方が第２室外電動膨張弁（EV-B）を介して１次側熱源熱交換器（12）にそれぞれ接続されている。

冷却熱交換器（72）のガス側は、圧縮機（11）の吸込側に接続され、１次側熱源熱交換器（12）のガス側は、四路切換弁（22）を介して圧縮機（11）の吸入側と吐出側とに切換わる。上記室外熱交換器（14）と加熱熱交換器（71）とを接続する配管には、室外熱交換器（14）から加熱熱交換器（71）への冷媒の流通のみを許容する逆止弁（ＣＶ6）が、１次側熱源熱交換器（12）と四路切換弁（22）とを接続する配管には、１次側熱源熱交換器（12）から四路切換弁（22）への冷媒の流通のみを許容する逆止弁（ＣＶ7）がそれぞれ設けられている。

上記第２室外電動膨張弁（EV-B）と加熱熱交換器（71）との間には第５電磁弁（ＳＶ5）が設けられている。この電磁弁（ＳＶ5）及び加熱熱交換器（71）の間と、１次側熱源熱交換器（12）及び逆止弁（ＣＶ7）の間とはガス冷媒バイパス管（ＧＢＬ）によって接続されている。このガス冷媒バイパス管（ＧＢＬ）には、第６電磁弁（ＳＶ6）と、加熱熱交換器（71）から１次側熱源熱交換器（12）への冷媒の流通のみを許容す

る逆止弁（ＣＶ８）とが設けられている。上記逆止弁（ＣＶ７）及び四路切換弁（22）の間と、逆止弁（ＣＶ６）及び加熱熱交換器（71）の間とは吐出ガスバイパス管(OGL)によって接続され、この吐出ガスバイパス管（ＯＧＬ）には、加熱熱交換器（71）へ向かう吐出ガスの流通のみを許容する逆止弁（ＣＶ９）が設けられている。

次に、２次側冷媒回路（Ｂ）について説明する。尚、ここでは、上述した第１１実施形態で述べた２次側冷媒回路（Ｂ）との相違点について説明する。

図２８に示すように、２次側冷媒回路（Ｂ）は、液配管（７）における第４逆止弁（ＣＶ４）と室内熱交換器（３）との間に、室内の冷房運転時に開口し、暖房運転時に閉鎖する第７電磁弁（ＳＶ７）が設けられ、液配管（７）における第３逆止弁（ＣＶ３）と２次側熱源熱交換器（１）との間に、室内の冷房運転時に開口し、暖房運転時に閉鎖する第８電磁弁（ＳＶ８）が設けられている。

上記液配管（７）における第７電磁弁（ＳＶ７）と室内熱交換器（３）との間には回収側暖房液配管（34）の一端が接続され、該回収側暖房液配管（34）の他端は、液配管（７）における第３逆止弁（ＣＶ３）と第８電磁弁（ＳＶ８）との間に接続されている。この回収側暖房液配管（34）には、暖房運転時に開口し、冷房運転時に閉鎖する第９電磁弁（ＳＶ９）が設けられている。

上記液配管（７）における第４逆止弁（ＣＶ４）と第７電磁弁（ＳＶ７）との間には供給側暖房液配管（35）の一端が接続され、該供給側暖房液配管（35）の他端は、液配管（７）における第８電磁弁（ＳＶ８）と２次側熱源熱交換器（１）との間に接続されている。この供給側暖房液配管（35）には、暖房運転時に開口し、冷房運転時に閉鎖する第１０電磁弁（SV10）が設けられている。その他の構成は、上述した第１１実施形態のものと略同様である。

以下、室内の冷房及び暖房運転動作について説明する。

冷房運転時には、先ず、１次側冷媒回路（Ａ）では、四路切換弁（22）が実線側に切換えられ、第５電磁弁（ＳＶ５）を開口し、第６電磁弁（ＳＶ６）を閉鎖する。一方、２次側冷媒回路（Ｂ）では、第７電磁弁（ＳＶ７）及び第８電磁弁（ＳＶ８）を開口し、第９電磁弁（ＳＶ９）及び第１０電磁弁（SV10）を閉鎖する。

この状態で、１次側冷媒回路（Ａ）にあっては、図２８に実線の矢印で示す如く、圧縮機（11）から吐出した高温高圧のガス冷媒が、室外熱交換器（14）及び加熱熱交換器（71）において凝縮する。その後、この冷媒は、冷却熱交換器（72）及び１次側熱源熱交換器（12）に分流し、各室外電動膨張弁（EV-A, EV-B）において減圧された後、冷却熱交換器（72）においては循環用凝縮器（61）の冷媒と熱交換を行い、１次側熱源熱交換器（12）においては２次側熱源熱交換器（１）の冷媒と熱交換を行って蒸発する。これら蒸発した冷媒は圧縮機（11）に戻る。この循環動作を繰り返す。

一方、２次側冷媒回路（Ｂ）にあっては、上述と同様にして、タンク（Ｔ）に対する加圧動作及び減圧動作が繰り返される。つまり、加圧動作によって、タンク（Ｔ）から押し出された液冷媒は、図２８に実線の矢印で示すように、室内電動膨張弁（EV１）によって減圧された後、室内熱交換器（３）において室内空気と熱交換を行い、蒸発して室内空気を冷却し、その後、２次側熱源熱交換器（１）において凝縮する。そして、減圧動作によって、この凝縮した液冷媒がタンク（Ｔ）に回収される。この加圧及び減圧を繰り返し、２次側冷媒回路（Ｂ）では冷媒が循環し、室内を冷房する。

次に、室内の暖房運転時について説明する。

暖房運転時には、先ず、１次側冷媒回路（Ａ）では、四路切換弁（22）が破線側に切換えられ、第６電磁弁（ＳＶ6）を開口し、第５電磁弁（ＳＶ5）を閉鎖する。一方、２次側冷媒回路（Ｂ）では、第７電磁弁（ＳＶ7）及び第８電磁弁（ＳＶ8）を閉鎖し、第９電磁弁（ＳＶ9）及び第１０電磁弁（SV10）を開口する。

この状態で、１次側冷媒回路（Ａ）にあっては、図２８に破線の矢印で示す如く、圧縮機（11）から吐出した高温高圧のガス冷媒が、加熱熱交換器（71）において循環用蒸発器（51）の冷媒と熱交換を行って顕熱変化する。その後、このガス冷媒は、ガス冷媒バイパス管（ＧＢＬ）を経て１次側熱源熱交換器（12）に流れ、ここで２次側熱源熱交換器（１）の冷媒との間で熱交換を行って凝縮する。この凝縮した液冷媒は、一部が室外熱交換器（14）で蒸発し、四路切換弁（22）を経て圧縮機（11）に戻り、他が第１室外電動膨張弁（EV-A）において減圧された後、冷却熱交換器（72）において循環用凝縮器（61）の冷媒と熱交換を行って蒸発し、圧縮機（11）に戻る。この循

環動作を繰り返す。

一方、２次側冷媒回路（Ｂ）にあっては、上述と同様にして、タンク（Ｔ）に対する加圧動作及び減圧動作が繰り返される。つまり、加圧動作によって、タンク（Ｔ）から押し出された液冷媒は、図２８に破線の矢印で示すように、供給側暖房液配管（35）を経て２次側熱源熱交換器（１）に流れて蒸発し、その後、室内熱交換器（３）において室内空気と熱交換を行い、凝縮して室内空気を加温する。そして、減圧動作によって、この凝縮した液冷媒が回収側暖房液配管（34）を経てタンク（Ｔ）に回収される。この加圧及び減圧を繰り返し、２次側冷媒回路（Ｂ）では冷媒が循環し、室内を暖房する。

－第１８実施形態－

次に、本発明の第１８実施形態について説明する。尚、本実施形態もヒートポンプの空気調和装置に適用したものであり、１次側冷媒回路（Ａ）は、上述した第１７実施形態のものと同様であるので、ここでは説明を省略する。また、２次側冷媒回路（Ｂ）は、第１７実施形態との相違点についてのみ説明する。

図２９に示すように、２次側冷媒回路（Ｂ）は、液配管（７）に四路切換弁（10）を備えている。詳しくは、この四路切換弁（10）は、２次側熱源熱交換器（１）の液側から延びる第１の液配管（7A）と、タンク（Ｔ）から延びる第２及び第３の液配管（7B, 7C）と、室内熱交換器（３）の液側から延びる第４の液配管（7D）とがそれぞれ接続されている。これにより、タンク（Ｔ）から押し出された冷媒の２次側熱源熱交換器（１）及び室内熱交換器（３）への供給状態が切換わる。

以下、室内の冷房及び暖房運転時について説明する。

先ず、冷房運転時には、上述した第１７実施形態の場合と同様に、１次側冷媒回路（Ａ）では四路切換弁（22）が実線側に切換えられる。一方、２次側冷媒回路（Ｂ）においても四路切換弁（10）が実線側に切換えられる。この状態で、加圧回路（50）及び減圧回路（60）による加圧及び減圧が繰り返され、図２９に実線の矢印で示すように、２次側冷媒回路（Ｂ）では冷媒が循環し、室内を冷房する。

一方、室内の暖房運転時には、各四路切換弁（22, 10）が共に破線側に切換わり、

この状態で、加圧回路（50）及び減圧回路（60）による加圧及び減圧が繰り返され、２次側冷媒回路（Ｂ）では、図２９に破線の矢印で示すように、上記冷房運転とは逆向きに冷媒が循環し、室内を暖房する。

－第１９実施形態－

次に、本発明の第１９実施形態について図３０～図３２を用いて説明する。尚、本実施形態もヒートポンプの空気調和装置に適用したものである。

先ず、１次側冷媒回路（Ａ）は、上述した第１７実施形態のものと同様に、圧縮機（11）、四路切換弁（22）、室外熱交換器（14）、加熱熱交換器（71）、第１及び第２の室外電動膨張弁（EV-A, EV-B）、冷却熱交換器（72）、１次側熱源熱交換器（12）が冷媒配管（16）によって接続されて構成されている。

詳しくは、室外熱交換器（14）のガス側は、四路切換弁（22）によって圧縮機（11）の吸入側と吐出側とに切換わる。この室外熱交換器（14）は、冷房ガス供給管（ＣＧＬ）によって加熱熱交換器（71）に接続されている。この冷房ガス供給管（ＣＧＬ）には、加熱熱交換器（71）に向かう冷媒の流通のみを許容する逆止弁（ＣＶ1）が設けられている。上記加熱熱交換器（71）の液側は第１電磁弁（ＳＶ1）を介して第１及び第２の液分岐管（LSL-1, LSL-2）に分岐され、第１分岐管（LSL-1）は第１室外電動膨張弁（EV-A）を介して冷却熱交換器（72）に、第２分岐管（LSL-2）は第２室外電動膨張弁（EV-B）を介して１次側熱源熱交換器（12）にそれぞれ接続されている。

上記冷却熱交換器（72）のガス側は圧縮機（11）の吸入側に接続されている。上記１次側熱源熱交換器（12）のガス側は、逆止弁（ＣＶ2）を介して四路切換弁（22）に接続され、この四路切換弁（22）によって圧縮機（11）の吸入側と吐出側とに切換わる。

上記第２分岐管（LSL-2）と室外熱交換器（14）とは暖房液配管（ＷＬＬ）によって接続され、この暖房液配管（ＷＬＬ）には、室外熱交換器（14）へ向かう冷媒の流通のみを許容する逆止弁（ＣＶ3）が設けられている。

上記加熱熱交換器（71）及び第１電磁弁（ＳＶ1）の間と、１次側熱源熱交換器（12）及び逆止弁（ＣＶ2）の間は、暖房ガス供給管（ＷＧＬ）によって接続されている。この暖房ガス供給管（ＷＧＬ）には、１次側熱源熱交換器（12）への冷媒供給のみを許

容する逆止弁（ＣＶ4）と、第２電磁弁（ＳＶ2）とが設けられている。上記逆止弁（CV２）及び四路切換弁（22）の間と、逆止弁（ＣＶ1）及び加熱熱交換器（71）の間とは吐出ガスバイパス管（ＧＰＬ）によって接続され、この吐出ガスバイパス管（ＧＰＬ）には加熱熱交換器（71）へ向かう冷媒の流通のみを許容する逆止弁（ＣＶ5）が設けられている。

次に、２次側冷媒回路（Ｂ）について説明する。

２次側冷媒回路（Ｂ）は、上記加熱熱交換器（71）との間で熱交換する循環用蒸発器（51）と、冷却熱交換器（72）との間で熱交換する循環用凝縮器（61）と、１次側熱源熱交換器（12）との間で熱交換する２次側熱源熱交換器（１）と、該２次側熱源熱交換器（１）に対し、ガス配管（６）及び液配管（７）によって並列に接続された複数の室内熱交換器（3，3，3）及び室内電動膨張弁（EV1，EV1，EV1）と、２個のメインタンク（T1，T2）と、２個のサブタンク（ST1，ST2）とを備えている。

詳しくは、循環用蒸発器（51）の上端部に接続するガス供給管（52）は、４本の分岐管（５２a～５２d）に分岐されてそれぞれが各メインタンク（T1，T2）及び各サブタンク（ST1，ST2）の上端部に個別に接続されている。これら各分岐管（５２a～５２d）には、第１～第４のタンク加圧電磁弁（SV-P1～SV-P4）が設けられている。

上記循環用蒸発器（51）の下端部に接続する液回収管（53）は、２本の分岐管（53a，53b）に分岐されてそれぞれが各サブタンク（ST1，ST2）の下端部に個別に接続されている。これら分岐管（53a，53b）には、サブタンク（ST1，ST2）からの冷媒の流出のみを許容する逆止弁（CV6，CV6）が設けられている。

一方、循環用凝縮器（61）の上端部に接続するガス回収管（62）は、４本の分岐管（６２a～６２d）に分岐されてそれぞれが各メインタンク（T1，T2）及び各サブタンク（ST1，ST2）の上端部に個別に接続されている。これら各分岐管（６２a～６２d）には、第１～第４のタンク減圧電磁弁（SV-V1～SV-V4）が設けられている。

上記循環用凝縮器（61）の下端部に接続する液供給管（63）は、２本の分岐管（63a，63b）に分岐されてそれぞれが各メインタンク（T1，T2）の下端部に個別に接続されている。これら分岐管（63a，63b）には、メインタンク（T1，T2）への冷媒の回収のみを許容する逆止弁（CV7，CV7）が設けられている。

上記室内熱交換器（３）から延びる液配管（７）は第１及び第２の液配管（7A, 7B）に分岐され、第１分岐液配管（7A）は、第３電磁弁（ＳＶ3）を介して上記各分岐管（63a, 63b）に接続されている。第２分岐液配管（7B）は、第４電磁弁（ＳＶ4）及び第５電磁弁（ＳＶ5）を介して２次側熱源熱交換器（１）の液側に接続されている。この第２分岐液配管（7B）における第４電磁弁（ＳＶ4）と第５電磁弁（ＳＶ5）との間には、接続管（１７a～１７d）を介して各メインタンク（T1, T2）及び各サブタンク（ST1, ST2）の下端部が接続されている。

尚、この各接続管（１７a～１７d）には、各メインタンク（T1, T2）及び各サブタンク（ST1, ST2）から第２分岐液配管（7B）へ向かう冷媒の流通のみを許容する逆止弁（CV8, CV8, …）が設けられている。第２分岐液配管（7B）における第５電磁弁（SV５）及び２次側熱源熱交換器（１）の間と、分岐管（63a, 63b）とは、冷房液回収管（ＣＬＬ）によって接続されており、この冷房液回収管には第６電磁弁（ＳＶ6）が設けられている。

以下、室内の冷房及び暖房運転動作について説明する。

冷房運転時には、先ず、１次側冷媒回路（Ａ）では、四路切換弁（22）が実線側に切換えられ、第１電磁弁（ＳＶ1）を開口し、第２電磁弁（ＳＶ2）を閉鎖する。一方、２次側冷媒回路（Ｂ）では、第１及び第３のタンク加圧電磁弁（SV-P1, SV-P3）、第２及び第４のタンク減圧電磁弁（SV-V2, SV-V3）、第４電磁弁（ＳＶ4）及び第６電磁弁（ＳＶ6）を開口し、第２及び第４のタンク加圧電磁弁（SV-P2, SV-P4）、第１及び第３のタンク減圧電磁弁（SV-V1, SV-V4）、第３電磁弁（ＳＶ3）及び第５電磁弁（SV５）を閉鎖する。

この状態で、１次側冷媒回路（Ａ）にあっては、図３１に実線の矢印で示す如く、圧縮機（11）から吐出した高温高圧のガス冷媒が、室外熱交換器（14）及び加熱熱交換器（71）を順に流れて凝縮する。その後、この冷媒は、冷却熱交換器（72）及び１次側熱源熱交換器（12）に分流し、各室外電動膨張弁（EV-A, EV-B）において減圧された後、冷却熱交換器（72）においては循環用凝縮器（61）の冷媒と熱交換し、１次側熱源熱交換器（12）においては２次側熱源熱交換器（１）の冷媒と熱交換をして蒸発する。これら蒸発した冷媒は圧縮機（11）に戻る。この循環動作を繰り返す。

一方、２次側冷媒回路（Ｂ）にあっては、第１メインタンク（T1）及び第１サブタンク（ＳＴ1）の内圧が高圧となり、逆に、第２メインタンク（T2）及び第２サブタンク（ＳＴ2）の内圧が低圧となる。これにより、図３１に実線の矢印で示すように、第１メインタンク（T1）から押し出された液冷媒が、第２分岐液配管（7B）を経て、室内電動膨張弁（ＥＶ1）によって減圧された後、室内熱交換器（３）において室内空気と熱交換し、蒸発して室内空気を冷却する。その後、この冷媒は、２次側熱源熱交換器（１）において凝縮し、冷房液回収管（ＣＬＬ）を経て第２メインタンク（T2）に回収される。一方、第１サブタンク（ＳＴ1）は循環用蒸発器（51）と均圧されているので、図３１に破線の矢印で示すように、該第１サブタンク（ＳＴ1）の液冷媒が循環用蒸発器（51）に供給される。更に、この際、第２サブタンク（ＳＴ2）には、第２分岐液配管（7B）を流れている冷媒の一部が回収される。

この動作を所定時間行った後、２次側冷媒回路（Ｂ）の電磁弁を切換える。つまり、第１及び第３のタンク加圧電磁弁（SV-P1，SV-P3）、第２及び第４のタンク減圧電磁弁（SV-V2，SV-V3）を閉鎖し、第２及び第４のタンク加圧電磁弁（SV-P2，SV-P4）、第１及び第３のタンク減圧電磁弁（SV-V1，SV-V4）を開口する。これにより、第１メインタンク（T1）及び第１サブタンク（ＳＴ1）の内圧が低圧となり、逆に、第２メインタンク（T2）及び第２サブタンク（ＳＴ2）の内圧が高圧となる。このため、第２メインタンク（T2）から押し出された液冷媒が循環して第１メインタンク（T1）に回収される冷媒循環状態となる。また、第２サブタンク（ＳＴ2）の液冷媒が循環用蒸発器（51）に供給され、第１サブタンク（ＳＴ1）に、第２分岐液配管（7B）を流れている冷媒の一部が回収される。

この電磁弁の切換え動作を繰り返し、２次側冷媒回路（Ｂ）では冷媒が循環し、室内を冷房する。

次に、室内の暖房運転時について説明する。

暖房運転時には、先ず、１次側冷媒回路（Ａ）では、四路切換弁（22）が破線側に切換えられ、第２電磁弁（ＳＶ2）を開口し、第１電磁弁（ＳＶ1）を閉鎖する。一方、２次側冷媒回路（Ｂ）では、第３電磁弁（ＳＶ3）及び第５電磁弁（ＳＶ5）を開口し、第４電磁弁（ＳＶ4）及び第６電磁弁（ＳＶ6）を閉鎖した状態において、他の電磁弁の

開閉動作が繰り返される。

つまり、上述した冷房運転の場合と同様に、第１及び第３のタンク加圧電磁弁（SV-P1, SV-P3）及び第２及び第４のタンク減圧電磁弁（SV-V2, SV-V3）を開口し、第２及び第４のタンク加圧電磁弁（SV-P2, SV-P4）及び第１及び第３のタンク減圧電磁弁（SV-V1, SV-V4）を閉鎖した状態と、逆に、第１及び第３のタンク加圧電磁弁（SV-P1, SV-P3）及び第２及び第４のタンク減圧電磁弁（SV-V2, SV-V3）を閉鎖し、第２及び第４のタンク加圧電磁弁（SV-P2, SV-P4）及び第１及び第３のタンク減圧電磁弁（SV-V1, SV-V4）を開口した状態とが交互に切換えられる。

これにより、１次側冷媒回路（Ａ）にあっては、図３２に実線の矢印で示す如く、圧縮機（11）から吐出した高温高圧のガス冷媒が、吐出ガスバイパス管（ＧＰＬ）を経て加熱熱交換器（71）において循環用蒸発器（51）の冷媒と熱交換して顕熱変化する。その後、このガス冷媒は、暖房ガス供給管（ＷＧＬ）を経て１次側熱源熱交換器（12）に流れ、ここで２次側熱源熱交換器（１）の冷媒と熱交換して凝縮する。この凝縮した液冷媒は、第２室外電動膨張弁（EV-B）において減圧された後、分流し、冷却熱交換器（72）及び室外熱交換器（14）に流れる。冷却熱交換器（72）おいては循環用凝縮器（61）の冷媒と熱交換し、室外熱交換器（14）においては外気と熱交換して蒸発し、それぞれ圧縮機（11）に戻る。この循環動作を繰り返す。

一方、２次側冷媒回路（Ｂ）にあっては、例えば、第１メインタンク（T1）及び第１サブタンク（ＳＴ1）の内圧が高圧となり、逆に、第２メインタンク（T2）及び第２サブタンク（ＳＴ2）の内圧が低圧となる状態では、図３２に実線の矢印で示すように、第１メインタンク（T1）から押し出された液冷媒が、第２分岐液配管（7B）を２次側熱源熱交換器（１）に向かって流れ、該２次側熱源熱交換器（１）において蒸発した後、室内熱交換器（３）において凝縮し、第１分岐液配管（7A）を経て第２メインタンク（T2）に回収される。この際にも図３２に破線の矢印で示すように、第１サブタンク（ＳＴ1）の液冷媒が循環用蒸発器（51）に供給され、第２サブタンク（ＳＴ2）には、第２分岐液配管（7A）から冷媒が回収されている。そして、上述の如く電磁弁の切換え動作が繰り返され、２次側冷媒回路（Ｂ）では冷媒が循環し、室内を暖房する。

－第２０実施形態－

次に、本発明の第２０実施形態を図３３～図３５に基づき説明する。尚、本実施形態もヒートポンプの空気調和装置に適用したものであり、主たる構成は、上述した第１９実施形態のものと同様であるので、ここでは、この第１９実施形態との相違点についてのみ説明する。

１次側冷媒回路（Ａ）は、加熱熱交換器（71）と第１電磁弁（ＳＶ1）との間に電動膨張弁（EV-C）を備えると共に、この電動膨張弁（EV-C）をバイパスするバイパス管（ＢＰＬ）を備えている。このバイパス管（ＢＰＬ）にはキャピラリチューブ（CT）が設けられている。１次側熱源熱交換器（12）の液側には上記電動膨張弁（EV-B）に代えてキャピラリチューブ（CT）が設けられている。その他の構成は、上述した第１９実施形態のものと略同様である。

一方、２次側冷媒回路（Ｂ）は、第１サブタンク（ＳＴ1）の上端部が第１メインタンク（T1）の上端部に、第１サブタンク（ＳＴ1）の上端部が第１メインタンク（T1）の上端部にそれぞれ接続されている。各メインタンク（T1, T2）に対する循環用蒸発器（51）及び循環用凝縮器（61）の接続状態を切換える電磁弁（SV-P1, SV-P2, SV-V1, SV-V2）の切換え動作のみで各サブタンク（ST1, ST2）に対する循環用蒸発器（51）及び循環用凝縮器（61）の接続状態が切換わる。各メインタンク（T1, T2）に繋がる接続管（17a, 17b）は、一部が分岐管（17a-A, 17b-A）に分岐され、逆止弁（CV9, CV9）を介して第１液配管（7A）にも接続されている。その他の構成は、上述した第１９実施形態のものと略同様である。

本実施形態の冷媒回路における冷房運転時には図３４に矢印で示すように、また、暖房運転時には図３５に矢印で示すように、それぞれ冷媒が循環し、室内の冷房及び暖房を行う。これら冷媒循環動作は、上述した第１９実施形態の場合と略同様であるので、ここでは、詳細を省略する。尚、１次側冷媒回路（Ａ）において、冷房運転時は、電動膨張弁（EV-C）を閉鎖し、キャピラリチューブ（CT）によって冷媒の減圧が行われる一方、暖房運転時は、この電動膨張弁（EV-C）を開口し、冷媒が減圧されることなく１次側熱源熱交換器（12）に流れる。

また、２次側冷媒回路（Ｂ）において、冷房運転時は、２次側熱源熱交換器（１）

で凝縮した冷媒が、第１液配管（7A）及び分岐管（17b-A）を経てメインタンク（T2）に回収される。一方、暖房運転時は、室内熱交換器（３）で凝縮した冷媒が、第１液配管（7A）及び分岐管（17b-A）を経てメインタンク（T2）に回収される。

－第２１実施形態－

次に、本発明の第２１実施形態について説明する。尚、本実施形態もヒートポンプの空気調和装置に適用したものであり、主たる構成は、上述した第２０実施形態のものと同様であるので、ここでは、この第２０実施形態との相違点についてのみ説明する。

１次側冷媒回路（Ａ）は、上述した第２０実施形態のものに対し、暖房ガス供給管（ＷＧＬ）が設けられておらず、また、１次側熱源熱交換器（12）と四路切換弁（22）とを接続する配管には逆止弁が設けられていない。

２次側冷媒回路（Ｂ）は、各メインタンク（T1, T2）及び各サブタンク（ST1, ST2）の上端部に対する循環用蒸発器（51）及び循環用凝縮器（61）の接続状態は、上述した第１９実施例のものが採用されている。

また、本実施形態では、暖房運転時における循環用蒸発器（51）の蒸発温度が２次側熱源熱交換器（１）における蒸発温度よりも高く設定されている。つまり、第１１実施形態において、循環用凝縮器（61）における凝縮温度を２次側熱源熱交換器（１）における凝縮温度よりも低くしたのと同様に、熱交換器の能力に差をもたせている。

具体的に、加熱熱交換器（71）に繋がる第１ガス供給管（GL-1）と１次側熱源熱交換器（12）に繋がる第２ガス供給管（GL-2）とは、配管径が異なり、第１ガス供給管（GL-1）の流量が第２ガス供給管（GL-2）の流量よりも所定の比率だけ小さく設定されている。これに対し、加熱熱交換器（71）と循環用蒸発器（51）との熱交換面積は、１次側熱源熱交換器（12）と２次側熱源熱交換器（１）の熱交換面積よりも小さく設定され、その比率は上記所定の比率よりも小さく設定されている。

例えば、第１ガス供給管（GL-1）の流量と第２ガス供給管（GL-2）の流量との比が１：１０である場合に、加熱熱交換器（71）と循環用蒸発器（51）との熱交換面積と、１次側熱源熱交換器（12）と２次側熱源熱交換器（１）の熱交換面積の比は２：

１０に設定されている。これにより、冷媒流量に対する熱交換器としての能力は、循環用蒸発器（51）の方が２次側熱源熱交換器（１）よりも高く、このため、循環用蒸発器（51）の蒸発温度は２次側熱源熱交換器（１）の蒸発温度よりも高くなる。この結果、循環用蒸発器（51）の内部圧力が２次側熱源熱交換器（１）の内部圧力よりも高くなり、暖房運転時には、メインタンク（T1, T2）から２次側熱源熱交換器（１）に向かって液冷媒を押し出す。その他の構成は、上述した第２０実施形態のものと略同様である。

そして、本実施形態の冷媒回路における冷房運転時には、図３７に矢印で示すように、各冷媒回路（A, B）において冷媒が循環し、室内の冷房を行う。この冷媒循環動作は、上述した第１９実施形態或いは第２０実施形態の場合と略同様であるので、ここでは、詳細を省略する。

一方、暖房運転時の１次側冷媒回路は、図３８に矢印で示すように、圧縮機（11）からの吐出冷媒が、加熱熱交換器（71）と１次側熱源熱交換器（12）とに分流し、それぞれにおいて凝縮する。そして、加熱熱交換器（71）で凝縮した冷媒は、分岐管（LSL-1）を経て冷却熱交換器（72）に流れ、循環用凝縮器（61）との間で熱交換して蒸発した後、圧縮機（11）に戻る。一方、１次側熱源熱交換器（12）で凝縮した冷媒は、暖房液配管（ＷＬＬ）を経て室外熱交換器（14）に流れ、外気と熱交換して蒸発した後、圧縮機（11）に戻る。

２次側冷媒回路（Ｂ）では、図３８に矢印で示すように冷媒が循環して室内を暖房する。この２次側冷媒回路（Ｂ）における冷媒循環動作は、上述した第１９実施形態或いは第２０実施形態の場合と略同様であるので、ここでは、詳細を省略する。

－その他の実施形態－

尚、上述した各実施形態では、本発明に係る熱搬送装置を空気調和装置の冷媒回路に適用した場合について説明したが、本発明はこれに限らず、その他種々の冷凍機に適用してもよい。

また、第１～第１０実施形態では、タンク（Ｔ）を接続管（17）によって液配管（７）に接続するようにしたが、該タンク（Ｔ）を液配管（７）に直接的に接続する

ようにしてもよい。

また、上述した各実施形態のうちタンク（T1, T2）やサブタンク（ST1, ST2）を複数設けたものにあっては、これらを３個以上配置してもよい。つまり、２つ以上の第１タンク（T1）及び第１サブタンク（ＳＴ1）と、２つ以上の第２タンク（T2）及び第２サブタンク（ＳＴ2）を設け、第１タンク（T1）及び第１サブタンク（ＳＴ1）が同一の機能を果たし、第２タンク（T2）及び第２サブタンク（ＳＴ2）が同一の機能を果たすようにしてもよい。

[産業上の利用可能性]

以上のように、本発明によれば、空気調和装置の冷媒回路などとして利用可能な熱搬送装置に有用でありに、特に、ポンプ等の駆動源を必要とすることなしに熱搬送媒体を循環させて熱搬送を行う熱搬送装置に適している。

請求の範囲

1．熱源側熱交換手段（1）と利用側熱交換手段（3）とがガス配管（6）及び液配管（7）により冷媒循環が可能に接続されて成り、上記熱源側熱交換手段（1）が熱源手段（A）との間で熱交換を行う冷媒回路（B）と、

上記液配管（7）に連通し、液冷媒を貯留するタンク手段（T）と、

該タンク手段（T）の内部圧力を上昇させる加圧動作と、内部圧力を下降させる減圧動作とを交互に行う圧力調節手段（18）と、

該圧力調節手段（18）の加圧動作時には上記タンク手段（T）から蒸発器となる熱交換手段への液冷媒の供給のみを許容する一方、減圧動作時には凝縮器となる熱交換手段からタンク手段（T）への液冷媒の回収のみを許容し、冷媒回路（B）の冷媒を循環させて利用側熱交換手段（3）に吸熱又は放熱を行わせる冷媒制御手段（H）とを備えていることを特徴とする熱搬送装置。

2．請求項1記載の熱搬送装置において、

利用側熱交換手段（3）は吸熱を行う蒸発器であり、

冷媒制御手段（H）は、

圧力調節手段（18）の加圧動作時、タンク手段（T）から利用側熱交換手段（3）への液冷媒の供給を許容すると共に、タンク手段（T）から熱源側熱交換手段（1）への液冷媒の供給を阻止する一方、

圧力調節手段（18）の減圧動作時、熱源側熱交換手段（1）からタンク手段（T）への液冷媒の回収を許容すると共に、利用側熱交換手段（3）からタンク手段（T）への液冷媒の回収を阻止することを特徴とする熱搬送装置。

3．請求項1記載の熱搬送装置において、

利用側熱交換手段（3）は放熱を行う凝縮器であり、

冷媒制御手段（H）は、

圧力調節手段（18）の加圧動作時、タンク手段（T）から熱源側熱交換手段（1）への液冷媒の供給を許容すると共に、タンク手段（T）から利用側熱交換手段（3）

への液冷媒の供給を阻止する一方、

圧力調節手段（18）の減圧動作時、利用側熱交換手段（3）からタンク手段（T）への液冷媒の回収を許容すると共に、熱源側熱交換手段（1）からタンク手段（T）への液冷媒の回収を阻止することを特徴とする熱搬送装置。

4．請求項1、2又は3記載の熱搬送装置において、

圧力調節手段（18）は、加圧動作時、タンク手段（T）の冷媒に熱を与えて該タンク手段（T）の内圧を上昇させる一方、減圧動作時、タンク手段（T）の冷媒から熱を奪って該タンク手段（T）の内圧を下降させることを特徴とする請求項1、2又は3記載の熱搬送装置。

5．請求項4記載の熱搬送装置において、

圧力調節手段（18）は、タンク手段（T）に隣接され、タンク手段（T）の冷媒に熱を与える加熱動作と、タンク手段（T）の冷媒から熱を奪う冷却動作とを切換えて行う熱交換手段（18b）であることを特徴とする請求項4記載の熱搬送装置。

6．請求項4記載の熱搬送装置において、

圧力調節手段（18）は、圧縮機（D1）と第1熱交換器（D3）と減圧機構（D4）と第2熱交換器（D5）と圧縮機（D1）の吐出側に対する第1熱交換器（D3）及び第2熱交換器（D5）の接続状態を交互に切換える切換え手段（D2）とを有する冷媒回路（D）を備え、

上記第1熱交換器（D3）が、タンク手段（T）との間で熱交換し、切換え手段（D2）の切換え動作に伴ってタンク手段（T）の冷媒を加熱及び冷却することを特徴とする熱搬送装置。

7．請求項1、2又は3記載の熱搬送装置において、

圧力調節手段（18）は、タンク手段（T）に圧力管（19）を介して連結された圧力発生手段（18a, 18c, 19c）を備え、加圧動作時、圧力発生手段（18a, 18c, 19c）からタンク手段（T）の内部に高圧を作用させる一方、減圧動作時、圧力発生手段（18

a, 18c, 19c）からタンク手段（T）の内部に低圧を作用させることを特徴とする熱搬送装置。

8．請求項7記載の熱搬送装置において、

圧力発生手段は、液冷媒が貯留可能な貯留容器（18a）であり、

圧力調節手段（18）は、加圧動作時、貯留容器（18a）の液冷媒に熱を与えて該液冷媒を蒸発させ、該貯留容器（18a）の内圧を上昇させる一方、減圧動作時、貯留容器（18a）のガス冷媒から熱を奪って該ガス冷媒を凝縮させ、該貯留容器（18a）の内圧を下降させることを特徴とする熱搬送装置。

9．請求項7記載の熱搬送装置において、

圧力発生手段は、圧縮機（18c）であり、

圧力管（19）の圧縮機（18c）に対する接続状態が、切換え手段（I）によって圧縮機（18c）の吐出側と吸入側とに切換わり、

圧力調節手段（18）は、切換え手段（I）の切換え動作により、加圧動作時に圧力管（19）を圧縮機（18c）の吐出側に接続する一方、減圧動作時に圧力管（19）を圧縮機（18c）の吸入側に接続することを特徴とする請求項7記載の熱搬送装置。

10．請求項7記載の熱搬送装置において、

圧力発生手段は、冷媒が貯留可能な熱交換器（19c）であり、

圧力調節手段（18）は、加圧動作時に熱交換器（19c）の冷媒に熱を与えて該熱交換器（19c）の内圧を上昇させる一方、減圧動作時に熱交換器（19c）の冷媒から熱を奪って該熱交換器（19c）の内圧を下降させることを特徴とする熱搬送装置。

11．請求項10記載の熱搬送装置において、

圧力調節手段（18）は、圧縮機（D1）と第1熱交換器（D3）と減圧機構（D4）と第2熱交換器（D5）と圧縮機（D1）の吐出側に対する第1熱交換器（D3）及び第2熱交換器（D5）の接続状態を交互に切換える切換え手段（D2）とを有する冷媒回路（D）を備え、

上記第１熱交換器（D3）が、熱交換器（１９c）との間で熱交換し、切換え手段（D2）の切換え動作に伴って熱交換器（１９c）の冷媒を加熱及び冷却することを特徴とする熱搬送装置。

１２．請求項２記載の熱搬送装置において、

冷媒制御手段（H）は、

液配管（７）におけるタンク手段（T）の接続位置と熱源側熱交換手段（１）との間に設けられ、圧力調節手段（18）の加圧動作時に閉鎖し、減圧動作時に開口する第１の電磁弁（SV-A）と、

液配管（７）におけるタンク手段（T）の接続位置と利用側熱交換手段（３）との間に設けられ、圧力調節手段（18）の加圧動作時に開口し、減圧動作時に閉鎖する第２の電磁弁（SV-B）とによって構成されていることを特徴とする熱搬送装置。

１３．請求項３記載の熱搬送装置において、

冷媒制御手段（H）は、

液配管（７）におけるタンク手段（T）の接続位置と熱源側熱交換手段（１）との間に設けられ、圧力調節手段（18）の加圧動作時に開口し、減圧動作時に閉鎖する第１の電磁弁（SV-A）と、

液配管（７）におけるタンク手段（T）の接続位置と利用側熱交換手段（３）との間に設けられ、圧力調節手段（18）の加圧動作時に閉鎖し、減圧動作時に開口する第２の電磁弁（SV-B）とによって構成されていることを特徴とする熱搬送装置。

１４．請求項２記載の熱搬送装置において、

冷媒制御手段（H）は、

液配管（７）におけるタンク手段（T）の接続位置と熱源側熱交換手段（１）との間に設けられ、熱源側熱交換手段（１）からタンク手段（T）への液冷媒の流通のみを許容する第１の逆止弁（CV1）と、

液配管（７）におけるタンク手段（T）の接続位置と利用側熱交換手段（３）との間に設けられ、タンク手段（T）から利用側熱交換手段（３）への液冷媒の流通の

みを許容する第２の逆止弁（ＣＶ２）とによって構成されていることを特徴とする熱搬送装置。

１５．請求項３記載の熱搬送装置において、

冷媒制御手段（Ｈ）は、

液配管（７）におけるタンク手段（Ｔ）の接続位置と熱源側熱交換手段（１）との間に設けられ、タンク手段（Ｔ）から熱源側熱交換手段（１）への液冷媒の流通のみを許容する第１の逆止弁（ＣＶ３）と、

液配管（７）におけるタンク手段（Ｔ）の接続位置と利用側熱交換手段（３）との間に設けられ、利用側熱交換手段（３）からタンク手段（Ｔ）への液冷媒の流通のみを許容する第２の逆止弁（ＣＶ４）とによって構成されていることを特徴とする熱搬送装置。

１６．熱源側熱交換手段（１）と利用側熱交換手段（３）とがガス配管（６）及び液配管（７）により冷媒循環が可能に接続されて成り、上記熱源側熱交換手段（１）が熱源手段（Ａ）との間で熱交換を行う冷媒回路（Ｂ）と、

上記液配管（７）に互いに並列に接続され、液冷媒を貯留する１以上の第１タンク手段（T1）及び１以上の第２タンク手段（T2）と、

該第１タンク手段（T1）の内部圧力を上昇させると共に第２タンク手段（T2）の内部圧力を下降させる第１圧力状態と、第１タンク手段（T1）の内部圧力を下降させると共に第２タンク手段（T2）の内部圧力を上昇させる第２圧力状態とに交互に切換える圧力調節手段（18）と、

該圧力調節手段（18）の第１圧力状態時には、第１タンク手段（T1）から蒸発器となる熱交換手段に液冷媒を供給すると共に、凝縮器となる熱交換手段から第２タンク手段（T2）に液冷媒を回収する一方、第２圧力状態時には、第２タンク手段（T2）から蒸発器となる熱交換手段に液冷媒を供給すると共に、凝縮器となる熱交換手段から第１タンク手段（T1）に液冷媒を回収し、冷媒回路（Ｂ）の冷媒を循環させて利用側熱交換手段（３）に吸熱又は放熱を連続して行わせる冷媒制御手段（Ｈ）とを備えていることを特徴とする熱搬送装置。

１７．請求項１６記載の熱搬送装置において、

利用側熱交換手段（３）は吸熱を行う蒸発器であり、

冷媒制御手段（Ｈ）は、

圧力調節手段（18）の第１圧力状態時、第１タンク手段（T1）から利用側熱交換手段（３）へ液冷媒を供給すると共に、熱源側熱交換手段（１）から第２タンク手段（T2）へ液冷媒を回収する一方、

圧力調節手段（18）の第２圧力状態時、第２タンク手段（T2）から利用側熱交換手段（３）へ液冷媒を供給すると共に、熱源側熱交換手段（１）から第１タンク手段（T1）へ液冷媒を回収するように液配管（７）における冷媒流通状態を切換えることを特徴とする熱搬送装置。

１８．請求項１６記載の熱搬送装置において、

利用側熱交換手段（３）は放熱を行う凝縮器であり、

冷媒制御手段（Ｈ）は、

圧力調節手段（18）の第１圧力状態時、第１タンク手段（T1）から熱源側熱交換手段（１）へ液冷媒を供給すると共に、利用側熱交換手段（３）から第２タンク手段（T2）へ液冷媒を回収する一方、

圧力調節手段（18）の第２圧力状態時、第２タンク手段（T2）から熱源側熱交換手段（１）へ液冷媒を供給すると共に、利用側熱交換手段（３）から第１タンク手段（T1）へ液冷媒を回収するように液配管（７）における冷媒流通状態を切換えることを特徴とする熱搬送装置。

１９．請求項１６、１７又は１８記載の熱搬送装置において、

圧力調節手段（18）は、各タンク手段（T1, T2）に圧力管（19d, 19e）を介して連結された圧力発生手段（18A, 18B, D1, E1, E2）を備え、該圧力発生手段（18A, 18B, D1, E1, E2）により、第１圧力状態時、第１タンク手段（T1）の内部に高圧を作用させ、第２タンク手段（T2）の内部に低圧を作用させる一方、第２圧力状態時、第２タンク手段（T2）の内部に高圧を作用させ、第１タンク手段（T1）の内部に低圧を作用させ

ることを特徴とする熱搬送装置。

２０．請求項１９記載の熱搬送装置において、

圧力発生手段は、第１タンク手段（T1）に接続されて液冷媒が貯留可能な第１の貯留容器（１８A）及び第２タンク手段（T2）に接続されて液冷媒が貯留可能な第２の貯留容器（１８B）であり、

圧力調節手段（18）は、第１圧力状態時、第１の貯留容器（１８A）の液冷媒に熱を与えて該液冷媒を蒸発させ、該貯留容器（１８A）の内圧を上昇させると共に、第２の貯留容器（１８B）のガス冷媒から熱を奪って該ガス冷媒を凝縮させ、該貯留容器（１８B）の内圧を下降させる一方、第２圧力状態時、第２の貯留容器（１８B）の液冷媒に熱を与えて該液冷媒を蒸発させ、該貯留容器（１８B）の内圧を上昇させると共に、第１の貯留容器（１８A）のガス冷媒から熱を奪って凝該ガス冷媒を縮させ、該貯留容器（１８A）の内圧を下降させることを特徴とする熱搬送装置。

２１．請求項１９記載の熱搬送装置において、

圧力発生手段は、圧縮機（D1）であり、

第１タンク手段（T1）及び第２タンク手段（T2）の圧縮機（D1）に対する接続状態の切換えが、切換え手段（Ｉ）によって圧力管（19d, 19e）を圧縮機（D1）の吐出側と吸入側とに切換えて行い、

圧力調節手段（18）は、第１圧力状態時、圧縮機（D1）の吐出側を第１タンク手段（T1）に接続し、圧縮機（D1）の吸入側を第２タンク手段（T2）に接続する一方、第２圧力状態時、圧縮機（D1）の吐出側を第２タンク手段（T2）に接続し、圧縮機（D1）の吸入側を第１タンク手段（T1）に接続することを特徴とする熱搬送装置。

２２．請求項１９記載の熱搬送装置において、

圧力発生手段は、第１タンク手段（T1）に接続されて冷媒が貯留可能な第１の熱交換器（E1）及び第２タンク手段（T2）に接続されて冷媒が貯留可能な第２の熱交換器（E2）であり、

圧力調節手段（18）は、第１圧力状態時、第１の熱交換器（E1）の冷媒に熱を与

えて該熱交換器（E1）の内圧を上昇させると共に、第２の熱交換器（E2）の冷媒から熱を奪って該熱交換器（E2）の内圧を下降させる一方、第２圧力状態時、第２の熱交換器（E2）の冷媒に熱を与えて該熱交換器（E2）の内圧を上昇させると共に、第１の熱交換器（E1）の冷媒から熱を奪って該熱交換器（E1）の内圧を下降させることを特徴とする熱搬送装置。

２３．請求項２２記載の熱搬送装置において、

圧力調節手段（18）は、圧縮機（D1）と第１熱交換器（D3）と減圧機構（D4）と第２熱交換器（D5）と圧縮機（D1）の吐出側に対する第１熱交換器（D3）及び第２熱交換器（D5）の接続状態を交互に切換える切換え手段（Ｉ）とを有する冷媒回路（Ｄ）を備え、

第１熱交換器（D3）が第１タンク手段（T1）に接続された第１の熱交換器（E1）との間で熱交換し、第２熱交換器（D5）が第２タンク手段（T2）に接続された第２の熱交換器（E2）との間で熱交換し、切換え手段（Ｉ）の切換え動作に伴って第１圧力状態と第２圧力状態とに切換わることを特徴とする熱搬送装置。

２４．請求項１９記載の熱搬送装置において、

圧力発生手段は、加熱用熱交換器（D3）によって加熱される加圧用熱交換器（E2）と、冷却用熱交換器（D5）によって冷却される減圧用熱交換器（E1）とによって構成され、

圧力調節手段（18）は、第１圧力状態時、加圧用熱交換器（E2）を第１タンク手段（T1）に接続すると共に、減圧用熱交換器（E1）を第２タンク手段（T2）に接続する一方、第２圧力状態時、加圧用熱交換器（E2）を第２タンク手段（T2）に接続すると共に、減圧用熱交換器（E1）を第１タンク手段（T1）に接続することを特徴とする熱搬送装置。

２５．請求項２４記載の熱搬送装置において、

圧力調節手段（18）は、圧縮機（D1）と加熱用熱交換器（D3）と減圧機構（D4）と冷却用熱交換器（D5）とが冷媒配管によって順に接続されて成る冷媒回路（Ｄ）を

備えていることを特徴とする熱搬送装置。

２６．請求項１記載の熱搬送装置において、

圧力調節手段（18）は、タンク手段（Ｔ）の内部圧力を上昇させて該タンク手段（Ｔ）の液冷媒を液配管（７）に押出す加圧動作を行う加圧手段（50）と、タンク手段（Ｔ）の内部圧力を下降させて液配管（７）から該タンク手段（Ｔ）へ液冷媒を回収する減圧動作を行う減圧手段（60）とを備え、

上記減圧手段（60）は、タンク手段（Ｔ）に接続され且つ冷媒を凝縮することによってタンク手段（Ｔ）の内部圧力を下降させる循環用凝縮器（61）を備え、

該循環用凝縮器（61）の凝縮圧力が、凝縮器となる熱交換手段の凝縮圧力より低く設定されていることを特徴とする熱搬送装置。

２７．請求項１記載の熱搬送装置において、

圧力調節手段（18）は、タンク手段（Ｔ）の内部圧力を上昇させて該タンク手段（Ｔ）の液冷媒を液配管（７）に押出す加圧動作を行う加圧手段（50）と、タンク手段（Ｔ）の内部圧力を下降させて液配管（７）から該タンク手段（Ｔ）へ液冷媒を回収する減圧動作を行う減圧手段（60）とを備え、

上記加圧手段（50）は、タンク手段（Ｔ）に接続され且つ冷媒を蒸発することによってタンク手段（Ｔ）の内部圧力を上昇させる循環用蒸発器（51）を備え、

該循環用蒸発器（51）の蒸発圧力が、蒸発器となる熱交換手段の蒸発圧力より高く設定されていることを特徴とする熱搬送装置。

２８．請求項２７記載の熱搬送装置において、

循環用蒸発器（51）の上方には補助タンク手段（ST）が設けられ、

減圧手段（60）による減圧動作時、補助タンク手段（ST）を減圧手段（60）及び液配管（７）にそれぞれ連通させて液配管（７）の液冷媒を補助タンク手段（ST）に回収する一方、加圧手段（50）による加圧動作時、補助タンク手段（ST）を加圧手段（50）に連通させて補助タンク手段（ST）の液冷媒を循環用蒸発器（51）に落下供給させる切換え手段（Ｉ）が設けられていることを特徴とする熱搬送装置。

２９．請求項２７記載の熱搬送装置において、

循環用蒸発器（51）の上方には、１以上の第１補助タンク手段（ＳＴ1）及び１以上の第２補助タンク手段（ＳＴ2）が設けられ、

第１補助タンク手段（ＳＴ1）を減圧手段（60）及び液配管（７）にそれぞれ連通させて液配管（７）の液冷媒を該第１補助タンク手段（ＳＴ1）に回収させると共に、第２補助タンク手段（ＳＴ2）を加圧手段（50）に連通させて該第２補助タンク手段（ＳＴ2）の液冷媒を循環用蒸発器（51）に落下供給させる第１切換え状態と、第２補助タンク手段（ＳＴ2）を減圧手段（60）及び液配管（７）にそれぞれ連通させて液配管（７）の液冷媒を該第２補助タンク手段（ＳＴ2）に回収させると共に、第１補助タンク手段（ＳＴ1）を加圧手段（50）に連通させて該第１補助タンク手段（ＳＴ1）の液冷媒を循環用蒸発器（51）に落下供給させる第２切換え状態とに切換える切換え手段（Ｉ）が設けられていることを特徴とする熱搬送装置。

３０．請求項１６記載の熱搬送装置において、

圧力調節手段（18）は、第１タンク手段（T1）及び第２タンク手段（T2）に対し、一方のタンク手段の内部圧力を上昇させて該タンク手段の液冷媒を液配管（７）に押出す加圧動作を行う加圧手段（50）と、他方のタンク手段の内部圧力を下降させて液配管（７）から該タンク手段へ液冷媒を回収する減圧動作を行う減圧手段（60）とを備え、

上記減圧手段（60）は、各タンク手段（T1，T2）に接続され且つ冷媒を凝縮することによって各タンク手段（T1，T2）の内部圧力を下降させる循環用凝縮器（61）を備え、

該循環用凝縮器（61）の凝縮圧力が、凝縮器となる熱交換手段の凝縮圧力より低く設定され、

上記圧力調節手段（18）は、第１圧力状態時に、第１タンク手段（T1）を加圧手段（50）により加圧すると共に第２タンク手段（T2）を減圧手段（60）により減圧する一方、第２圧力状態時に、第２タンク手段（T2）を加圧手段（50）により加圧すると共に第１タンク手段（T1）を減圧手段（60）により減圧することを特徴とする熱搬

送装置。

３１．請求項１６記載の熱搬送装置において、

圧力調節手段（18）は、第１タンク手段（T1）及び第２タンク手段（T2）に対し、一方のタンク手段の内部圧力を上昇させて該タンク手段の液冷媒を液配管（７）に押出す加圧動作を行う加圧手段（50）と、他方のタンク手段の内部圧力を下降させて液配管（７）から該タンク手段へ液冷媒を回収する減圧動作を行う減圧手段（60）とを備え、

上記加圧手段（50）は、各タンク手段（T1，T2）に接続され且つ冷媒を蒸発することによって各タンク手段（T1，T2）の内部圧力を上昇させる循環用蒸発器（51）を備え、

該循環用蒸発器（51）の蒸発圧力が、蒸発器となる熱交換手段の蒸発圧力より高く設定され、

上記圧力調節手段（18）は、第１圧力状態時に、第１タンク手段（T1）を加圧手段（50）により加圧すると共に第２タンク手段（T2）を減圧手段（60）により減圧する一方、第２圧力状態時に、第２タンク手段（T2）を加圧手段（50）により加圧すると共に第１タンク手段（T1）を減圧手段（60）により減圧することを特徴とする熱搬送装置。

３２．請求項３１記載の熱搬送装置において、

循環用蒸発器（51）の上方には、１以上の第１補助タンク手段（ＳＴ1）及び１以上の第２補助タンク手段（ＳＴ2）が設けられ、

該第１補助タンク手段（ＳＴ1）を減圧手段（60）及び液配管（７）にそれぞれ連通させて液配管（７）の液冷媒を該第１補助タンク手段（ＳＴ1）に回収させると共に、第２補助タンク手段（ＳＴ2）を加圧手段（50）に連通させて該第２補助タンク手段（ＳＴ2）の液冷媒を循環用蒸発器（51）に落下供給させる第１切換え状態と、第２補助タンク手段（ＳＴ2）を減圧手段（60）及び液配管（７）にそれぞれ連通させて液配管（７）の液冷媒を該第２補助タンク手段（ＳＴ2）に回収させると共に、第１補助タンク手段（ＳＴ1）を加圧手段（50）に連通させて該第１補助タンク手段（ＳＴ1）の液

冷媒を循環用蒸発器（51）に落下供給させる第２切換え状態とに切換える手段（Ｉ）が設けられていることを特徴とする熱搬送装置。

３３．請求項２６又は３０記載の熱搬送装置において、

熱源手段（Ａ）は、熱源側熱交換手段（１）との間で熱交換を行う第１の熱交換手段（12）と、循環用凝縮器（61）との間で熱交換を行う第２の熱交換手段（72）とを備え、

利用側熱交換手段（３）の吸熱運転時における第１の熱交換手段（12）と第２の熱交換手段（72）との蒸発温度は同一である一方、上記第２の熱交換手段（72）を流れる冷媒の流量に対する循環用凝縮器（61）の容量の比は、第１の熱交換手段（12）を流れる冷媒の流量に対する熱源側熱交換手段（１）の容量の比よりも大きく設定されていることを特徴とする熱搬送装置。

３４．請求項２７又は３１記載の熱搬送装置において、

熱源手段（Ａ）は、熱源側熱交換手段（１）との間で熱交換を行う第１の熱交換手段（12）と、循環用蒸発器（51）との間で熱交換を行う第２の熱交換手段（71）とを備え、

利用側熱交換手段（３）の放熱運転時における第１の熱交換手段（12）と第２の熱交換手段（71）の凝縮温度は同一である一方、上記第２の熱交換手段（71）を流れる冷媒の流量に対する循環用蒸発器（51）の容量の比は、第１の熱交換手段（12）を流れる冷媒の流量に対する熱源側熱交換手段（１）の容量の比よりも大きく設定されていることを特徴とする熱搬送装置。

３５．請求項２６又は３０記載の熱搬送装置において、

減圧手段（60）は、タンク手段（Ｔ）の上端部と循環用凝縮器（61）のガス側とを接続するガス回収管（62）と、タンク手段（Ｔ）の下端部と循環用凝縮器（１）の液側とを接続する液供給管（63）とを備え、

上記液供給管（63）は、液配管（７）とは独立してタンク手段（Ｔ）の下端部に接続されていることを特徴とする熱搬送装置。

３６．請求項１又は１６記載の熱搬送装置において、

圧力調節手段（18）は、タンク手段（Ｔ）の内部圧力を上昇させて該タンク手段（Ｔ）の液冷媒を液配管（７）に押出す加圧動作を行う加圧手段（50）と、タンク手段（Ｔ）の内部圧力を下降させて液配管（７）から該タンク手段（Ｔ）へ液冷媒を回収する減圧動作を行う減圧手段（60）とを備え、

上記減圧手段（60）は、タンク手段（Ｔ）に接続され且つ冷媒を凝縮することによってタンク手段（Ｔ）の内部圧力を下降させる循環用凝縮器（61）を備え、

上記加圧手段（50）は、タンク手段（Ｔ）に接続され且つ冷媒を蒸発することによってタンク手段（Ｔ）の内部圧力を上昇させる循環用蒸発器（51）を備える一方、

熱源手段（Ａ）は、圧縮機（11）、熱源側熱交換手段（１）の間で熱交換を行う第１の熱交換手段（12）と、循環用凝縮器（61）の間で熱交換を行う第２の熱交換手段（72）と、循環用蒸発器（51）の間で熱交換を行う第３の熱交換手段（71）とを備え、利用側熱交換手段（３）の放熱運転時、圧縮機（11）から吐出したガス冷媒を、第３の熱交換手段（71）において循環用蒸発器（51）との間で熱交換を行って顕熱変化させた後、第１の熱交換手段（12）において熱源側熱交換手段（１）との間で熱交換を行って凝縮させ、更に、第２の熱交換手段（72）において循環用凝縮器（61）との間で熱交換を行って蒸発させることを特徴とする熱搬送装置。

３７．請求項１又は１６記載の熱搬送装置において、

圧力調節手段（18）は、タンク手段（Ｔ）の内部圧力を上昇させて該タンク手段（Ｔ）の液冷媒を液配管（７）に押出す加圧動作を行う加圧手段（50）と、タンク手段（Ｔ）の内部圧力を下降させて液配管（７）から該タンク手段（Ｔ）へ液冷媒を回収する減圧動作を行う減圧手段（60）とを備え、

上記減圧手段（60）は、タンク手段（Ｔ）に接続され且つ冷媒を凝縮することによってタンク手段（Ｔ）の内部圧力を下降させる循環用凝縮器（61）を備え、

上記加圧手段（50）は、タンク手段（Ｔ）に接続され且つ冷媒を蒸発することによってタンク手段（Ｔ）の内部圧力を上昇させる循環用蒸発器（51）を備える一方、

熱源手段（Ａ）は、圧縮機（11）、熱源側熱交換手段（１）の間で熱交換を行う

第１の熱交換手段（12）と、循環用凝縮器（61）の間で熱交換を行う第２の熱交換手段（72）と、循環用蒸発器（51）の間で熱交換を行う第３の熱交換手段（71）とを備え、利用側熱交換手段（３）の放熱運転時、圧縮機（11）から吐出したガス冷媒を、第３の熱交換手段（71）及び第１の熱交換手段（12）に分流し、第３の熱交換手段（71）において循環用蒸発器（51）との間で熱交換を行って凝縮させると共に、第１の熱交換手段（12）において熱源側熱交換手段（１）との間で熱交換を行って凝縮させ、その後、凝縮冷媒を、第２の熱交換手段（72）において循環用凝縮器（61）との間で熱交換を行って蒸発させることを特徴とする熱搬送装置。

Fig. 1

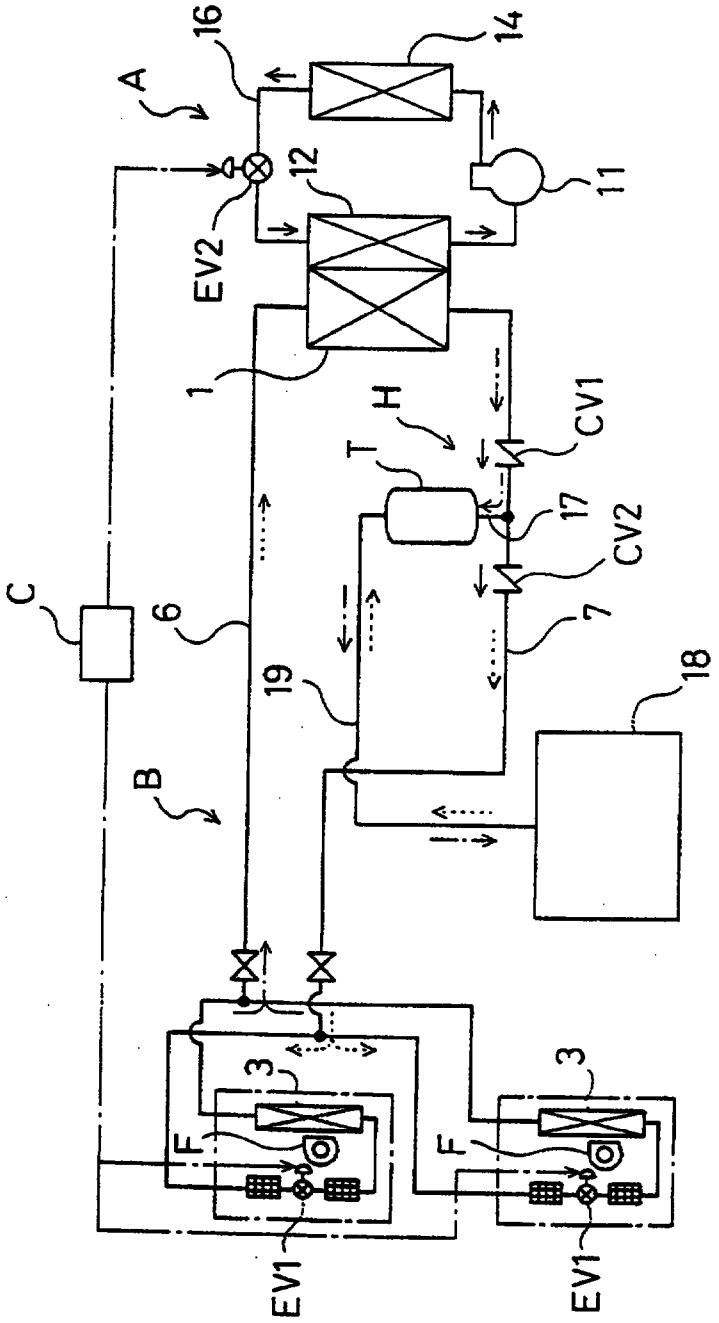


Fig. 2

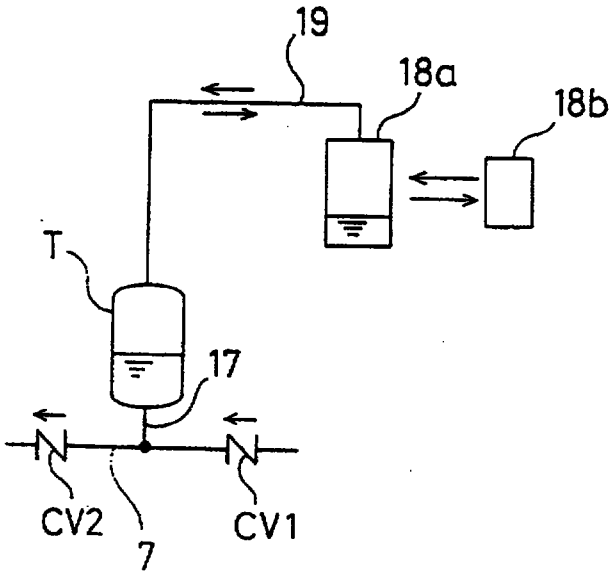


Fig. 3

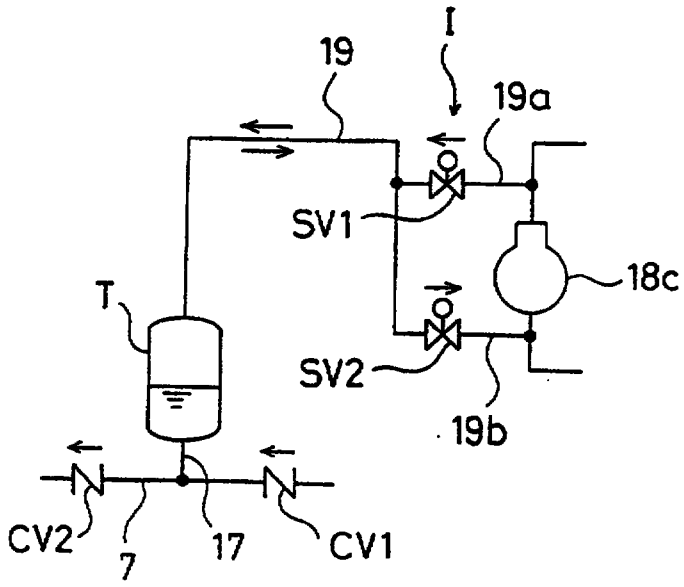


Fig. 4

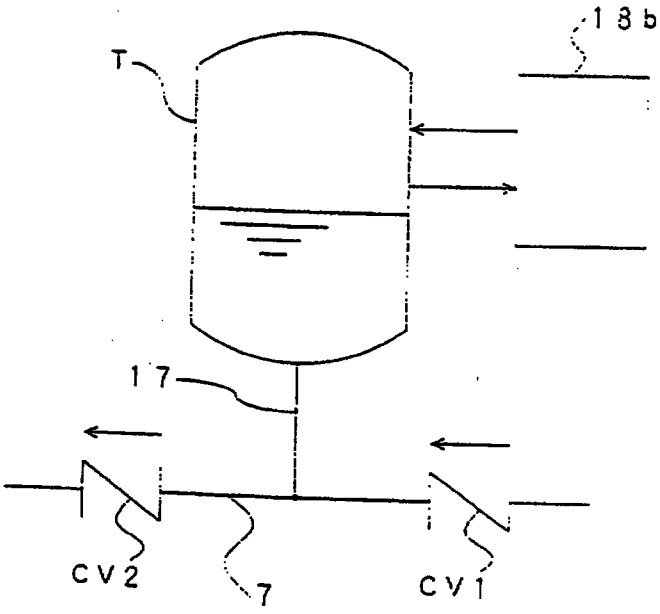


Fig. 5

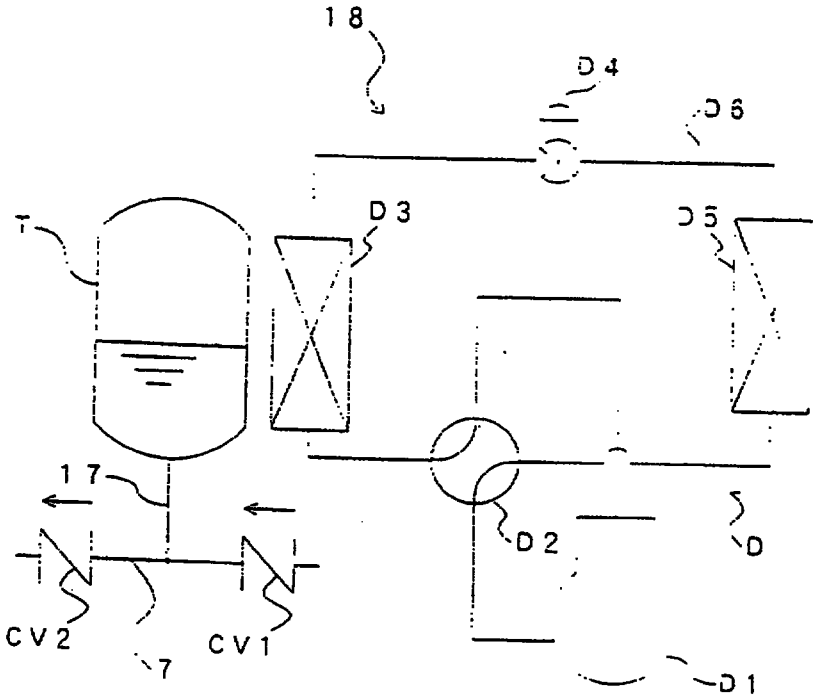


Fig. 6

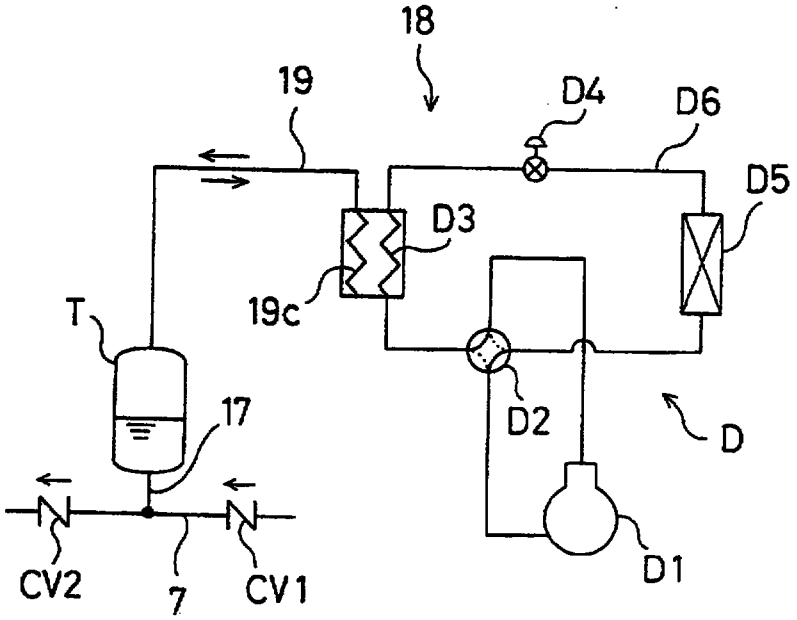


Fig. 7

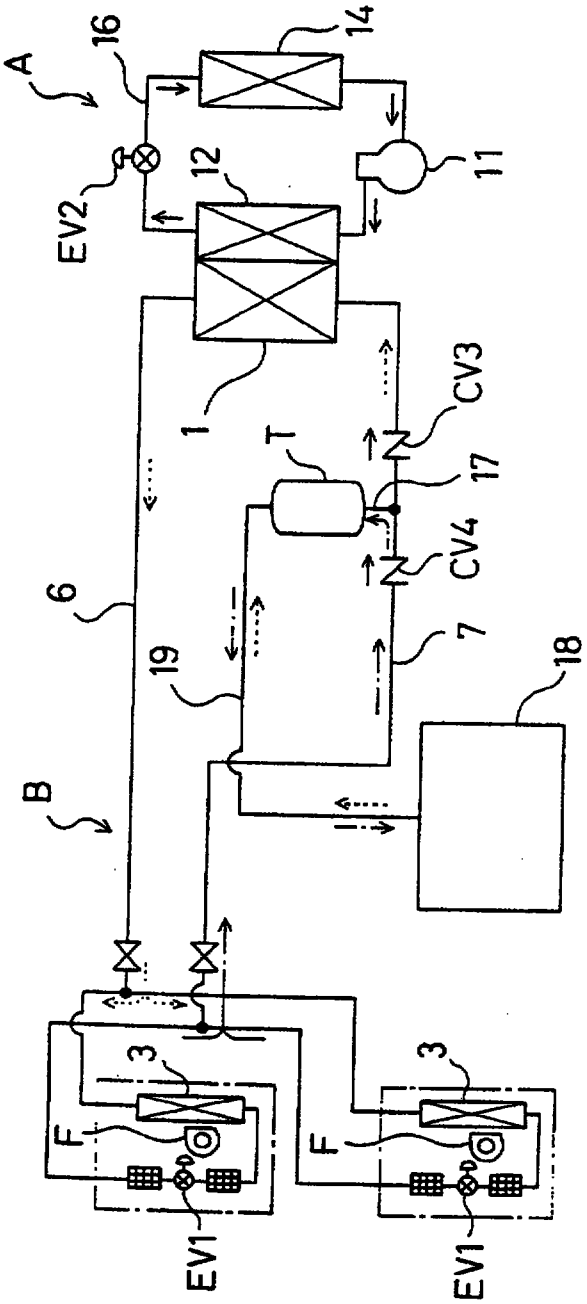


Fig. 8

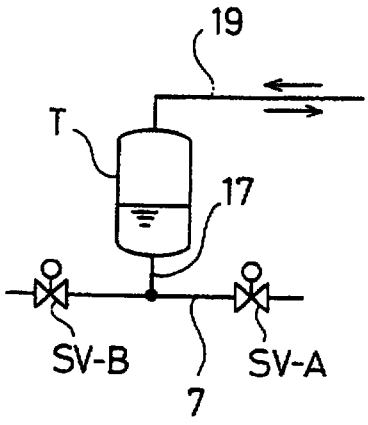


Fig. 9

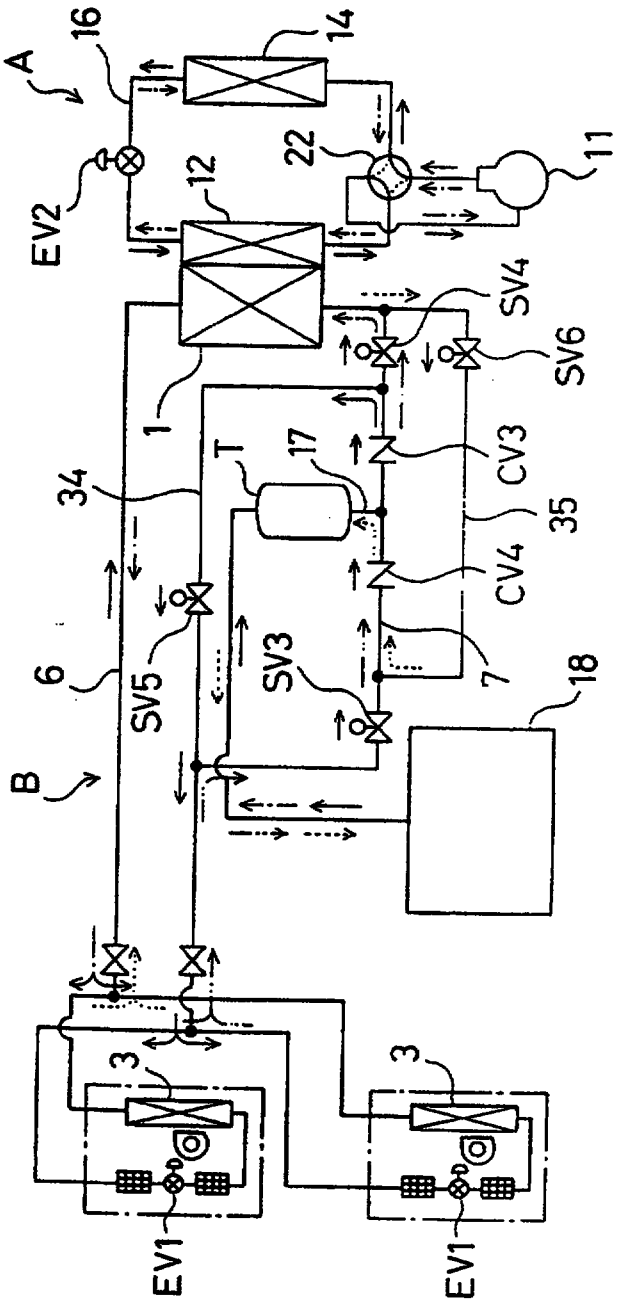

A
B
16
14
22
11
12
EV2
SV4
SV6
1
T
17
CV3
34
35
CV4
SV5
SV3
6
7
18
3
3
EV1
EV1

Fig. 10

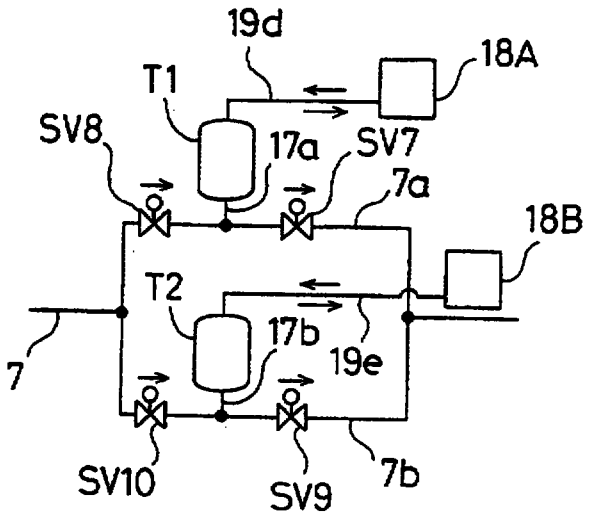

19d
18A
T1
SV8
17a
SV7
7a
18B
T2
17b
19e
7
SV10
SV9
7b

Fig. 11

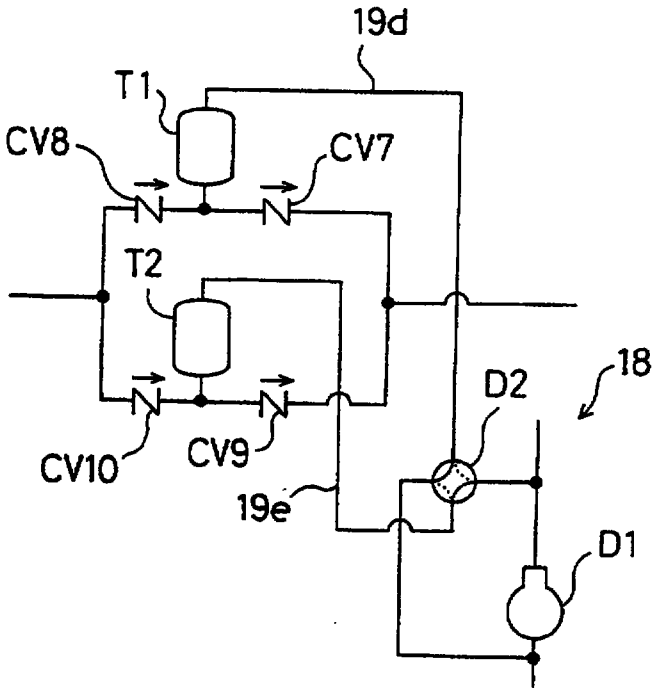

19d
T1
CV8
CV7
T2
18
D2
CV10
CV9
19e
D1

Fig. 12

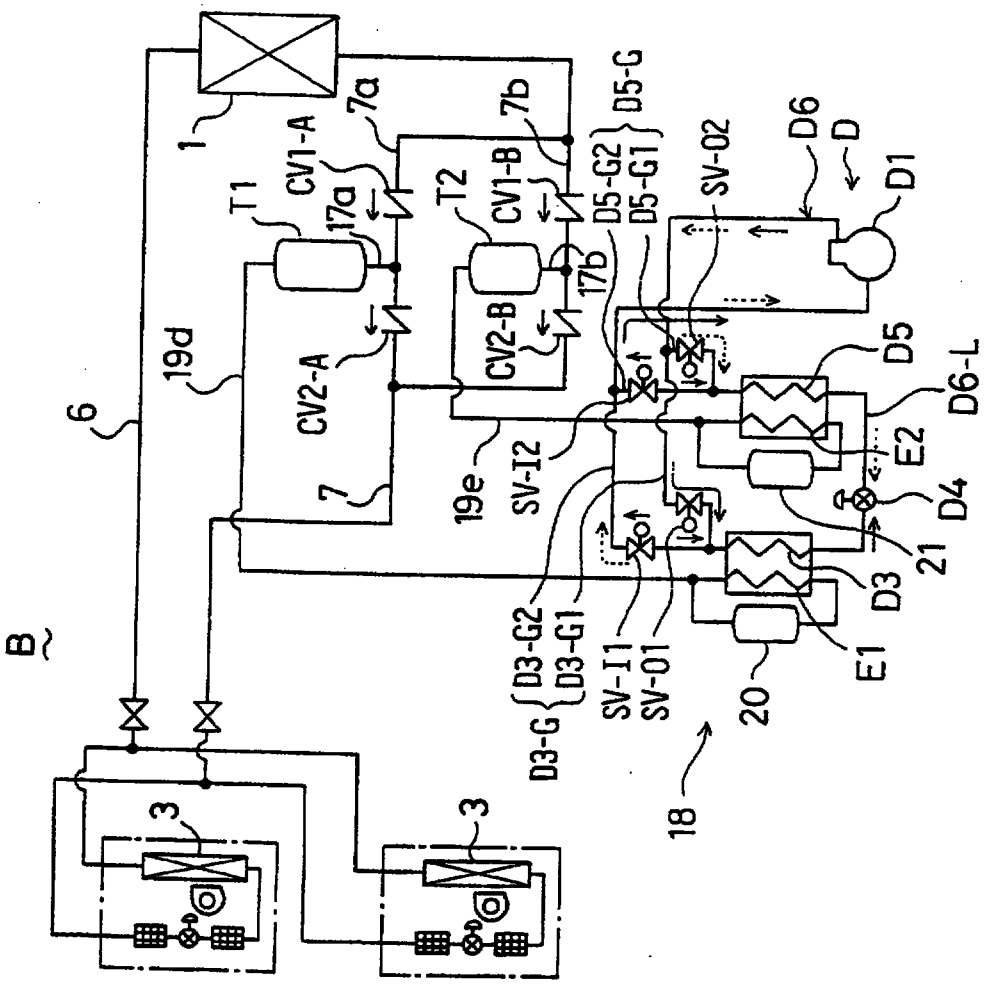

1
7a
7b
CV1-A
CV1-B
T1
T2
17a
17b
D5-G2
D5-G1
D5-G
SV-02
D6
D
D1
D5
D6-L
E2
D4
21
D3
E1
20
18
SV-01
SV-I1
D3-G1
D3-G2
D3-G
SV-I2
19e
CV2-B
CV2-A
7
19d
6
B
3
3

Fig. 13

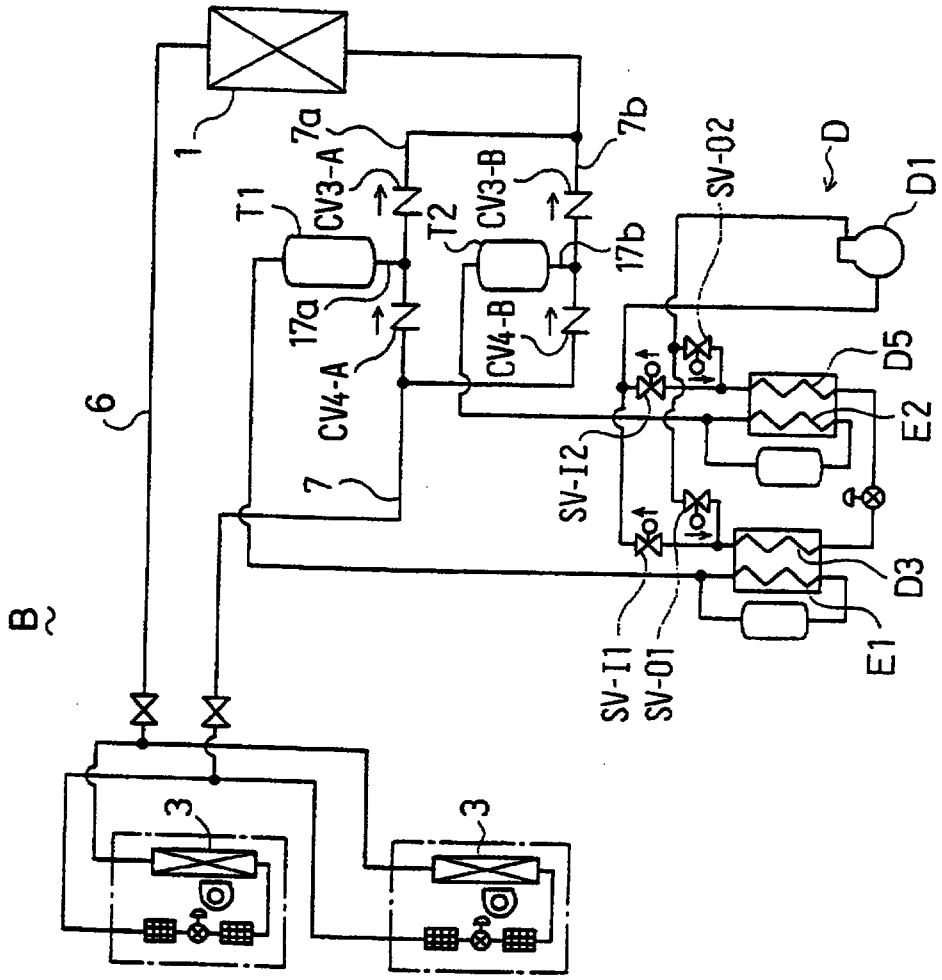

1
7a
7b
T1
CV3-A
T2
CV3-B
SV-02
D
D1
17b
17a
CV4-B
CV4-A
D5
6
7
SV-I2
E2
B
D3
SV-I1
SV-01
E1
3
3

Fig. 14

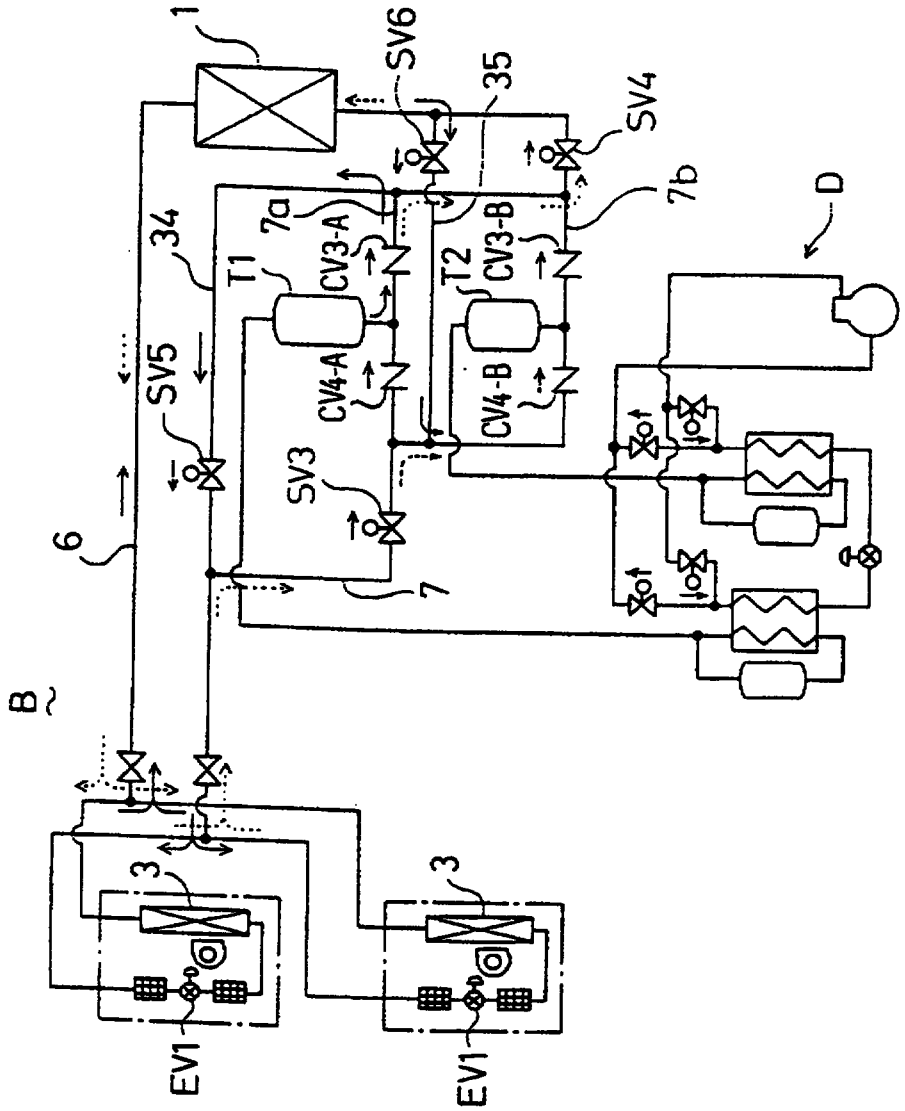

1
SV6
35
SV4
7b
D
34
7a
CV3-A
T1
T2
CV3-B
SV5
CV4-A
CV4-B
SV3
6
7
B
3
3
EV1
EV1

Fig. 15

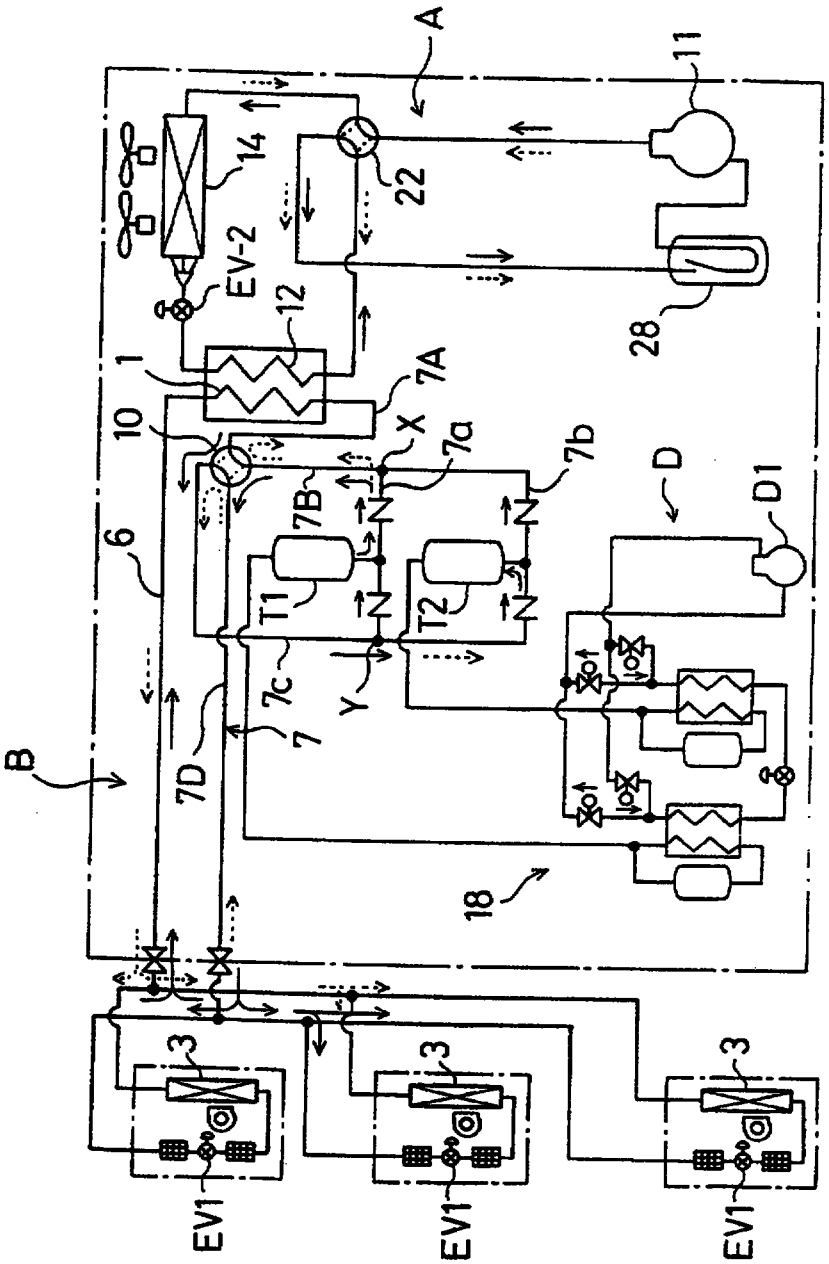


Fig. 16

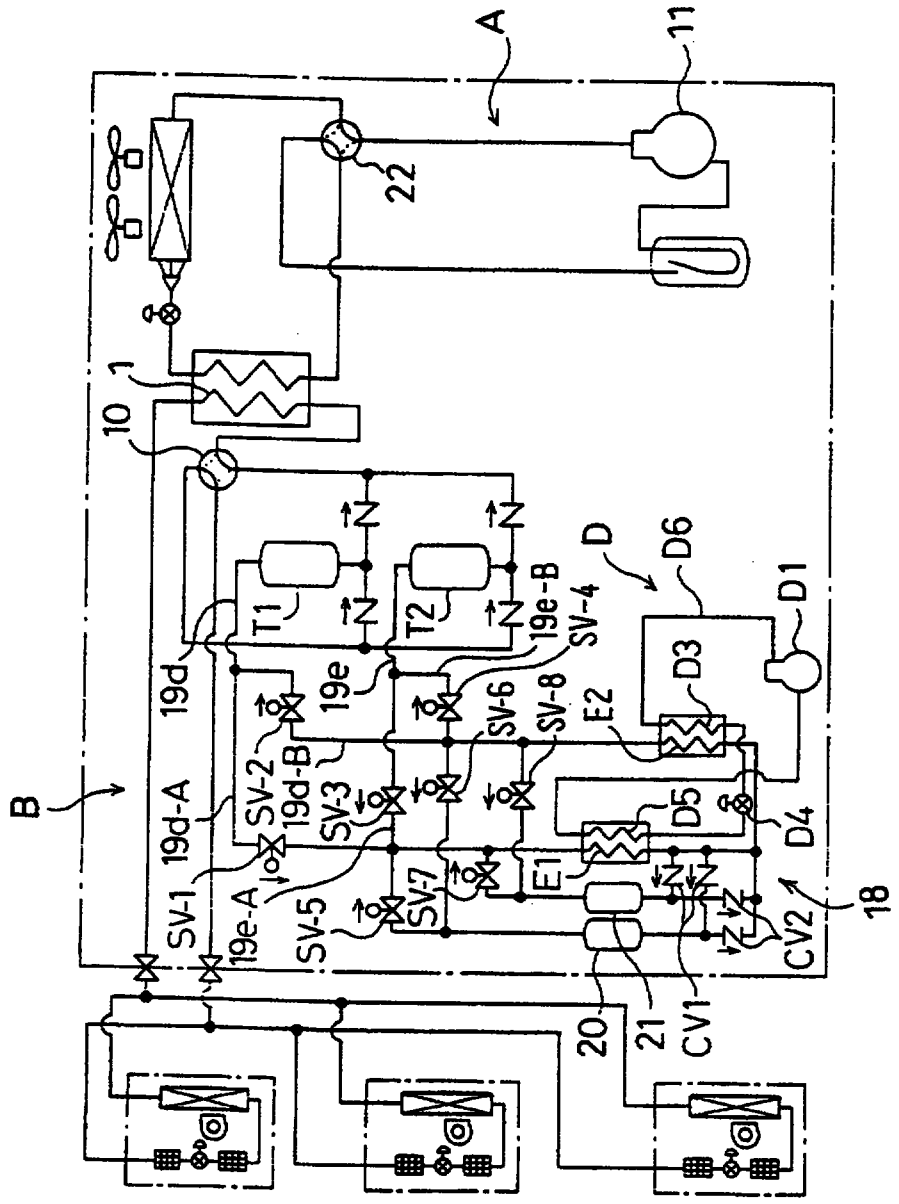

A
11
22
1
10
T1
T2
19d
19e
19e-B
SV-4
D
D6
D1
D3
SV-6
SV-8
E2
B
19d-A
SV-2
19d-B
SV-3
D5
D4
SV-1
19e-A
SV-5
SV-7
E1
CV2
18
20
21
CV1

Fig. 17

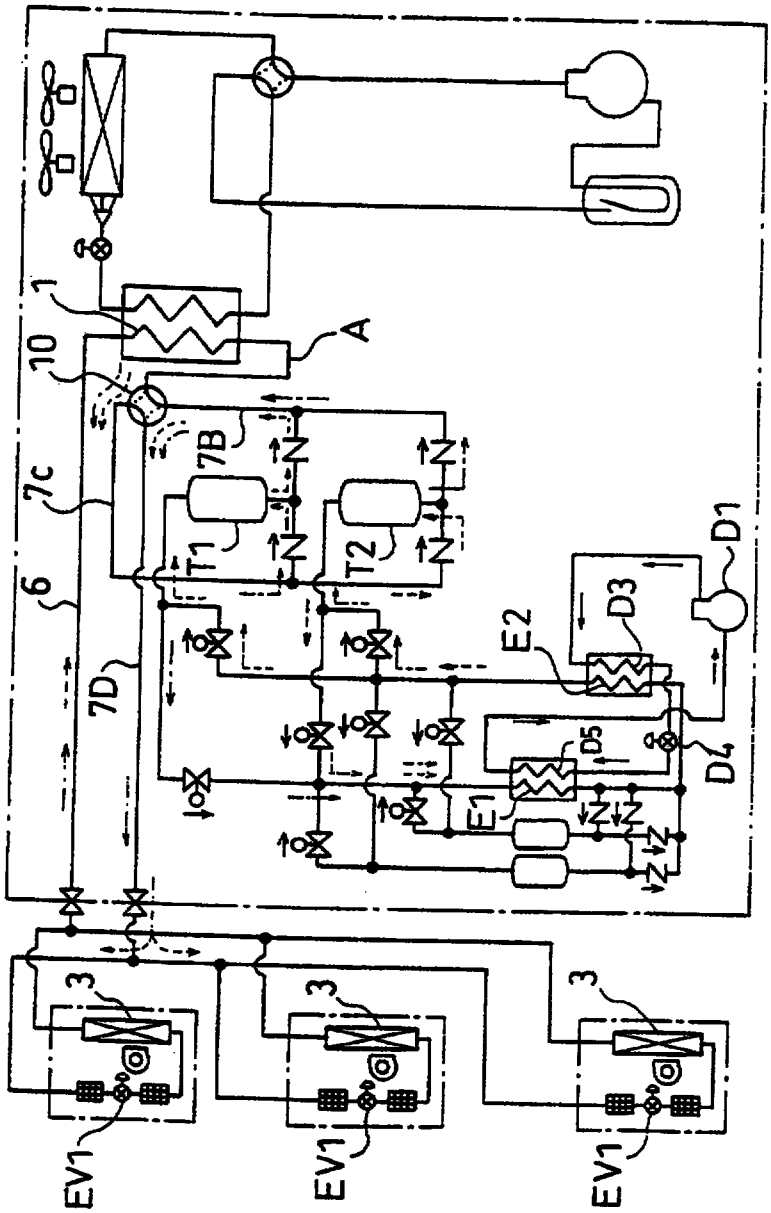


Fig. 18

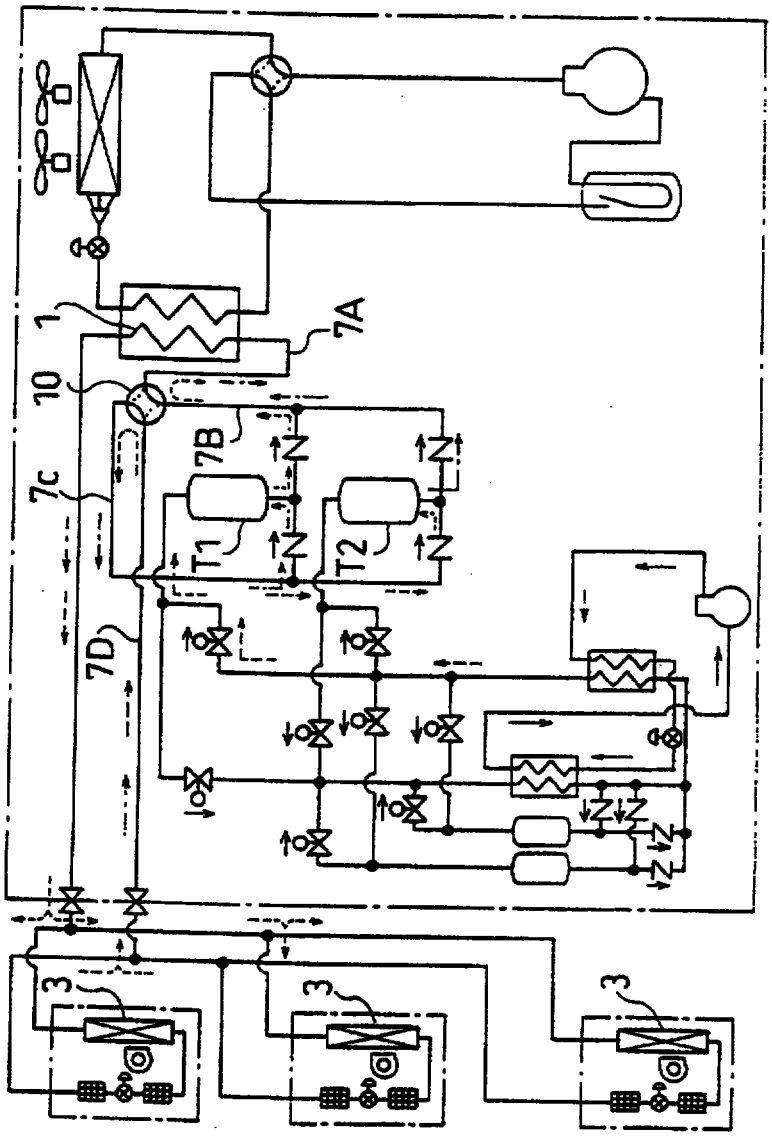


Fig. 19

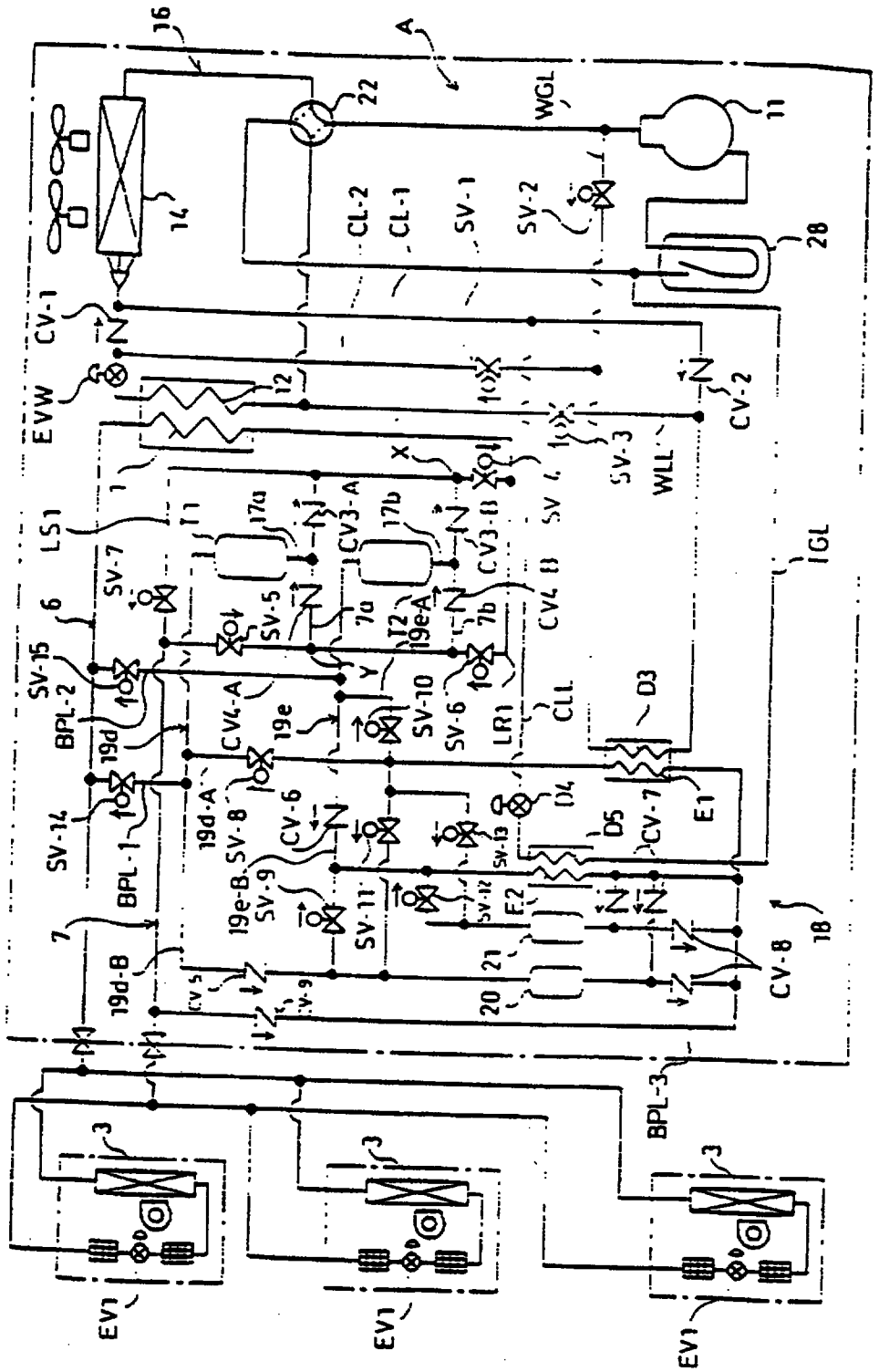

16
A
22
WGL
11
14
CL-2
CL-1
SV-1
SV-2
28
CV-1
EVW
12
CV-2
SV-3
WLL
X
LS1
SV-7
17a
CV3-A
17b
CV3-B
IGL
6
SV-5
7a
19e-A
7b
CV4-B
SV-15
Y
SV-10
SV-6
LR1
CLL
D3
BPL-2
19d
CV4-A
19e
19d-A
CV-6
SV-8
D4
SV-14
BPL-1
19e-B
SV-9
SV-13
D5
CV-7
E1
7
SV-11
SV-12
E2
19d-B
21
20
CV-8
18
BPL-3
3
EV1

Fig. 20

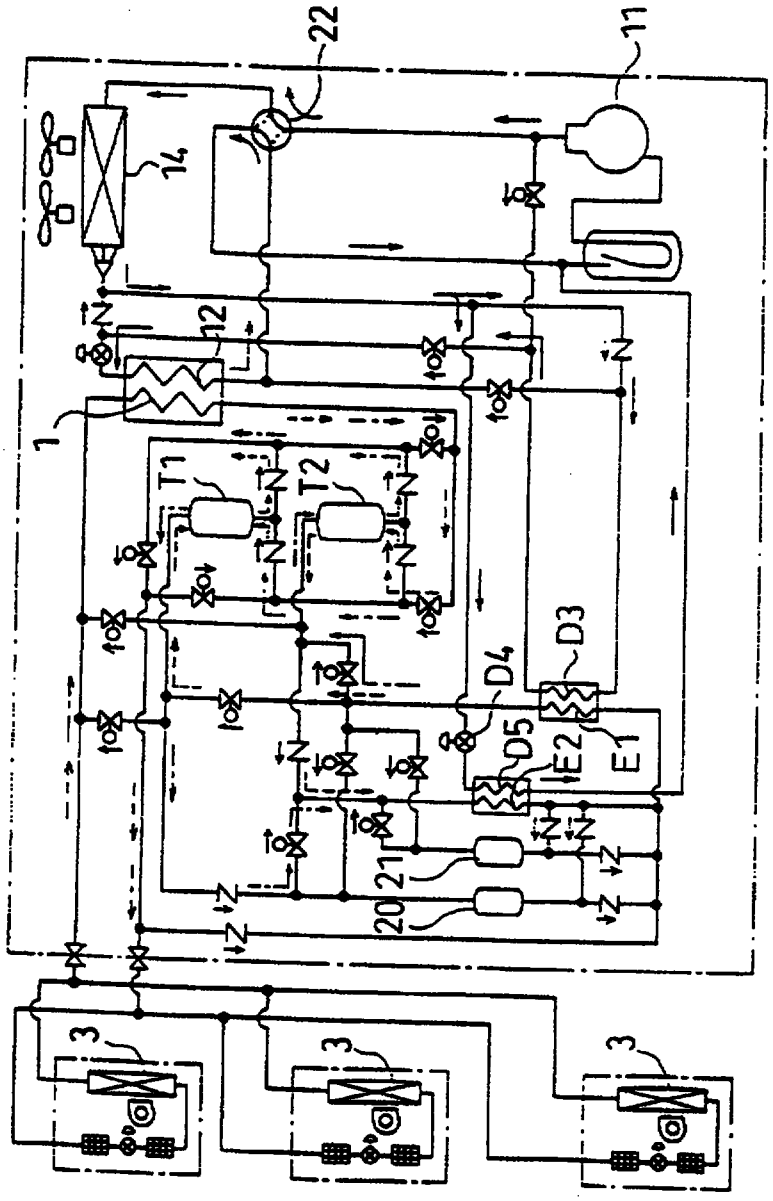


Fig. 21

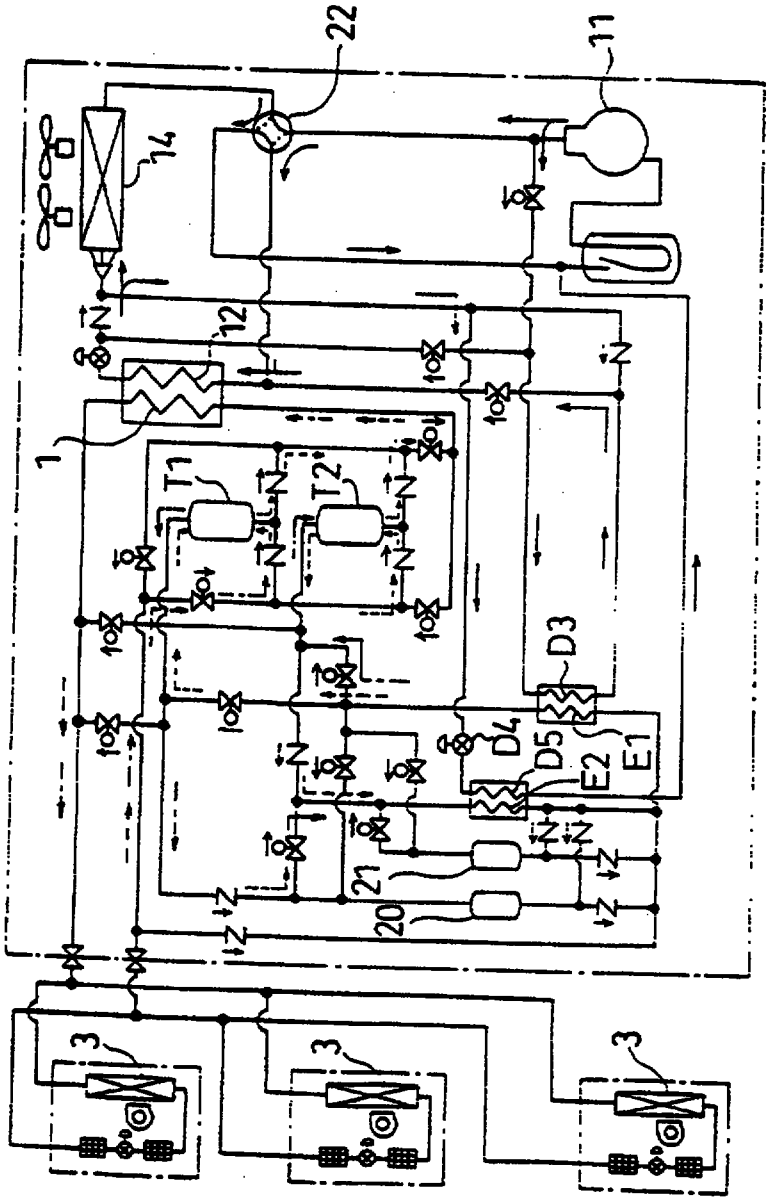


Fig. 22

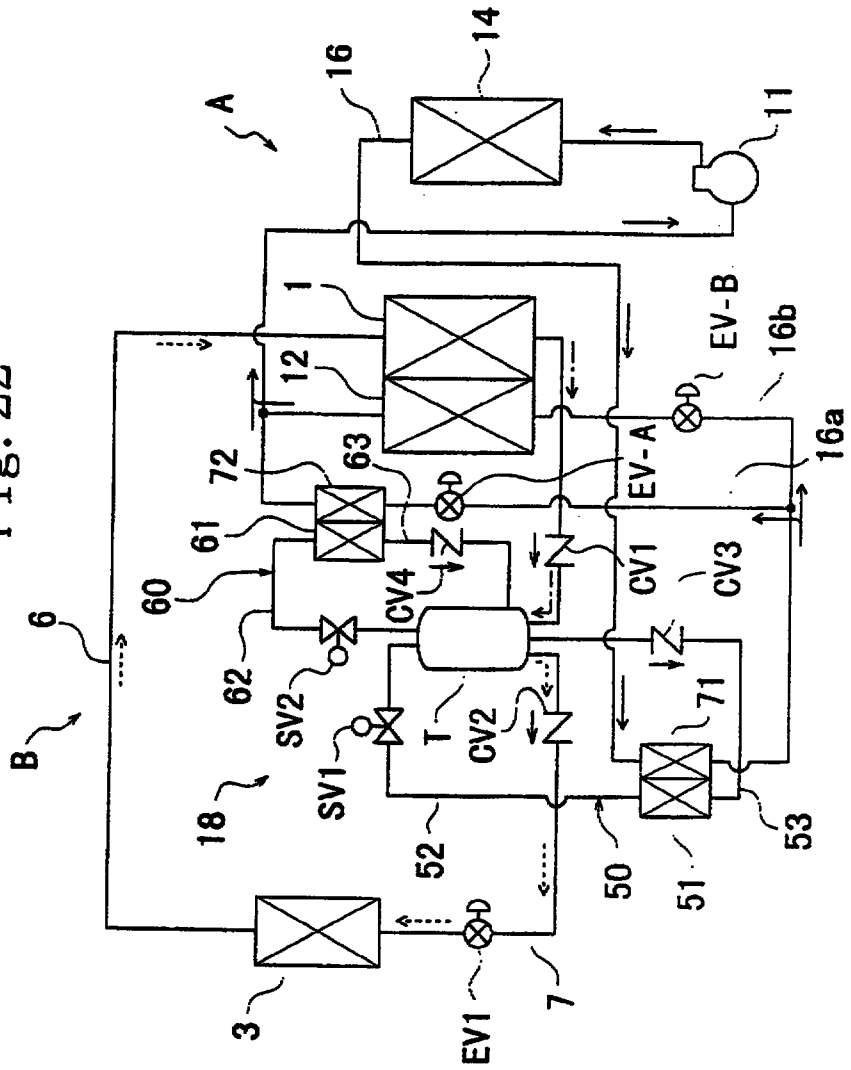

A
B
1
3
6
7
11
12
14
16
16a
16b
18
50
51
52
53
60
61
62
63
71
72
CV1
CV2
CV3
CV4
EV1
EV-A
EV-B
SV1
SV2
T

Fig. 23

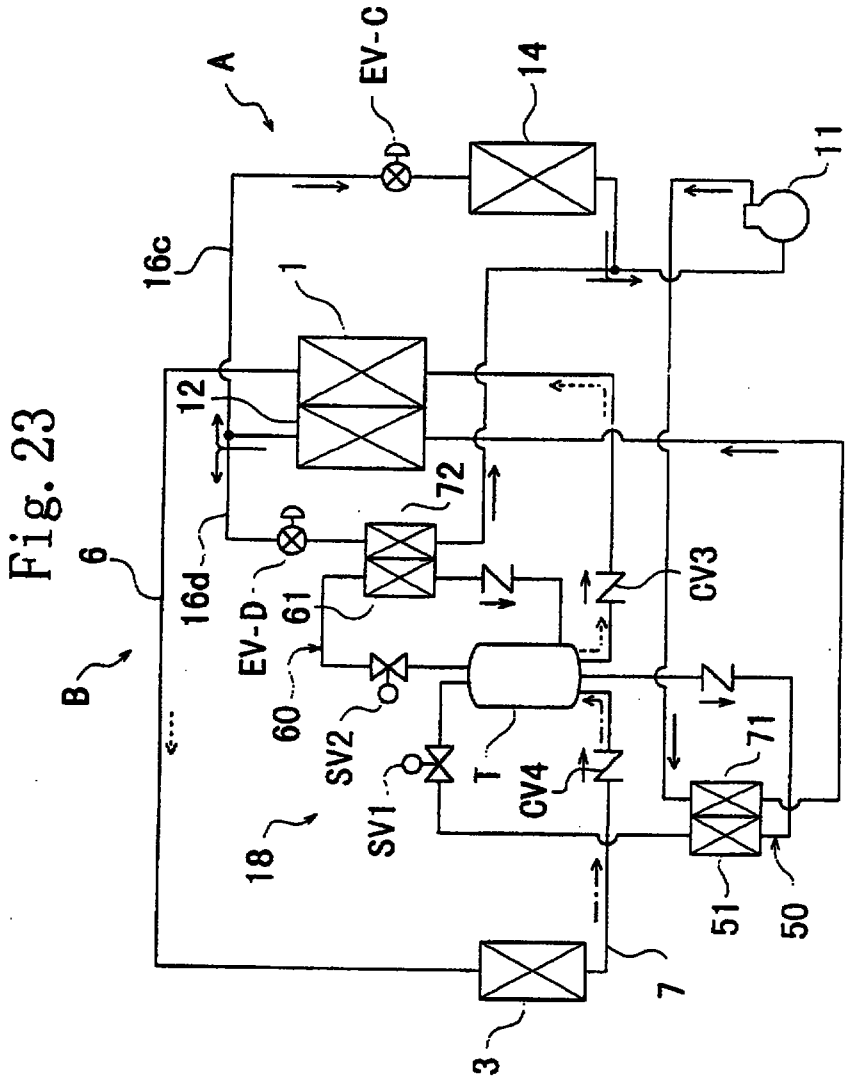

A
EV-C
14
11
16c
1
12
72
6
16d
EV-D
61
CV3
B
60
SV2
71
SV1
T
CV4
18
51
50
7
3

Fig. 24

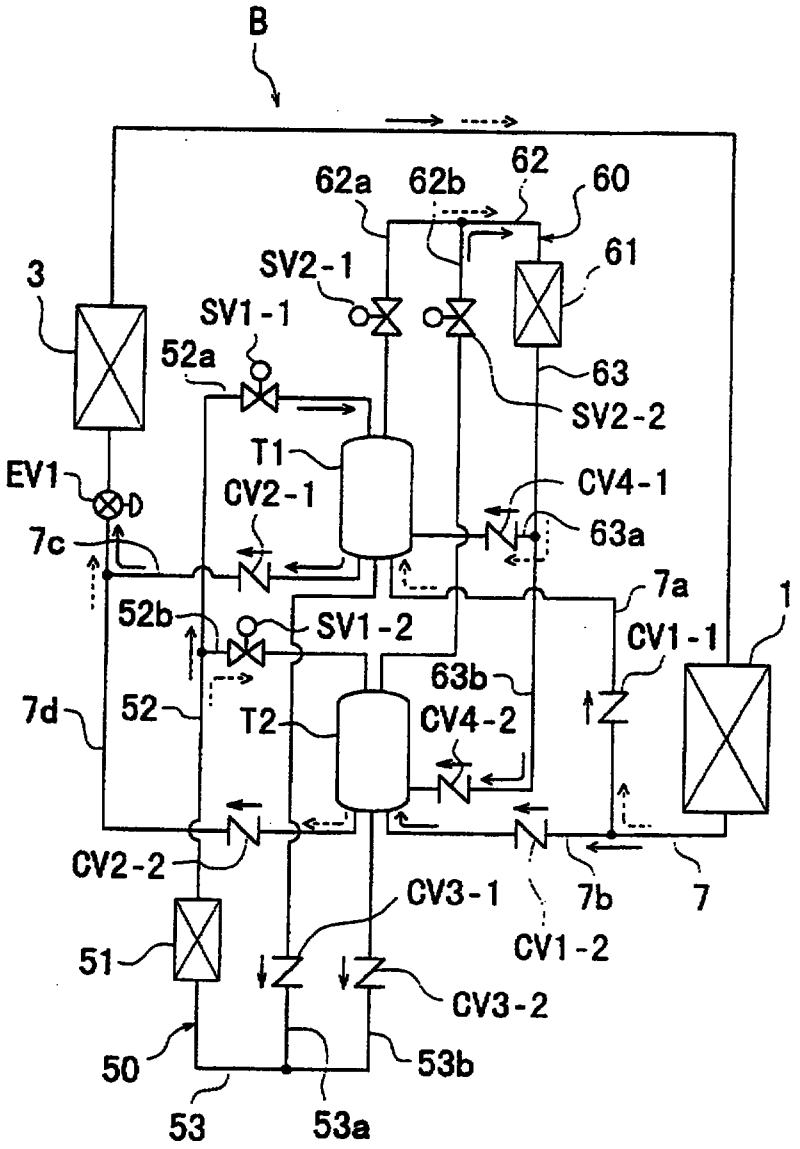

B
62a
62b
62
60
61
3
SV2-1
SV1-1
52a
63
SV2-2
T1
EV1
CV2-1
CV4-1
7c
63a
7a
52b
SV1-2
CV1-1
1
63b
7d
52
T2
CV4-2
CV2-2
CV3-1
7b
7
CV1-2
51
CV3-2
50
53b
53
53a

Fig. 25

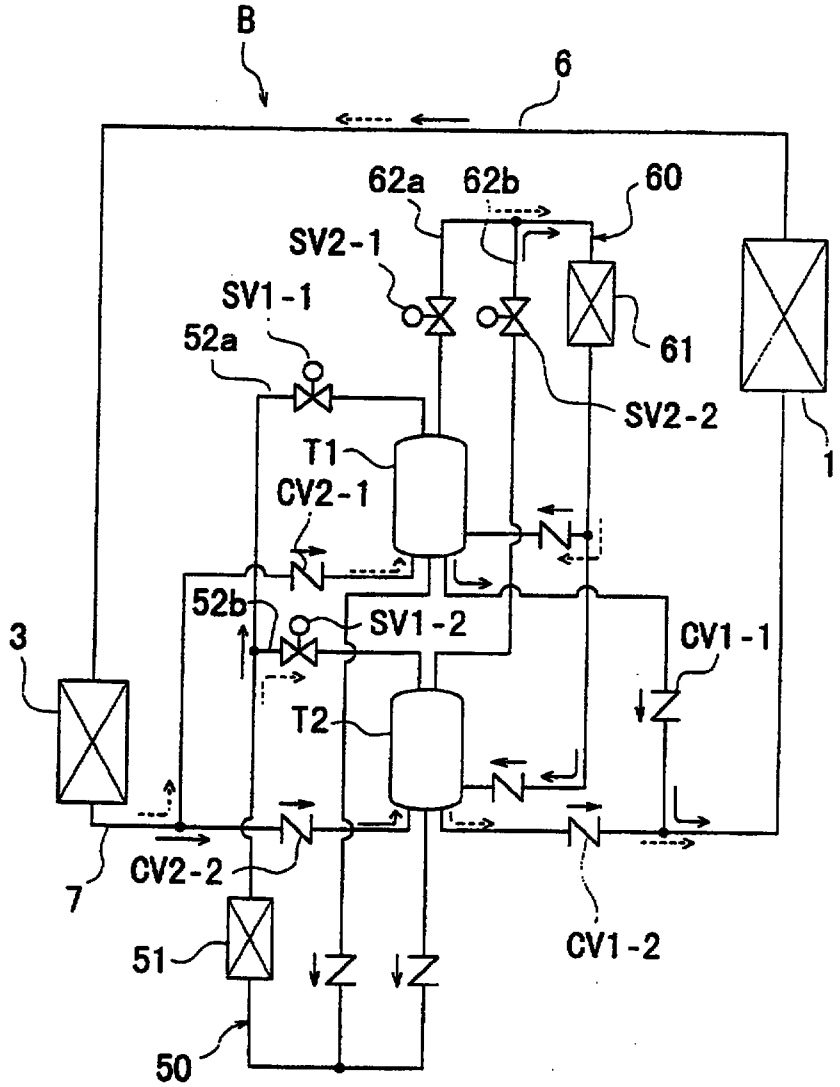

B
6
62a
62b
60
SV2-1
SV1-1
52a
61
SV2-2
1
T1
CV2-1
52b
SV1-2
3
CV1-1
T2
7
CV2-2
51
CV1-2
50

Fig. 26

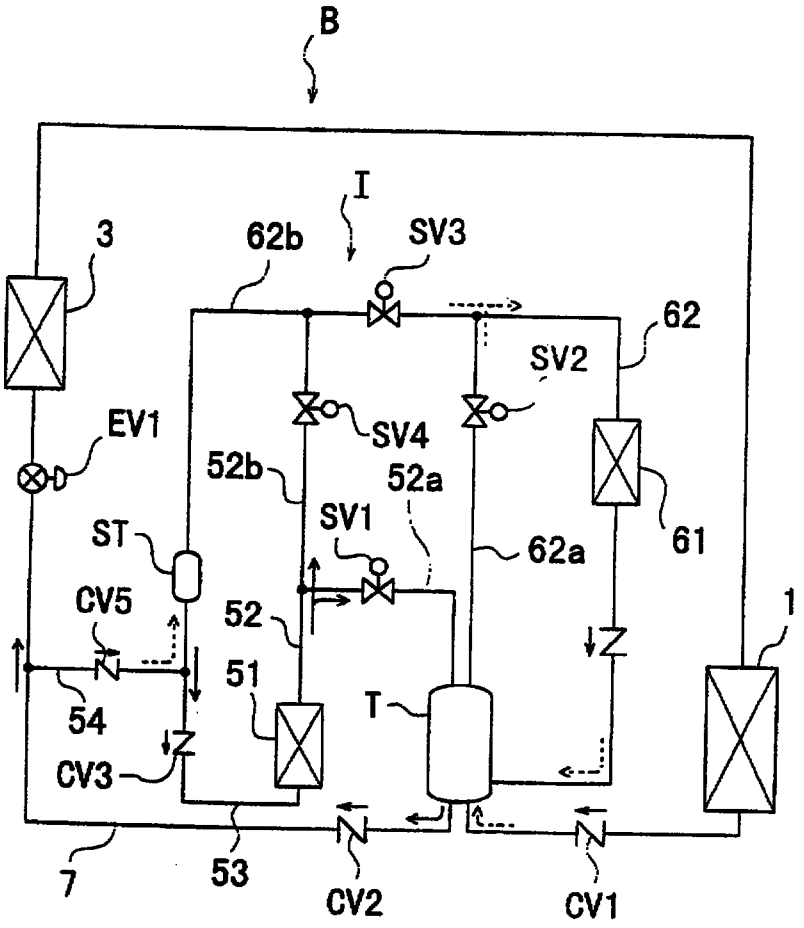

B
I
3
62b
SV3
62
SV2
SV4
EV1
52b
52a
SV1
61
ST
62a
CV5
52
1
51
54
T
CV3
7
53
CV2
CV1

Fig. 27

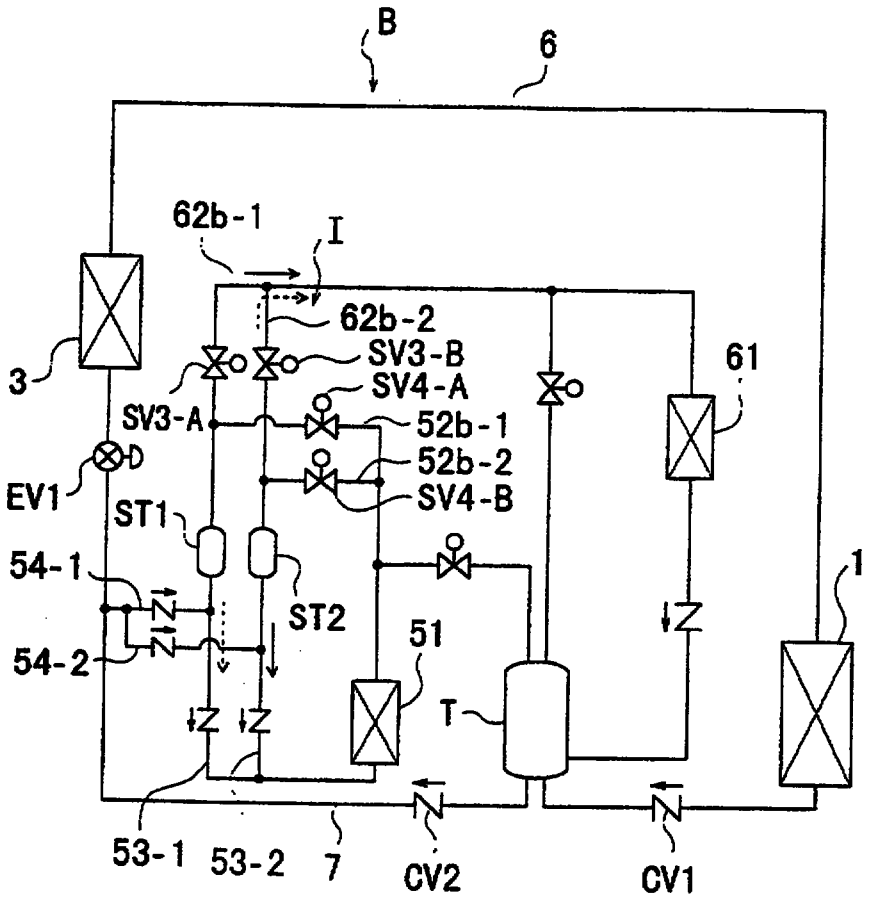

B
6
62b-1
I
62b-2
SV3-B
SV4-A
3
SV3-A
52b-1
61
52b-2
EV1
SV4-B
ST1
54-1
ST2
1
54-2
51
T
53-1
53-2
7
CV2
CV1

Fig. 28

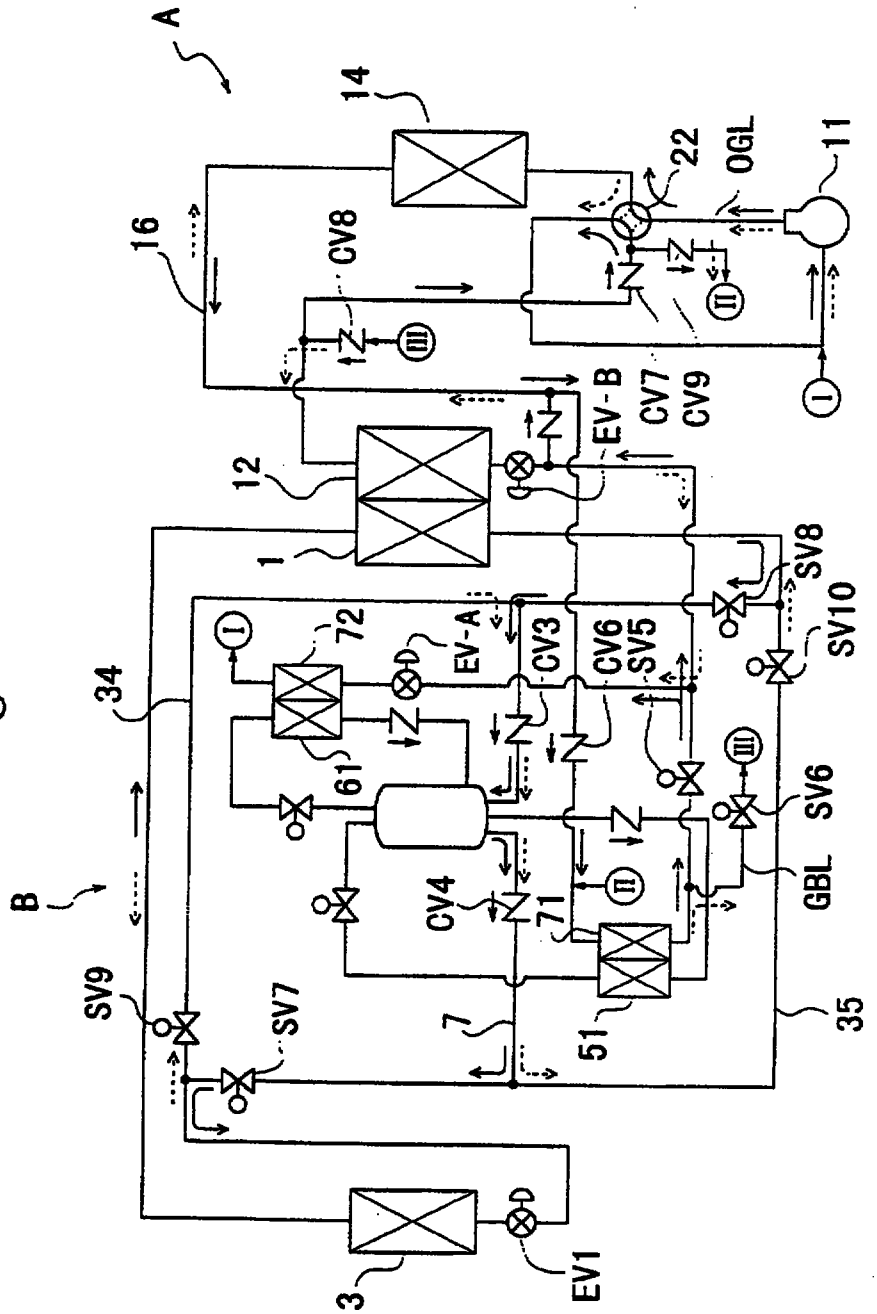

A
B
14
16
22
OGL
11
CV8
III
II
I
EV-B
CV7
CV9
12
1
SV8
SV10
72
EV-A
CV3
CV6
SV5
34
61
SV6
GBL
CV4
71
SV9
SV7
7
51
35
3
EV1

Fig. 29

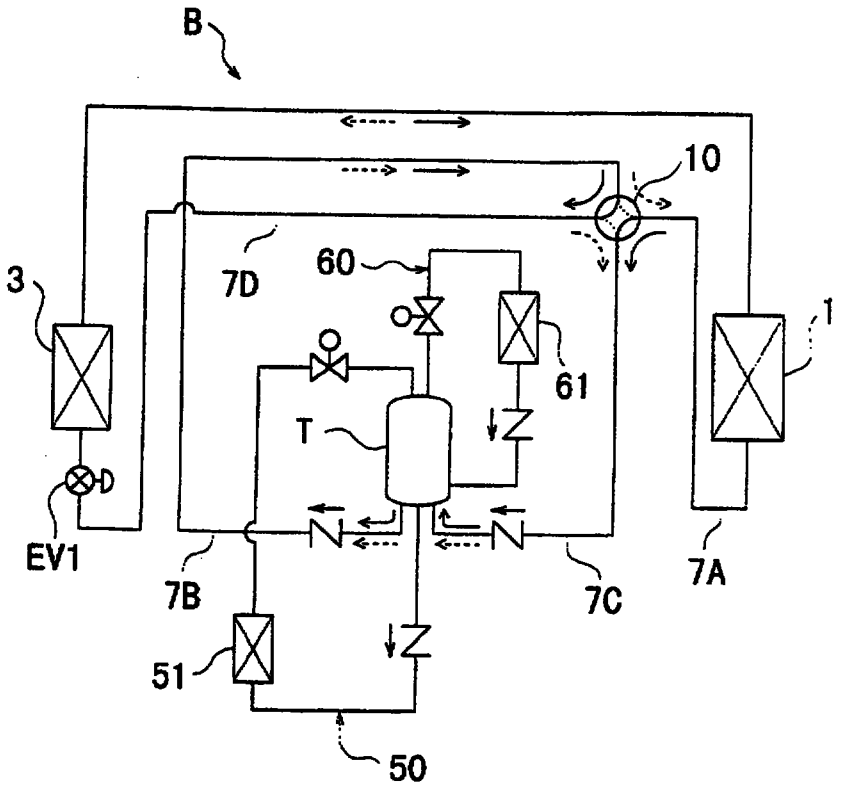


Fig. 30

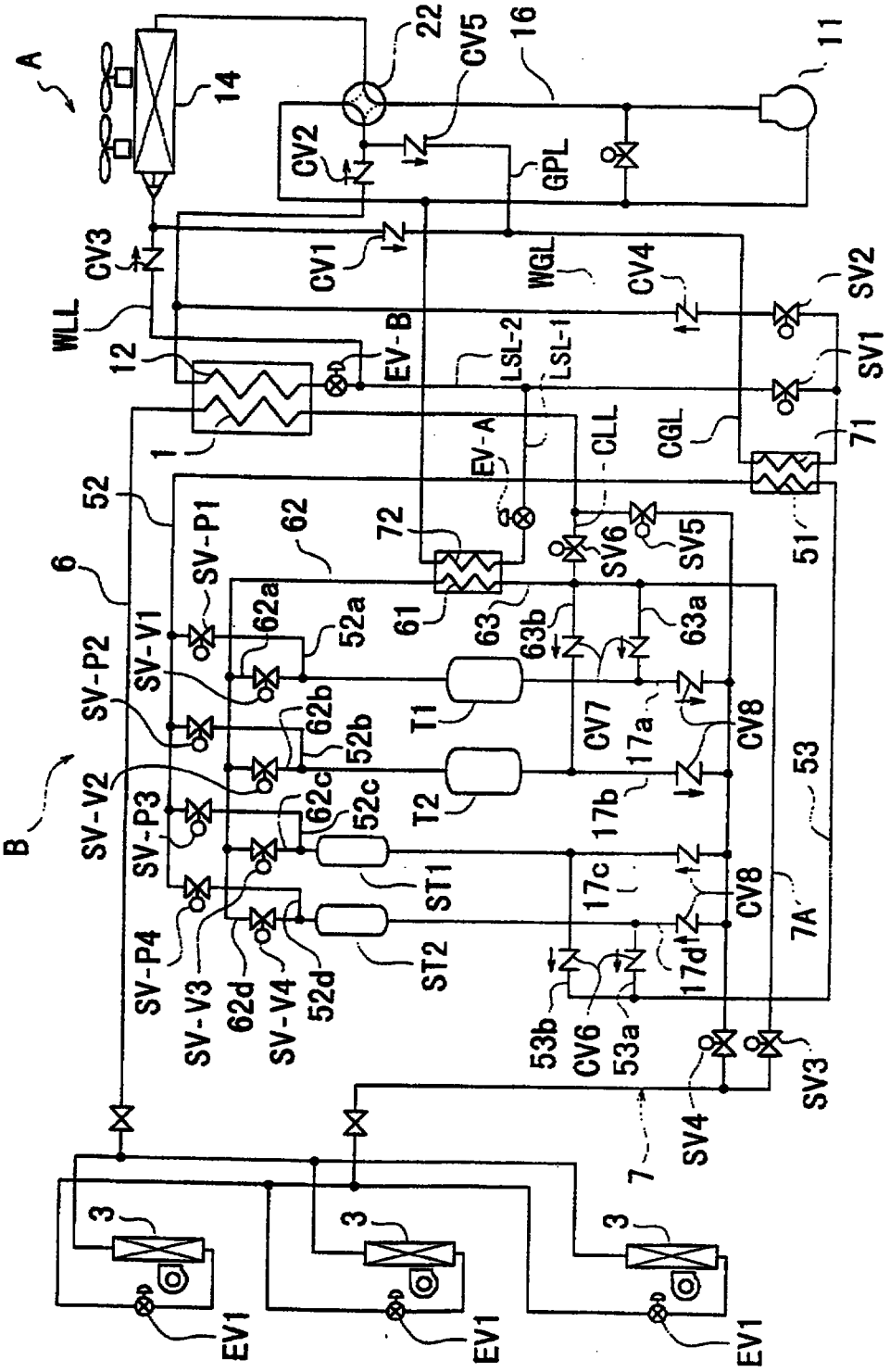

A
B
14
22
CV5
16
11
CV2
GPL
CV3
CV1
WGL
CV4
SV2
WLL
12
EV-B
LSL-2
LSL-1
SV1
1
EV-A
CLL
CGL
71
52
SV-P1
62
72
SV6
SV5
51
6
62a
52a
61
63
63b
63a
SV-V1
SV-P2
T1
CV7
CV8
62b
52b
17a
SV-V2
SV-P3
62c
52c
T2
17b
53
ST1
17c
CV8
SV-P4
ST2
17d
7A
SV-V3
62d
SV-V4
52d
53b
CV6
53a
SV3
SV4
7
3
EV1

Fig. 31

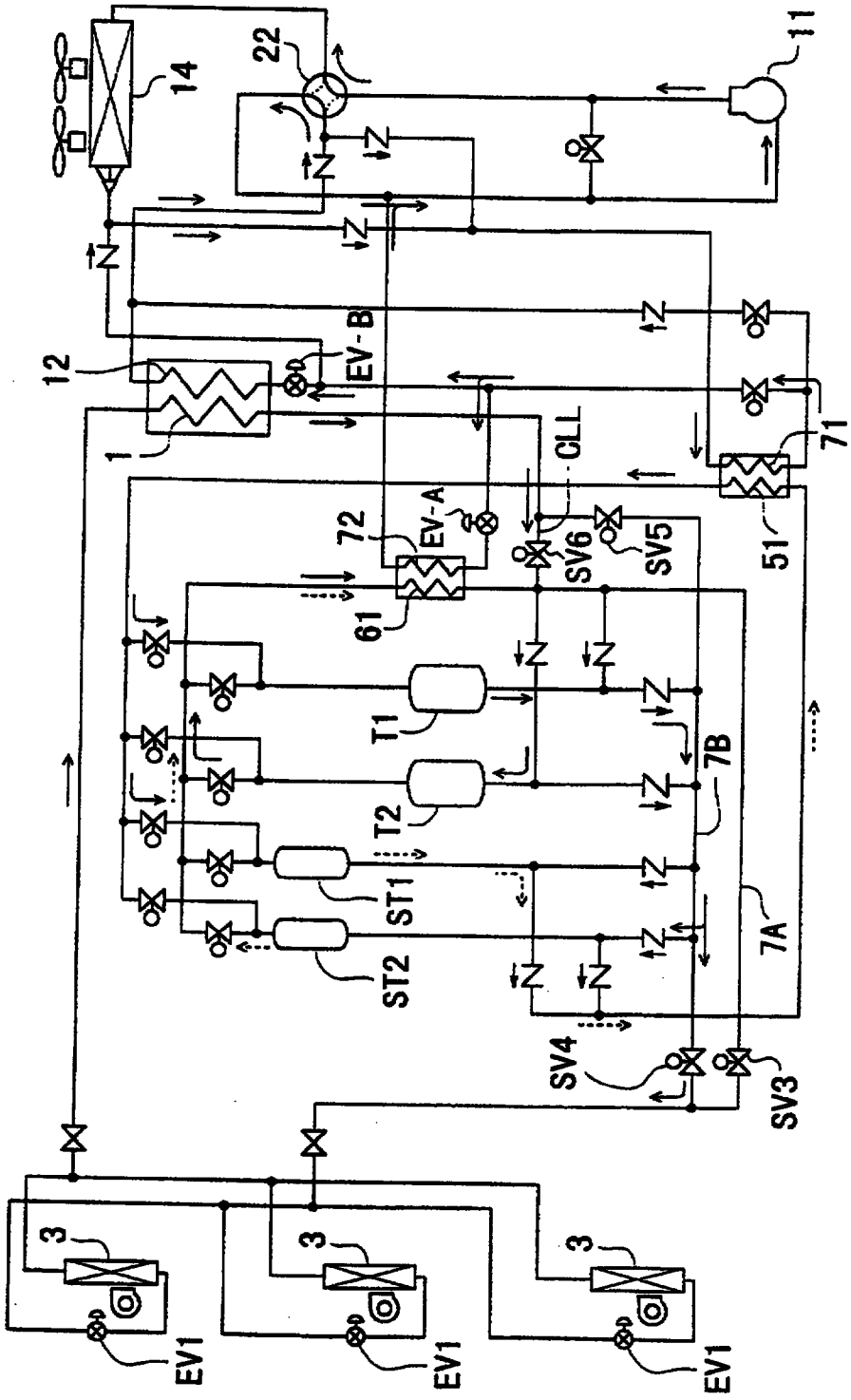

11
22
14
EV-B
12
1
CLL
71
72
EV-A
SV6
SV5
51
61
T1
7B
T2
ST1
ST2
7A
SV4
SV3
3
EV1

Fig. 32

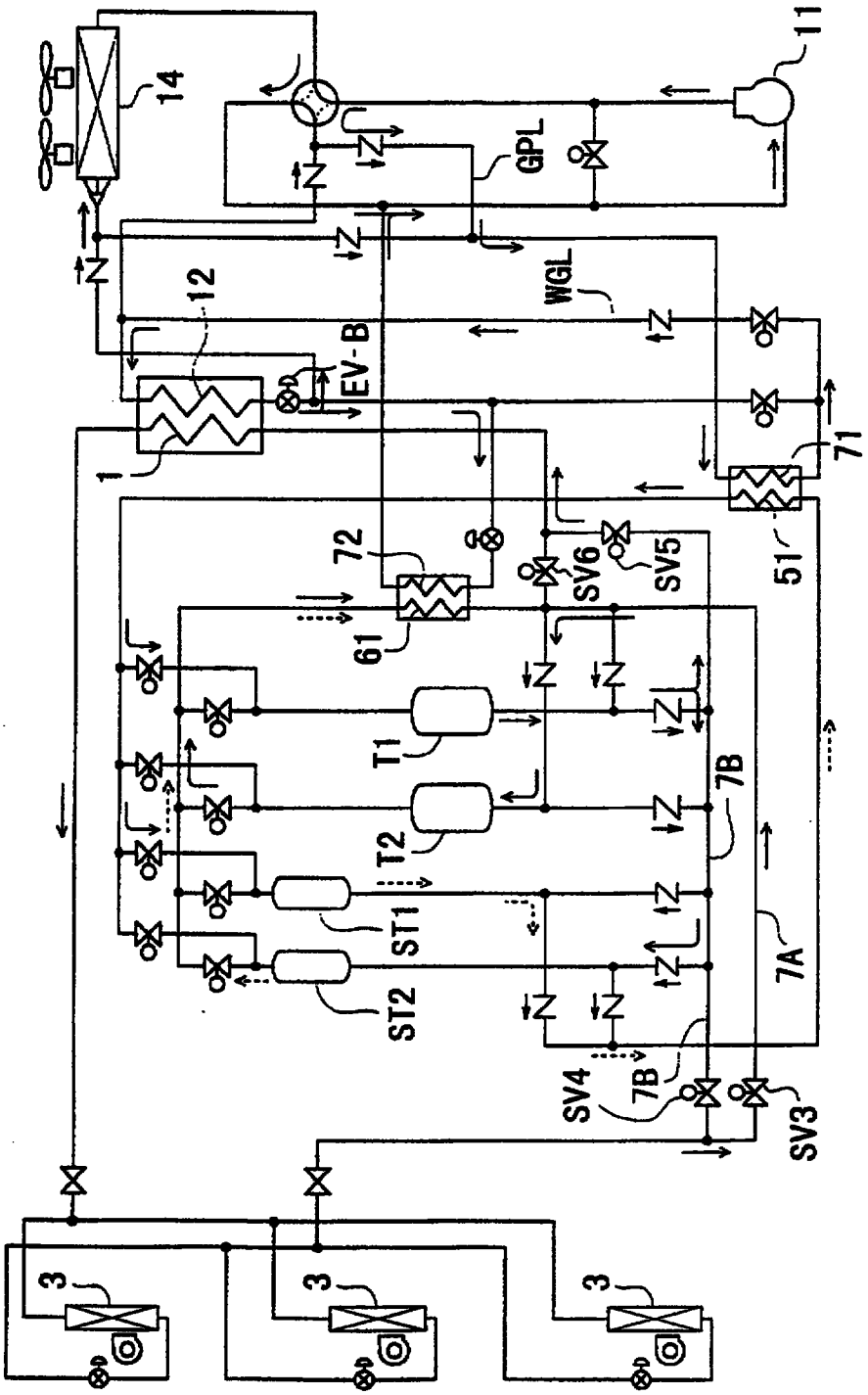

14
11
GPL
12
WGL
EV-B
1
71
72
SV6
SV5
51
61
T1
7B
T2
ST1
ST2
7A
SV4
7B
SV3
3
3
3

Fig. 33

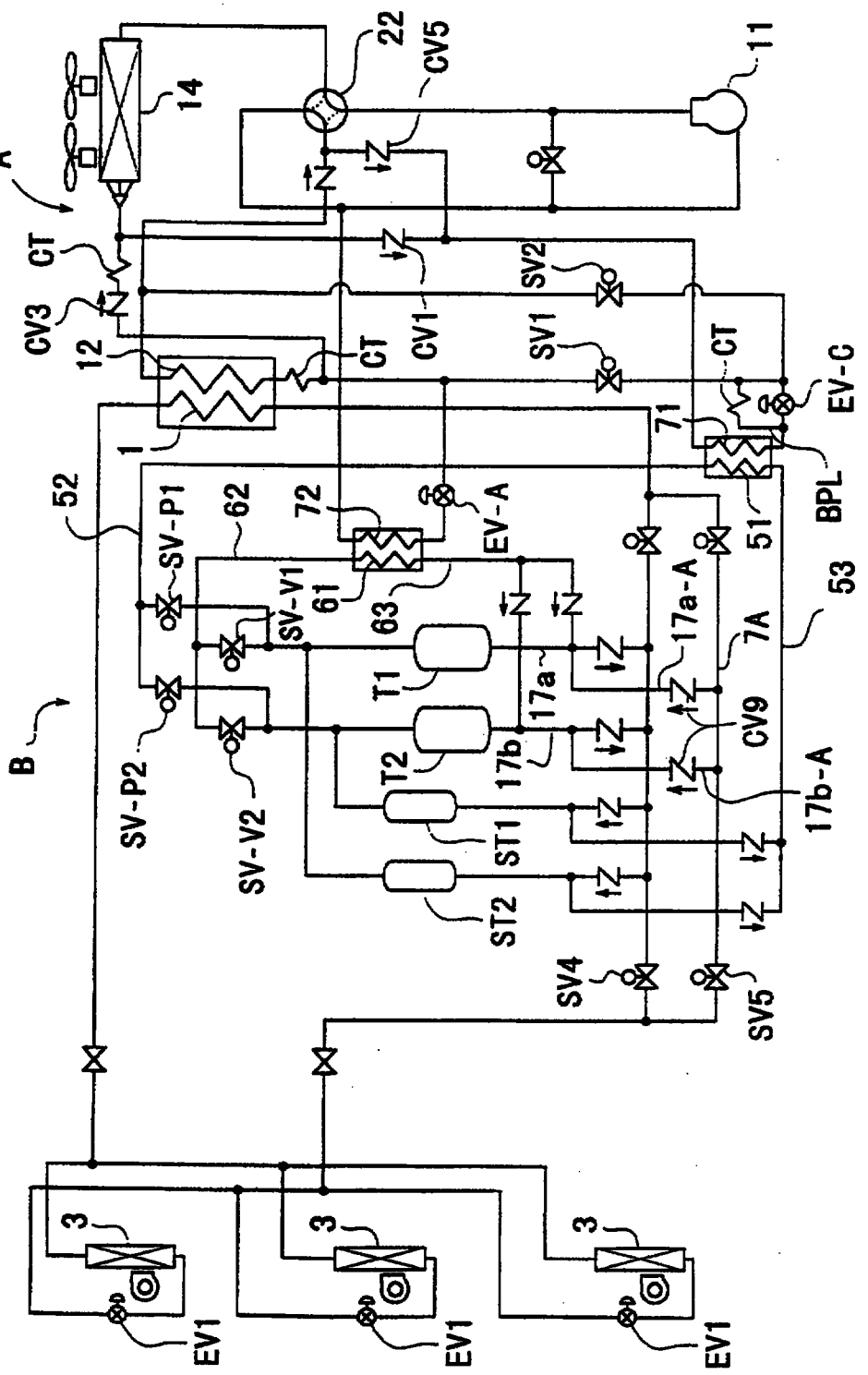

A
B
14
22
CV5
11
CT
CV3
12
CV1
SV2
SV1
EV-C
BPL
71
1
52
SV-P1
62
72
EV-A
61
63
SV-V1
T1
T2
ST1
ST2
17a
17b
17a-A
17b-A
7A
53
CV9
51
SV-P2
SV-V2
SV4
SV5
3
EV1

Fig. 34

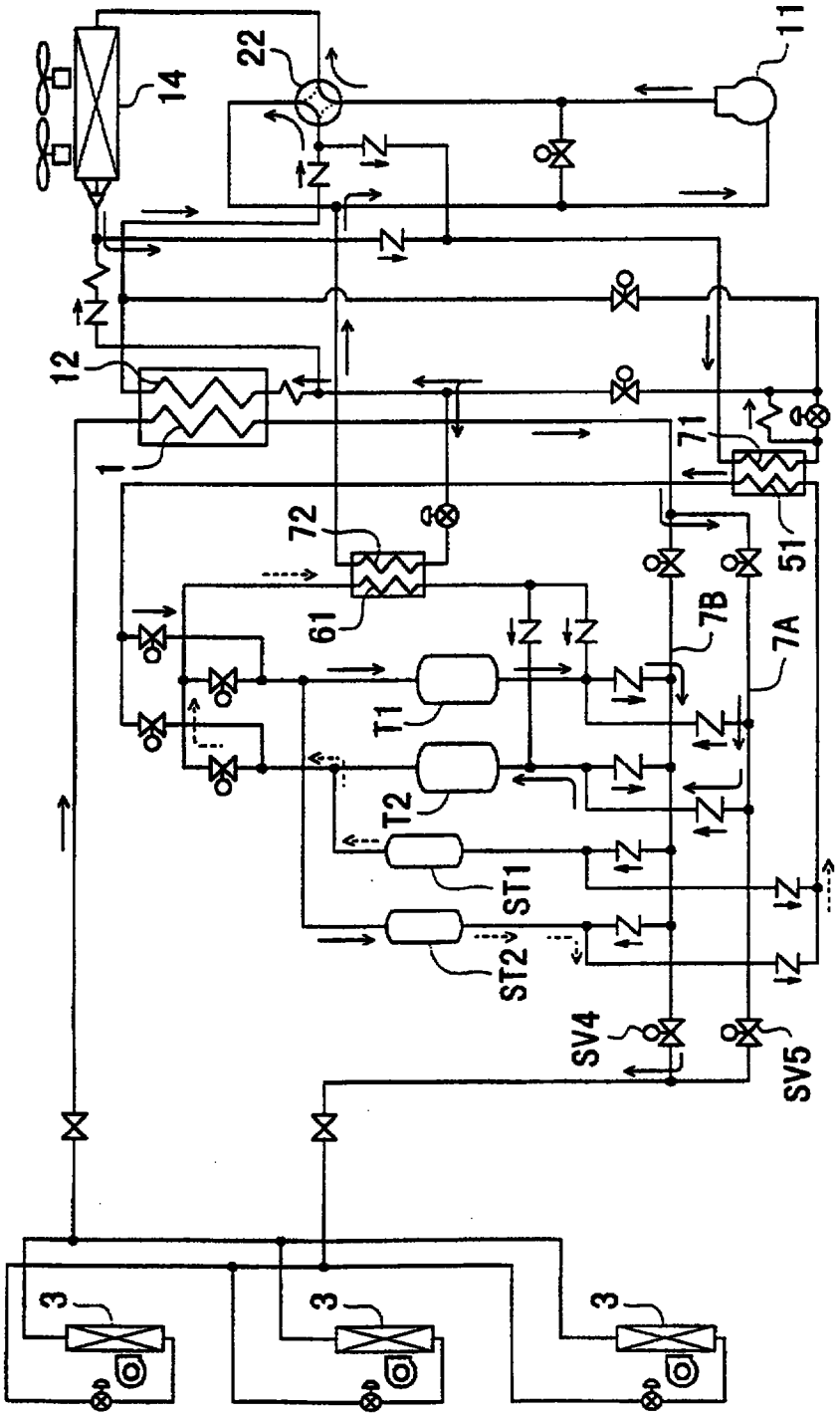


Fig. 35

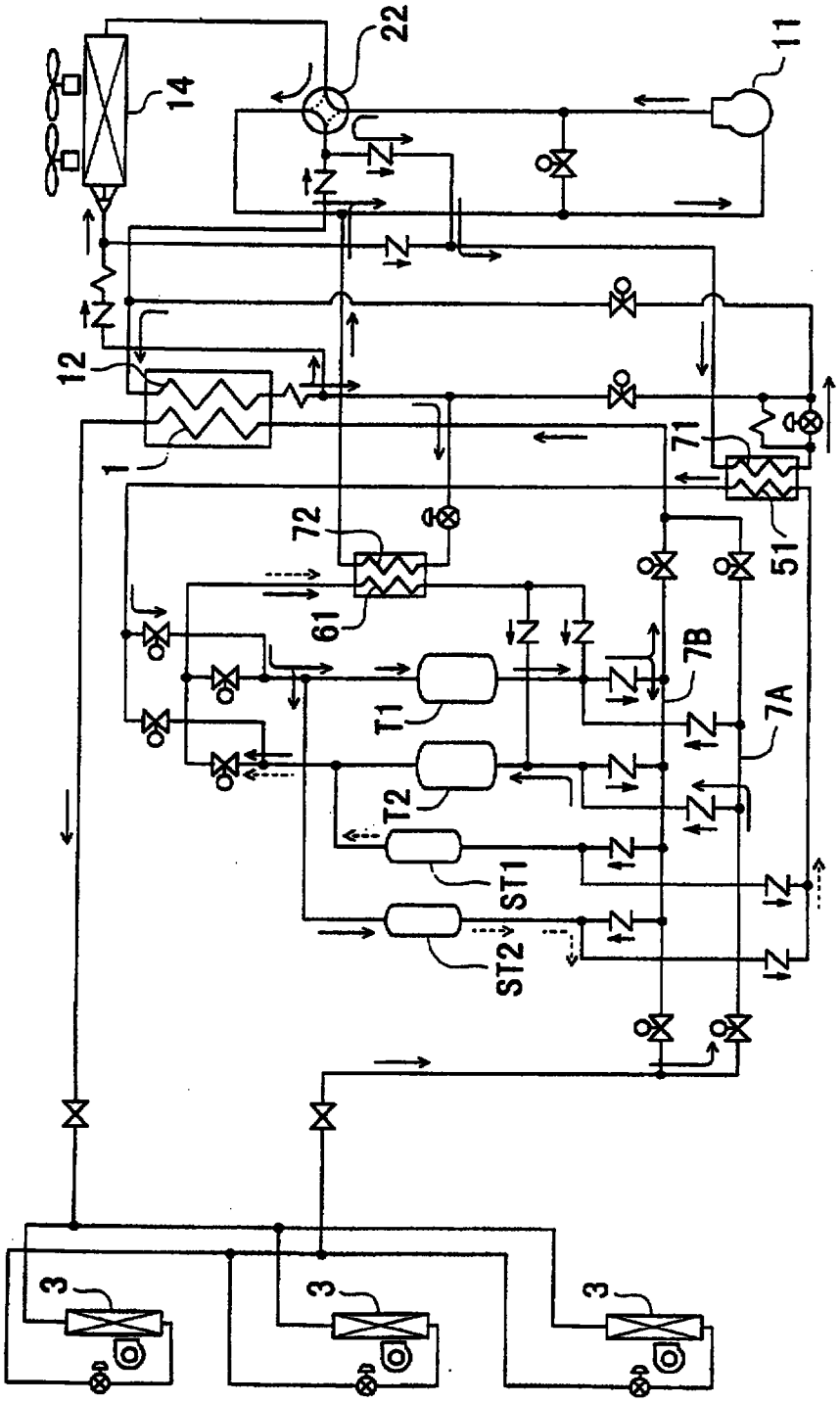


Fig. 36

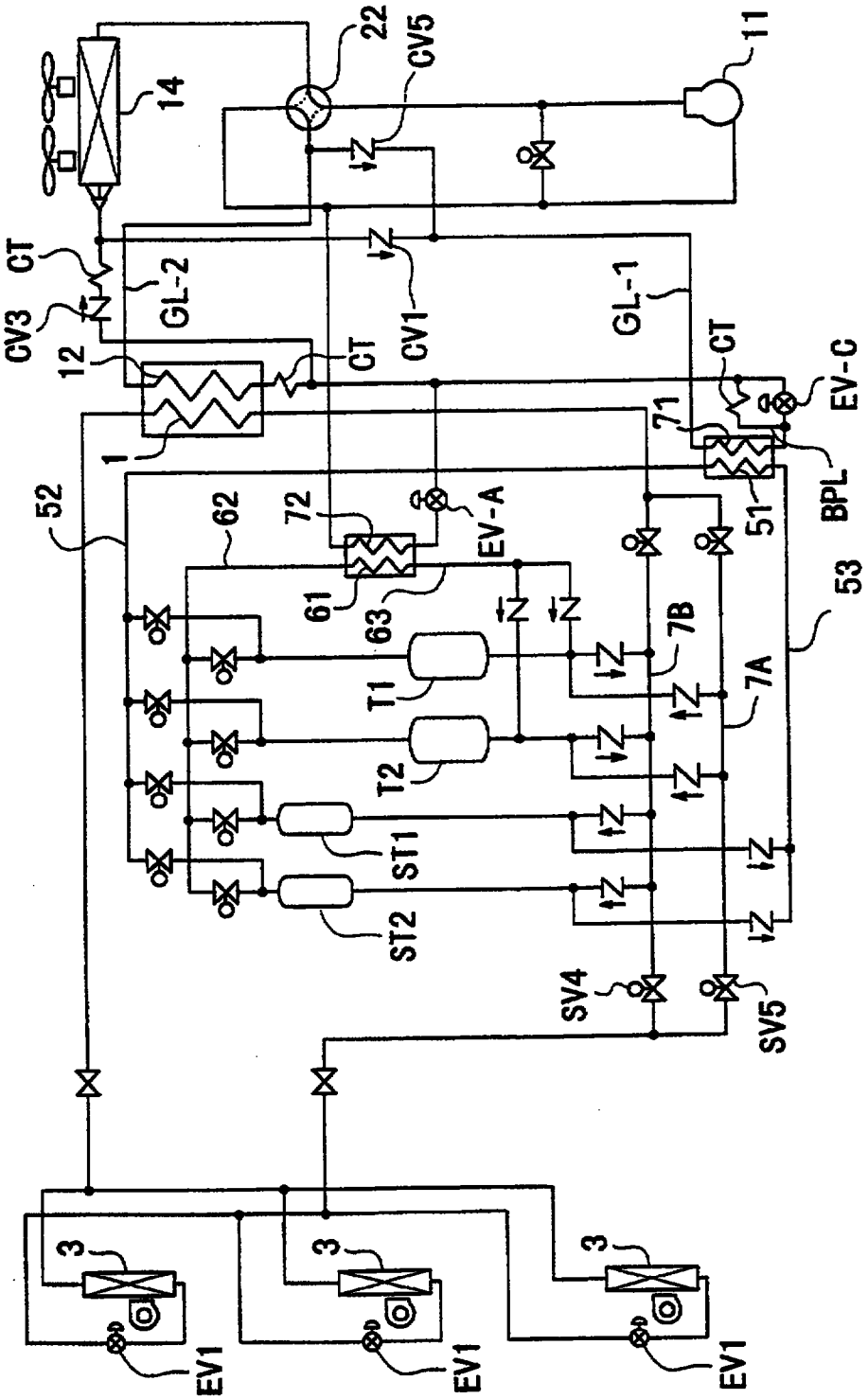

22
CV5
11
14
CT
CV3
GL-2
CV1
GL-1
12
CT
CT
EV-C
71
1
BPL
52
62
72
EV-A
51
53
61
63
7B
T1
7A
T2
ST1
ST2
SV4
SV5
3
3
3
EV1
EV1
EV1

Fig. 37

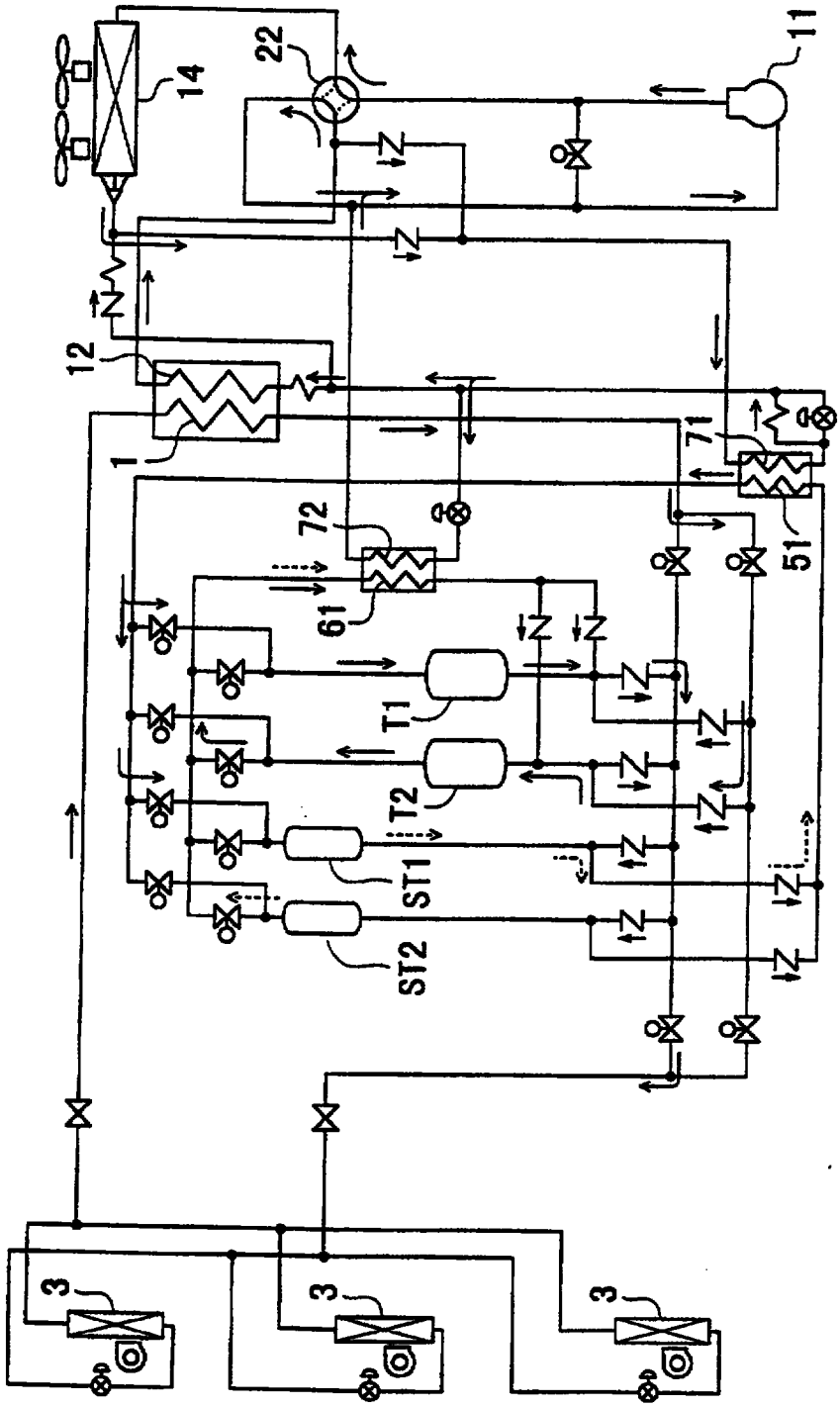
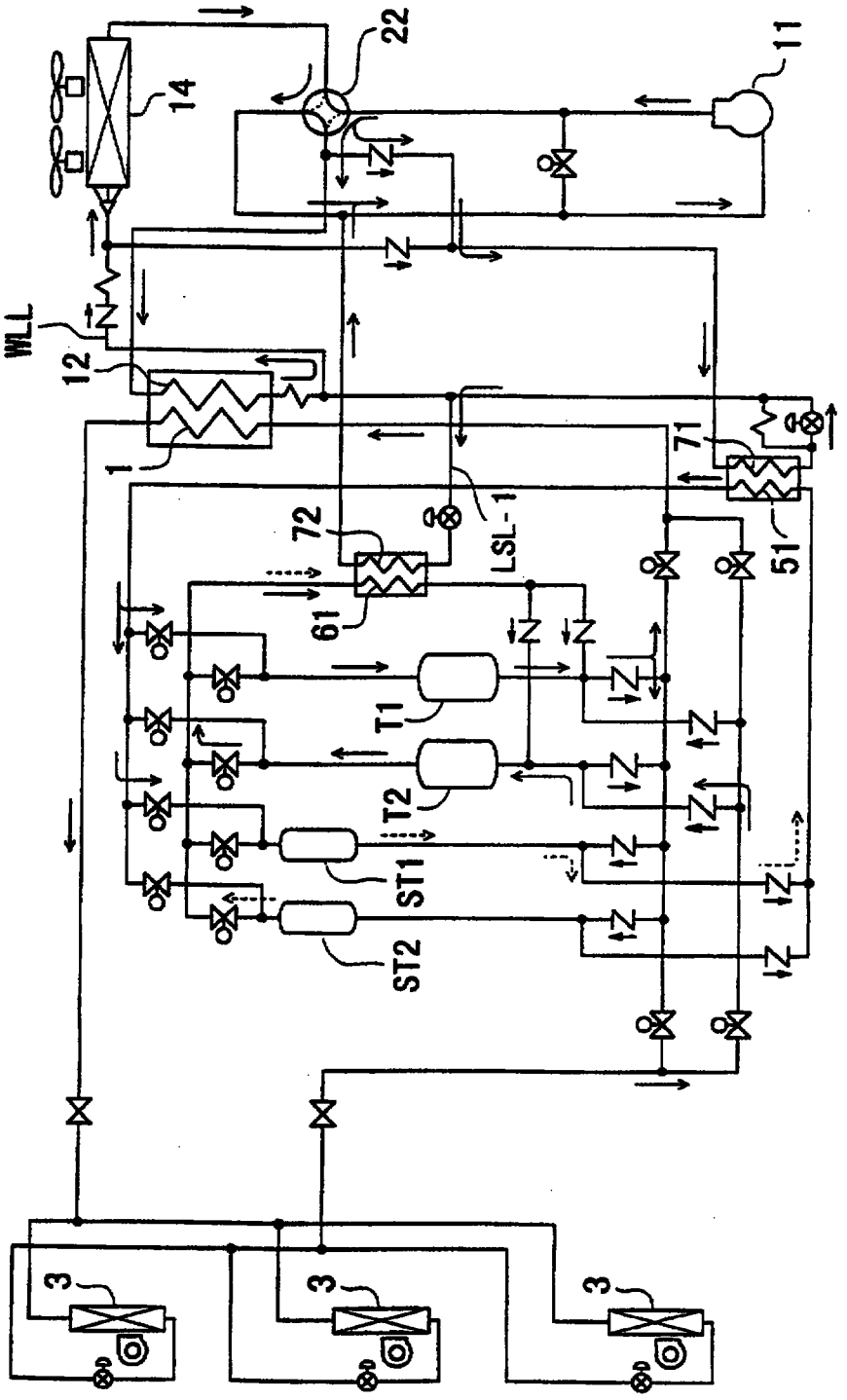


Fig. 38



HEAT TRANSPORT SYSTEM

TECHNICAL FIELD

The present invention relates to a heat transport system, which can be used as refrigerant circuitry for an air conditioning system, for example. More particularly, the present invention relates to a heat transport system for transporting heat by circulating a heat transport medium without requiring a driving source such as a pump.

BACKGROUND ART

As refrigerant circuitry for an air conditioning system, two-system refrigerant circuitry, such as that disclosed in Japanese Laid-Open Publication No. 62-238951, has conventionally been known. Refrigerant circuitry of this type includes: a primary refrigerant circuit in which a compressor, a heat exchanger on a first heat source side, a pressure reducing mechanism and a heat exchanger on a first application side are sequentially connected to each other through a refrigerant pipe; and a secondary refrigerant circuit in which a pump, a heat exchanger on a second heat source side and a heat exchanger on a second application side are connected to each other through a refrigerant pipe.

And, heat is exchanged between the heat exchanger on the first application side of the primary refrigerant circuit and

the heat exchanger on the second heat source side of the secondary refrigerant circuit, and the heat exchanger on the second application side is disposed within a room to be air-conditioned.

In this refrigerant circuitry, during the room cooling running, a refrigerant is evaporated in the heat exchanger on the first application side and is condensed in the heat exchanger on the second heat source side. In the heat exchanger on the second application side, the condensed refrigerant exchanges heat with the indoor air and is evaporated, thereby cooling the indoor air.

On the other hand, during the room heating running, a refrigerant is condensed in the heat exchanger on the first application side and is evaporated in the heat exchanger on the second heat source side. In the heat exchanger on the second application side, the evaporated refrigerant exchanges heat with the indoor air and is condensed, thereby heating the indoor air.

In this way, the piping length of the primary refrigerant circuit is intentionally shortened, thereby trying to improve the refrigerating capacity.

However, in such an arrangement, a pump is required as a discrete driving source for circulating the refrigerant in the secondary refrigerant circuit. As a result, the power consumption and the like are increased. In addition, since

such a driving source is required, the number of parts having such factors as to cause some failure is increased and thus the reliability of the entire system is adversely deteriorated.

As refrigerant circuitry for overcoming these problems, there exists a heat transport system of a so-called "non-powered" heat transport type, in which no driving source is provided for a secondary refrigerant circuit. The heat transport systems of such a type include a system disclosed in Japanese Laid-Open Publication No. 63-180022. In the heat transport system, the secondary refrigerant circuit is constructed such that a heater, a condenser and a sealed container are sequentially connected to each other through a refrigerant pipe and that the sealed container is disposed at a position higher than that of the heater. Moreover, the heater and the sealed container are connected to each other through an equalizer pipe including an opening/closing valve.

In such an arrangement, during the room heating running, the opening/closing valve is first closed. The gaseous refrigerant heated by the heater is condensed in the condenser so as to be liquefied. Then, the liquid refrigerant is recovered into the sealed container. Thereafter, the opening/closing valve is opened, the pressure in the heater is equalized by the equalizer pipe with the pressure in the sealed container, and then the liquid refrigerant is recovered

from the sealed container, disposed at a position higher than that of the heater, to the heater.

By repeating this operation, the circulation of the refrigerant is enabled without providing any driving source such as a pump for the secondary refrigerant circuit.

(Problems to be solved)

However, in such a heat transport system, if the gaseous refrigerant flows from the condenser into the sealed container, then the pressure in the sealed container rises. As a result, there is some possibility that the operation of circulating the refrigerant cannot be performed satisfactorily. Thus, the condenser is required to excessively cool the refrigerant so that the gaseous refrigerant does not flow out from the condenser.

Moreover, the heat transport system ameliorates the inner structure of the sealed container so as to suppress a rise in pressure within the sealed container. However, the system cannot be regarded as attaining sufficient reliability.

Furthermore, if the liquid refrigerant is to be introduced into the sealed container with certainty in this manner, then the condenser is required to be disposed at a position higher than that of the sealed container. Thus, since undue restriction is imposed on the positions where the respective units are disposed, it has been difficult to apply such a system to a large-scale system or a system having a long pipe.

In view of this point, the present invention has been made in order to accomplish an objective of alleviating the restriction on the positions where units are disposed and attaining high reliability and universality for a heat transport system of a non-powered heat transport type requiring no driving source.

DISCLOSURE OF THE INVENTION

In order to accomplish the above-described objective, according to the present invention, a refrigerant in a refrigerant circuit on an application side is pressurized, and is circulated in the refrigerant circuit on the application side by utilizing this pressure. In addition, the direction in which the refrigerant circulates is controlled such that heat exchange means on the application side can perform a predetermined operation.

Specifically, as shown in Figure 1, the first solution provided by the present invention includes a refrigerant circuit (**B**) constituted such that heat exchange means (**1**) on a heat source side is connected to heat exchange means (**3**) on an application side through a gas pipe (**6**) and a liquid pipe (**7**) so as to circulate a refrigerant therein, the heat exchange means (**1**) on the heat source side exchanging heat with heat source means (**A**).

And tank means (**T**) for communicating with the liquid

pipe (7) and reserving a liquid refrigerant therein is also provided. And pressure regulating means (18) for alternately performing a pressurizing operation for raising an internal pressure of the tank means (T) and a pressure reducing operation for lowering the internal pressure is further provided.

In addition, refrigerant control means (H) is further provided for allowing only a supply of the liquid refrigerant from the tank means (T) to any of the heat exchange means to be an evaporator during the pressurizing operation of the pressure regulating means (18) and allowing only a recovery of the liquid refrigerant from any of the heat exchange means to be a condenser to the tank means (T) during the pressure reducing operation thereof, thereby circulating the refrigerant of the refrigerant circuit (B) and making the heat exchange means (3) on the application side absorb or radiate heat.

In the first solution, the liquid refrigerant is supplied from the tank means (T) to the heat exchange means to be an evaporator during the pressurizing operation of the pressure regulating means (18). On the other hand, during the pressure reducing operation of the pressure regulating means (18), the liquid refrigerant is recovered from the heat exchange means to be a condenser to the tank means (T). Thus, the refrigerant is circulated in a predetermined direction between the heat exchange means (1) on the heat source

side and the heat exchange means (3) on the application side and the heat absorption or radiation is caused in the heat exchange means (3) on the application side.

In this manner, the refrigerant is circulated by utilizing the pressure applied from the pressure regulating means (18) to the tank means (T). Also, the recovery of the liquid refrigerant from the heat exchange means to be a condenser to the tank means (T) is performed by a low pressure generated in the tank means (T).

Thus, in accordance with the first solution, since a high-pressure state and a low-pressure state are alternately and switchably established in the tank means (T) connected to the liquid pipe (7) and the refrigerant is circulated between the heat exchange means (1) on the heat source side and the heat exchange means (3) on the application side by utilizing this pressure, special transport means, such as a refrigerant circulating pump for circulating the refrigerant is no longer necessary. As a result, the power consumption and the number of parts having such factors as to cause a failure can be reduced and the reliability of the entire system can be ensured.

In addition, since the liquid refrigerant in the heat exchange means to be a condenser is sucked by utilizing the low pressure of the tank means (T), the conventional restriction on the positions where units are disposed, such as dis-

posing the tank means (T) at a lower position than that of the heat exchange means, can be eliminated, thereby improving the practicality.

Moreover, since the refrigerant circulation operation in the refrigerant circuit (B) can be performed stably, the refrigerant can be circulated satisfactorily even when the entire circuitry is formed in a large size. Consequently, the system can be enlarged.

In the second solution provided by the present invention, the first solution is modified such that the heat exchange means (3) on the application side functions as an evaporator for absorbing heat as shown in Figure 1. The refrigerant control means (H) allows the liquid refrigerant to be supplied from the tank means (T) to the heat exchange means (3) on the application side and prevents the liquid refrigerant from being supplied from the tank means (T) to the heat exchange means (1) on the heat source side during the pressurizing operation of the pressure regulating means (18), and allows the liquid refrigerant to be recovered from the heat exchange means (1) on the heat source side to the tank means (T) and prevents the liquid refrigerant from being recovered from the heat exchange means (3) on the application side to the tank means (T) during the pressure reducing operation of the pressure regulating means (18).

In the second solution, during the heat absorption run-

ning of the heat exchange means (3) on the application side, the liquid refrigerant is supplied from the tank means (T) to the heat exchange means (3) on the application side, and is evaporated in the heat exchange means (3) on the application side. And the gaseous refrigerant is condensed in the heat exchange means (1) on the heat source side and then recovered into the tank means (T). Thus, a heat absorption operation is realized by the refrigerant evaporated in the heat exchange means (3) on the application side.

In accordance with the second solution, only the supply of the liquid refrigerant from the tank means (T) to the heat exchange means (3) on the application side is allowed during the pressurizing operation of the pressure regulating means (18), and only the recovery of the liquid refrigerant from the heat exchange means (1) on the heat source side to the tank means (T) is allowed during the pressure reducing operation of the pressure regulating means (18), thereby making the heat exchange means (3) on the application side perform heat absorption running. Thus, the heat absorption running of the heat exchange means (3) on the application side can be performed with certainty and the reliability of the system can be improved.

In the third solution provided by the present invention, the first solution is modified such that the heat exchange means (3) on the application side functions as a condenser

for radiating heat, as shown in Figure 7. The refrigerant control means (H) allows the liquid refrigerant to be supplied from the tank means (T) to the heat exchange means (1) on the heat source side and prevents the liquid refrigerant from being supplied from the tank means (T) to the heat exchange means (3) on the application side during the pressurizing operation of the pressure regulating means (18), and allows the liquid refrigerant to be recovered from the heat exchange means (3) on the application side to the tank means (T) and prevents the liquid refrigerant from being recovered from the heat exchange means (1) on the heat source side to the tank means (T) during the pressure reducing operation of the pressure regulating means (18).

In the third solution, during the heat radiation running of the heat exchange means (3) on the application side, the liquid refrigerant is supplied from the tank means (T) to the heat exchange means (1) on the heat source side, and is evaporated in the heat exchange means (1) on the heat source side. And the gaseous refrigerant is condensed in the heat exchange means (3) on the application side and then recovered into the tank means (T). Thus, a heat radiation operation is realized by the refrigerant condensed in the heat exchange means (3) on the application side.

In accordance with the third solution, only the supply of the liquid refrigerant from the tank means (T) to the heat

exchange means (1) on the heat source side is allowed during the pressurizing operation of the pressure regulating means (18), and only the recovery of the liquid refrigerant from the heat exchange means (3) on the application side to the tank means (T) is allowed during the pressure reducing operation of the pressure regulating means (18), thereby making the heat exchange means (3) on the application side perform heat radiation running. Thus, the heat radiation running of the heat exchange means (3) on the application side can be performed with certainty and the reliability of the system can be improved.

In the fourth solution provided by the present invention, the first, second or third solution is modified such that the pressure regulating means (18) applies heat to the refrigerant in the tank means (T) so as to raise the internal pressure of the tank means (T) during the pressurizing operation, and extracts heat from the refrigerant of the tank means (T) so as to lower the internal pressure of the tank means (T) during the pressure reducing operation as shown in Figures 4 and 5.

In the fourth solution, the refrigerant in the tank means (T) is directly heated and cooled, thereby changing the internal pressure of the tank means (T) and transporting the refrigerant.

In accordance with the fourth solution, the refrigerant

in the tank means (T) is directly heated and cooled by the pressure regulating means (18), thereby changing the internal pressure of the tank means (T) in this manner. Thus, the internal pressure of the tank means (T) can be changed with a mechanism having a relatively small heat loss and the efficiency of refrigerant transport can be improved.

In the fifth solution provided by the present invention, the fourth solution is modified such that the pressure regulating means (18) is constituted by heat exchange means (18b), which is adjacent to the tank means (T) and switchably performs a heating operation of applying heat to the refrigerant of the tank means (T) and a cooling operation of extracting heat from the refrigerant of the tank means (T), as shown in Figure 4.

In the sixth solution provided by the present invention, the fourth solution is modified such that the pressure regulating means (18) includes a refrigerant circuit (D) including: a compressor (D1); a first heat exchanger (D3); a pressure reducing mechanism (D4); a second heat exchanger (D5); and selector means (D2) for alternately switching connection states of the first heat exchanger (D3) and the second heat exchanger (D5) to an outlet side of the compressor (D1), as shown in Figure 5. And, the first heat exchanger (D3) exchanges heat with the tank means (T) and heats and cools the refrigerant in the tank means (T) in accordance with a

switching operation of the selector means (D2).

In accordance with the fifth and the sixth solutions, specific arrangements can be obtained for the pressure regulating means.

Thus, in accordance with the fifth and the sixth solutions, since the arrangements of the pressure regulating means are specified, the practicality of the system itself can be improved. In addition, since the internal pressure of the tank means (T) can be regulated accurately, the reliability of the running operation can also be improved.

In the seventh solution provided by the present invention, the first, second or third solution is modified such that the pressure regulating means (18) includes pressure generating means (18a, 18c, 19c) coupled to the tank means (T) through a pressure pipe (19), applies a high pressure from the pressure generating means (18a, 18c, 19c) to the inside of the tank means (T) during a pressurizing operation, and applies a low pressure from the pressure generating means (18a, 18c, 19c) to the inside of the tank means (T) during a pressure reducing operation as shown in Figures 2, 3 and 6.

In accordance with the seventh solution, the pressure generating means (18a, 18c, 19c), which is a source of generating pressure to be applied to the tank means (T), is coupled to the tank means (T) through a pressure pipe (19). Thus, the pressure generation source need not be disposed in

proximity of the tank means (T), and the flexibility of the installation positions can be improved.

In the eighth solution provided by the present invention, the seventh solution is modified such that the pressure generating means functions as a reservoir container (18a) in which a liquid refrigerant is reservable, as shown in Figure 2. The pressure regulating means (18) applies heat to the liquid refrigerant in the reservoir container (18a) so as to evaporate the liquid refrigerant and to raise an internal pressure of the reservoir container (18a) during a pressurizing operation, and extracts heat from a gaseous refrigerant in the reservoir container (18a) so as to condense the gaseous refrigerant and to lower the internal pressure of the reservoir container (18a) during a pressure reducing operation.

On the other hand, in the ninth solution provided by the present invention, the seventh solution is modified such that the pressure generating means functions as a compressor (18c) as shown in Figure 3. A connection state of the pressure pipe (19) to the compressor (18c) is switched to an outlet side and an inlet side of the compressor (18c) by selector means (I). And the pressure regulating means (18) connects the pressure pipe (19) to the outlet side of the compressor (18c) during a pressurizing operation and connects the pressure pipe (19) to the inlet side of the compressor (18c) dur-

ing a pressure reducing operation in accordance with a switching operation of the selector means (I).

Moreover, in the tenth solution provided by the present invention, the seventh solution is modified such that the pressure generating means functions as a heat exchanger (19c) in which a refrigerant is reservable as shown in Figure 6. The pressure regulating means (18) applies heat to the refrigerant in the heat exchanger (19c) so as to raise an internal pressure of the heat exchanger (19c) during a pressurizing operation and extracts heat from the refrigerant in the heat exchanger (19c) so as to lower the internal pressure of the heat exchanger (19c) during a pressure reducing operation.

Furthermore, in the eleventh solution provided by the present invention, the tenth solution is modified such that the pressure regulating means (18) includes a refrigerant circuit (D) including: a compressor (D1); a first heat exchanger (D3); a pressure reducing mechanism (D4); a second heat exchanger (D5); and selector means (D2) for alternately switching connection states of the first heat exchanger (D3) and the second heat exchanger (D5) to an outlet side of the compressor (D1) as shown in Figure 6. And, the first heat exchanger (D3) exchanges heat with the heat exchanger (19c) and heats and cools the refrigerant of the heat exchanger (19c) in accordance with a switching operation of the selec-

tor means (D2).

In the eighth to the eleventh solutions, specific arrangements can be obtained for the pressure generating means (18a, 18c, 19c) to be a pressure generation source for performing a pressurizing operation and a pressure reducing operation with respect to the tank means (T).

In accordance with the eighth to the eleventh solutions, the arrangements of the pressure generating means (18a, 18c, 19c) can be specified. Thus, the practicality of the system itself can be improved.

In the twelfth solution provided by the present invention, the second solution is modified such that the refrigerant control means (H) is constituted by: a first solenoid valve (SV-A), which is provided between a connecting point of the tank means (T) to the liquid pipe (7) and the heat exchange means (1) on the heat source side, closes during the pressurizing operation of the pressure regulating means (18) and opens during the pressure reducing operation thereof; and a second solenoid valve (SV-B), which is provided between the connecting point of the tank means (T) to the liquid pipe (7) and the heat exchange means (3) on the application side, opens during the pressurizing operation of the pressure regulating means (18) and closes during the pressure reducing operation thereof, as shown in Figure 8.

On the other hand, in the thirteenth solution provided

by the present invention, the third solution is modified such that the refrigerant control means (H) is constituted by: a first solenoid valve (SV-A), which is provided between a connecting point of the tank means (T) to the liquid pipe (7) and the heat exchange means (1) on the heat source side, opens during the pressurizing operation of the pressure regulating means (18) and closes during the pressure reducing operation thereof; and a second solenoid valve (SV-B), which is provided between the connecting point of the tank means (T) to the liquid pipe (7) and the heat exchange means (3) on the application side, closes during the pressurizing operation of the pressure regulating means (18) and opens during the pressure reducing operation thereof, as shown in Figure 8.

Moreover, in the fourteenth solution provided by the present invention, the second solution is modified such that the refrigerant control means (H) is constituted by: a first check valve (CV1), which is provided between a connecting point of the tank means (T) to the liquid pipe (7) and the heat exchange means (1) on the heat source side and allows only the liquid refrigerant to flow from the heat exchange means (1) on the heat source side to the tank means (T); and a second check valve (CV2), which is provided between the connecting point of the tank means (T) to the liquid pipe (7) and the heat exchange means (3) on the application side and allows only the liquid refrigerant to flow from the tank means (T) to

the heat exchange means (3) on the application side, as shown in Figure 1.

Furthermore, in the fifteenth solution provided by the present invention, the third solution is modified such that the refrigerant control means (H) is constituted by: a first check valve (CV3), which is provided between a connecting point of the tank means (T) to the liquid pipe (7) and the heat exchange means (1) on the heat source side and allows only the liquid refrigerant to flow from the tank means (T) to the heat exchange means (1) on the heat source side; and a second check valve (CV4), which is provided between the connecting point of the tank means (T) to the liquid pipe (7) and the heat exchange means (3) on the application side and allows only the liquid refrigerant to flow from the heat exchange means (3) on the application side to the tank means (T), as shown in Figure 7.

In the twelfth to the fifteenth solutions, specific arrangements can be obtained for the refrigerant control means (H).

In accordance with the twelfth to the fifteenth solutions, since specific arrangements can be obtained for the refrigerant control means (H), the refrigerant circulation direction can be accurately set in order to make the heat exchange means (3) on the application side perform heat absorption or radiation running. This also makes it possible to

improve the reliability of the running operation and the practicality.

In the sixteenth solution provided by the present invention, by providing a plurality of tank means (**T1**, **T2**), the heat radiation or absorption running of the heat exchange means (**3**) on the application side can be performed continuously.

Specifically, as shown in Figures **10**, **12**, etc., first, a refrigerant circuit (**B**), constituted such that heat exchange means (**1**) on a heat source side is connected to heat exchange means (**3**) on an application side through a gas pipe (**6**) and a liquid pipe (**7**) so as to circulate a refrigerant therein and that the heat exchange means (**1**) on the heat source side exchanges heat with heat source means (**A**), is provided.

And, at least one first tank means (**T1**) and at least one second tank means (**T2**), which are connected in parallel to the liquid pipe (**7**) and reserve a liquid refrigerant therein, are also provided.

Moreover, pressure regulating means (**18**) for alternately switching a first pressure state, in which an internal pressure of the first tank means (**T1**) is raised and an internal pressure of the second tank means (**T2**) is lowered, and a second pressure state, in which the internal pressure of the first tank means (**T1**) is lowered and the internal pressure of the second tank means (**T2**) is raised, is further provided.

In addition, refrigerant control means (H) is further provided for supplying the liquid refrigerant from the first tank means (T1) to any of the heat exchange means to be an evaporator and recovering the liquid refrigerant from any of the heat exchange means to be a condenser to the second tank means (T2) during the first pressure state of the pressure regulating means (18), and for supplying the liquid refrigerant from the second tank means (T2) to any of the heat exchange means to be an evaporator and recovering the liquid refrigerant from any of the heat exchange means to be a condenser to the first tank means (T1) during the second pressure state of the pressure regulating means (18), thereby circulating the refrigerant of the refrigerant circuit (B) and making the heat exchange means (3) on the application side continuously absorb or radiate heat.

In the sixteenth solution, by making the refrigerant control means (H) prevent the refrigerant from flowing while alternately switching the first pressure state and the second pressure state of the pressure regulating means (18), tank means supplying the liquid refrigerant to one heat exchange means and tank means recovering the refrigerant from the other heat exchange means are alternately switched. Thus, the heat absorption or radiation running of the heat exchange means (3) on the application side is performed continuously.

In accordance with the sixteenth solution, an operation

of applying a high pressure to the first tank means (T1) and a low pressure to the second tank means (T2) and an operation of applying a high pressure to the second tank means (T2) and a low pressure to the first tank means (T1) are alternately performed. Thus, since the heat absorption or radiation running of the heat exchange means (3) on the application side can be performed continuously, performance and practicality of the entire system can be improved.

In addition, in this solution as in the first solution described above, special transport means for circulating the refrigerant between the heat exchange means (1) on the heat source side and the heat exchange means (3) on the application side is not necessary, either. As a result, the power consumption and the number of parts having such factors as to cause a failure can be reduced and the reliability of the entire system can be ensured.

Moreover, since the restriction on the installation positions of the units can be alleviated, the universality can be improved.

In the seventeenth solution provided by the present invention, the sixteenth solution is modified such that, firstly, the heat exchange means (3) on the application side functions as an evaporator for absorbing heat as shown in Figure 12. And, the refrigerant control means (H) switches refrigerant flow states in the liquid pipe (7) so as to sup-

ply the liquid refrigerant from the first tank means (T1) to the heat exchange means (3) on the application side and recover the liquid refrigerant from the heat exchange means (1) on the heat source side to the second tank means (T2) during the first pressure state of the pressure regulating means (18), and so as to supply the liquid refrigerant from the second tank means (T2) to the heat exchange means (3) on the application side and recover the liquid refrigerant from the heat exchange means (1) on the heat source side to the first tank means (T1) during the second pressure state of the pressure regulating means (18).

In the seventeenth solution, a state where the liquid refrigerant is recovered from the heat exchange means (1) on the heat source side to the second tank means (T2) while the liquid refrigerant is supplied from the first tank means (T1) to the heat exchange means (3) on the application side, and a state where the liquid refrigerant is recovered from the heat exchange means (1) on the heat source side to the first tank means (T1) while the liquid refrigerant is supplied from the second tank means (T2) to the heat exchange means (3) on the application side are alternately established. As a result, the heat absorption running of the heat exchange means (3) on the application side is performed continuously.

On the other hand, in the eighteenth solution provided by the present invention, the sixteenth solution is modified

such that, firstly, the heat exchange means (3) on the application side functions as a condenser for radiating heat as shown in Figure 13. And the refrigerant control means (H) switches refrigerant flow states in the liquid pipe (7) so as to supply the liquid refrigerant from the first tank means (T1) to the heat exchange means (1) on the heat source side and recover the liquid refrigerant from the heat exchange means (3) on the application side to the second tank means (T2) during the first pressure state of the pressure regulating means (18), and so as to supply the liquid refrigerant from the second tank means (T2) to the heat exchange means (1) on the heat source side and recover the liquid refrigerant from the heat exchange means (3) on the application side to the first tank means (T1) during the second pressure state of the pressure regulating means (18).

In the eighteenth solution, a state where the liquid refrigerant is recovered from the heat exchange means (3) on the application side to the second tank means (T2) while the liquid refrigerant is supplied from the first tank means (T1) to the heat exchange means (1) on the heat source side, and a state where the liquid refrigerant is recovered from the heat exchange means (3) on the application side to the first tank means (T1) while the liquid refrigerant is supplied from the second tank means (T2) to the heat exchange means (1) on the heat source side are alternately established. As a result,

the heat radiation running of the heat exchange means (3) on the application side is performed continuously.

Thus, in accordance with the seventeenth solution, by alternately switching the pressure application states onto the respective tank means (T1, T2), the heat absorption running of the heat exchange means (3) on the application side can be performed continuously. On the other hand, in accordance with the eighteenth solution, by alternately switching the pressure application states onto the respective tank means (T1, T2) in a similar manner, the heat radiation running of the heat exchange means (3) on the application side can be performed continuously. Accordingly, when the air is conditioned by cooling or heating the indoor air while disposing the heat exchange means (3) on the application side indoors, the air condition in the room can always be kept satisfactory.

Moreover, in the nineteenth solution provided by the present invention, the sixteenth, seventeenth or eighteenth solution is modified such the pressure regulating means (18) includes pressure generating means (18A, 18B, D1, E1, E2) coupled to the respective tank means (T1, T2) through pressure pipes (19d, 19e), makes the pressure generating means (18A, 18B, D1, E1, E2) apply a high pressure to the inside of the first tank means (T1) and a low pressure to the inside of the second tank means (T2) during the first pressure state

and makes the pressure generating means (**18A**, **18B**, **D1**, **E1**, **E2**) apply a high pressure to the inside of the second tank means (**T2**) and a low pressure to the inside of the first tank means (**T1**) during the second pressure state, as shown in Figures **10** to **12**.

Thus, in accordance with the nineteenth solution, as for a unit for making the heat exchange means (**3**) on the application side perform heat absorption or radiation running continuously, the generation source of the pressure to be applied onto the respective tank means (**T1**, **T2**) need not be disposed in the proximity of the respective tank means (**T1**, **T2**) in the same way as in the seventh solution described above. As a result, the flexibility in installation positions can be improved.

In the twentieth solution provided by the present invention, the nineteenth solution is modified such that the pressure generating means are constituted by a first reservoir container (**18A**) which is connected to the first tank means (**T1**) and in which a liquid refrigerant is reservable, and a second reservoir container (**18B**) which is connected to the second tank means (**T2**) and in which a liquid refrigerant is reservable, as shown in Figure **10**.

And the pressure regulating means (**18**) applies heat to the liquid refrigerant in the first reservoir container (**18A**) so as to evaporate the liquid refrigerant and to raise an in-

ternal pressure of the reservoir container (**18A**), and extracts heat from a gaseous refrigerant in the second reservoir container (**18B**) so as to condense the gaseous refrigerant and to lower the internal pressure of the reservoir container (**18B**) during the first pressure state. And the pressure regulating means (**18**) applies heat to the liquid refrigerant in the second reservoir container (**18B**) so as to evaporate the liquid refrigerant and to raise an internal pressure of the reservoir container (**18B**), and extracts heat from a gaseous refrigerant in the first reservoir container (**18A**) so as to condense the gaseous refrigerant and to lower the internal pressure of the reservoir container (**18A**) during the second pressure state.

On the other hand, in the twenty-first solution provided by the present invention, the nineteenth solution is modified such that, firstly, the pressure generating means is constituted by a compressor (**D1**) as shown in Figure **11**. And switching of connection states of the first tank means (**T1**) and the second tank means (**T2**) to the compressor (**D1**) is performed by making selector means (**I**) switch the pressure pipes (**19d, 19e**) to an outlet side and an inlet side of the compressor (**D1**).

In addition, the pressure regulating means (**18**) connects the outlet side of the compressor (**D1**) to the first tank means (**T1**) and the inlet side of the compressor (**D1**) to the

second tank means (**T2**) during the first pressure state and connects the outlet side of the compressor (**D1**) to the second tank means (**T2**) and the inlet side of the compressor (**D1**) to the first tank means (**T1**) during the second pressure state.

Moreover, in the twenty-second solution provided by the present invention, the nineteenth solution is modified such that, firstly, the pressure generating means are constituted by a first heat exchanger (**E1**) which is connected to the first tank means (**T1**) and in which a refrigerant is reservable, and a second heat exchanger (**E2**) which is connected to the second tank means (**T2**) and in which a refrigerant is reservable, as shown in Figure **12**.

And, the pressure regulating means (**18**) applies heat to the refrigerant in the first heat exchanger (**E1**) so as to raise an internal pressure of the heat exchanger (**E1**) and extracts heat from the refrigerant in the second heat exchanger (**E2**) so as to lower the internal pressure of the heat exchanger (**E2**) during the first pressure state. And the pressure regulating means (**18**) applies heat to the refrigerant in the second heat exchanger (**E2**) so as to raise an internal pressure of the heat exchanger (**E2**) and extracts heat from the refrigerant in the first heat exchanger (**E1**) so as to lower the internal pressure of the heat exchanger (**E1**) during the second pressure state.

Furthermore, in the twenty-third solution provided by

the present invention, the twenty-second solution is modified such that, firstly, the pressure regulating means (18) includes a refrigerant circuit (D) including: a compressor (D1); a first heat exchanger (D3); a pressure reducing mechanism (D4); a second heat exchanger (D5); and selector means (I) for alternately switching connection states of the first heat exchanger (D3) and the second heat exchanger (D5) to an outlet side of the compressor (D1) as shown in Figure 12.

And, the first heat exchanger (D3) exchanges heat with the first heat exchanger (E1) connected to the first tank means (T1), the second heat exchanger (D5) exchanges heat with the second heat exchanger (E2) connected to the second tank means (T2), and the heat exchangers (E1, E2) are switched between the first pressure state and the second pressure state in accordance with a switching operation of the selector means (I).

Furthermore, in the twenty-fourth solution provided by the present invention, the nineteenth solution is modified such that, firstly, the pressure generating means is constituted by a pressurizing heat exchanger (E2) to be heated by a heating heat exchanger (D3) and a pressure reducing heat exchanger (E1) to be cooled by a cooling heat exchanger (D5), as shown in Figure 16.

And, the pressure regulating means (18) connects the pressurizing heat exchanger (E2) to the first tank means (T1)

and the pressure reducing heat exchanger (**E1**) to the second tank means (**T2**) during the first pressure state, and connects the pressurizing heat exchanger (**E2**) to the second tank means (**T2**) and the pressure reducing heat exchanger (**E1**) to the first tank means (**T1**) during the second pressure state.

Furthermore, in the twenty-fifth solution provided by the present invention, the twenty-fourth solution is modified such that the pressure regulating means (**18**) includes a refrigerant circuit (**D**) formed by connecting a compressor (**D1**), a heating heat exchanger (**D3**), a pressure reducing mechanism (**D4**) and a cooling heat exchanger (**D5**) in this order through a refrigerant pipe as shown in Figure **16**.

Thus, in accordance with the twentieth to the twenty-fifth solutions, specific arrangements can be obtained for the pressure generating means that can attain the effects of the nineteenth solution described above. As a result, the practicality can be further improved.

In the twenty-sixth solution provided by the present invention, the first solution is modified such the pressure regulating means (**18**) includes: pressurizing means (**50**) for performing a pressurizing operation of pushing the liquid refrigerant in the tank means (**T**) to the liquid pipe (**7**) by raising the internal pressure of the tank means (**T**); and pressure reducing means (**60**) for performing a pressure reducing operation of recovering the liquid refrigerant from the

liquid pipe (7) to the tank means (T) by lowering the internal pressure of the tank means (T), as shown in Figure 22.

And, the pressure reducing means (60) includes a circulating condenser (61), which is connected to the tank means (T) and which lowers the internal pressure of the tank means (T) by condensing the refrigerant. A condensing pressure of the circulating condenser (61) is set lower than a condensing pressure of the heat exchange means to be the condenser.

In the twenty-sixth solution, owing to the condensation of the refrigerant in the circulating condenser (61), a low pressure is built up inside the tank means (T). And, since this pressure is lower than the condensing pressure of the heat exchange means to be a condenser, the liquid refrigerant in the condenser is sucked into the tank means (T).

In accordance with the twenty-sixth solution, by condensing the refrigerant in the circulating condenser (61) of the pressure reducing means (60) connected to the tank means (T), a low pressure for recovering the liquid refrigerant from the condenser to the tank means (T) can be generated. Thus, in such a case, respective units can be disposed without receiving such restriction as disposing the tank means (T) at a position lower than that of the condenser.

In addition, the internal pressure of the tank means (T) can be changed only by switching the communication/non-communication states between the pressure reducing means (60)

and the tank means (T). Thus, if a solenoid valve or the like is employed for a closed circuit formed by the pressure reducing means (60) and the tank means (T), the refrigerant circulation operation can be performed only by opening/closing the valve. As a result, high reliability is realized and the number of parts having such factors as to cause some failure can be reduced.

In the twenty-seventh solution provided by the present invention, the first solution is modified such that, firstly, the pressure regulating means (18) includes: pressurizing means (50) for performing a pressurizing operation of pushing the liquid refrigerant in the tank means (T) to the liquid pipe (7) by raising the internal pressure of the tank means (T); and pressure reducing means (60) for performing a pressure reducing operation of recovering the liquid refrigerant from the liquid pipe (7) to the tank means (T) by lowering the internal pressure of the tank means (T), as shown in Figure 23.

And, the pressurizing means (50) includes a circulating evaporator (51), which is connected to the tank means (T) and which raises the internal pressure of the tank means (T) by evaporating the refrigerant. An evaporating pressure of the circulating evaporator (51) is set higher than an evaporating pressure of the heat exchange means to be the evaporator.

In the twenty-seventh solution, owing to the evaporation

of the refrigerant in the circulating evaporator (51), a high pressure is built up inside the tank means (T). And, since this pressure is higher than the evaporating pressure of the heat exchange means to be an evaporator, the liquid refrigerant is supplied from the tank means (T) to the evaporator.

In accordance with the twenty-seventh solution, by evaporating the refrigerant in the circulating evaporator (51) of the pressurizing means (50) connected to the tank means (T), a high pressure for supplying the liquid refrigerant from the tank means (T) to the evaporator can be generated. Thus, in such a case, restriction on the installation positions of the tank means (T) and the evaporator can be eliminated.

In addition, according to this invention, the internal pressure of the tank means (T) can also be changed only by switching the communication/non-communication states between the pressurizing means (50) and the tank means (T). As a result, high reliability is realized and the number of parts having such factors as to cause some failure can be reduced.

In the twenty-eighth solution provided by the present invention, the twenty-seventh solution is modified such that, firstly, auxiliary tank means (ST) is provided above the circulating evaporator (51) as shown in Figure 26. And selector means (I) is also provided for recovering the liquid refrigerant in the liquid pipe (7) to the auxiliary tank means (ST)

by making the auxiliary tank means (**ST**) communicate with the pressure reducing means (**60**) and the liquid pipe (**7**) during the pressure reducing operation of the pressure reducing means (**60**) and for dripping and supplying the liquid refrigerant in the auxiliary tank means (**ST**) to the circulating evaporator (**51**) by making the auxiliary tank means (**ST**) communicate with the pressurizing means (**50**) during the pressurizing operation of the pressurizing means (**50**).

In the twenty-eighth solution, if the auxiliary tank means (**ST**) of a relatively small size is disposed above the circulating evaporator (**51**), the liquid refrigerant can be supplied to the circulating evaporator (**51**). Thus, it is possible to eliminate a situation where the circulation of a refrigerant is disabled by the exhaustion of the liquid refrigerant in the circulating evaporator (**51**).

In accordance with the twenty-eighth solution, the auxiliary tank means (**ST**) is provided above the circulating evaporator (**51**) and the liquid refrigerant to be supplied to the circulating evaporator (**51**) is temporarily reserved in the auxiliary tank means (**ST**). Thus, a sufficient amount of liquid refrigerant can be supplied to the circulating evaporator (**51**) without receiving any restriction on the installation positions of the tank means (**T**) and the circulating evaporator (**51**) in the vertical direction.

In the twenty-ninth solution provided by the present in-

vention, the twenty-seventh solution is modified such that, firstly, at least one first auxiliary tank means (**ST1**) and at least one second auxiliary tank means (**ST2**) are provided above the circulating evaporator (**51**) as shown in Figure **27**. And, a selector means (**I**) is also provided for selecting a first selection state in which the liquid refrigerant in the liquid pipe (**7**) is recovered to the first auxiliary tank means (**ST1**) by making the first auxiliary tank means (**ST1**) communicate with the pressure reducing means (**60**) and the liquid pipe (**7**) and in which the liquid refrigerant in the second auxiliary tank means (**ST2**) is dripped and supplied to the circulating evaporator (**51**) by making the second auxiliary tank means (**ST2**) communicate with the pressurizing means (**50**) or a second selection state in which the liquid refrigerant in the liquid pipe (**7**) is recovered to the second auxiliary tank means (**ST2**) by making the second auxiliary tank means (**ST2**) communicate with the pressure reducing means (**60**) and the liquid pipe (**7**) and in which the liquid refrigerant in the first auxiliary tank means (**ST1**) is dripped and supplied to the circulating evaporator (**51**) by making the first auxiliary tank means (**ST1**) communicate with the pressurizing means (**50**).

In the twenty-ninth solution, a state where the liquid refrigerant is recovered into one auxiliary tank means and a state where the liquid refrigerant is supplied from the other

tank means to the circulating evaporator (51) simultaneously established. Thus, the number of times by which the operations of recovering/supplying the liquid refrigerant are repeatedly performed on the respective auxiliary tank means (ST1, ST2) can be reduced.

In accordance with the twenty-ninth solution, a plurality of auxiliary tank means (ST1, ST2) are provided and the liquid refrigerant is recovered into one of the auxiliary tank means and is supplied from the other auxiliary tank means to the circulating evaporator. Thus, it is no longer necessary to switch the operations of the auxiliary tank means (ST1, ST2) in synchronism with the pressurizing/pressure reducing operations with respect to the tank means (T). Accordingly, the number of times by which the operations of recovering/supplying the liquid refrigerant are repeatedly performed on the respective auxiliary tank means (ST1, ST2) can be reduced, and the lifetimes thereof can be lengthened.

In the thirtieth solution provided by the present invention, the sixteenth solution is modified such that, firstly, the pressure regulating means (18) includes: pressurizing means (50) for performing a pressurizing operation of pushing the liquid refrigerant in one of the first tank means (T1) and the second tank means (T2) to the liquid pipe (7) by raising the internal pressure of the one tank means (T1 or

T2); and pressure reducing means (60) for performing a pressure reducing operation of recovering the liquid refrigerant from the liquid pipe (7) to the other tank means (T2 or T1) by lowering the internal pressure of the other tank means (T2 or T1) as shown in Figure 24.

And, the pressure reducing means (60) includes a circulating condenser (61), which is connected to the respective tank means (T1, T2) and which lowers the internal pressure of each said tank means (T1, T2) by condensing the refrigerant. A condensing pressure of the circulating condenser (61) is set lower than a condensing pressure of the heat exchange means to be the condenser.

Furthermore, the pressure regulating means (18) makes the pressurizing means (50) pressurize the first tank means (T1) and makes the pressure reducing means (60) reduce a pressure of the second tank means (T2) during a first pressure state. And the pressure regulating means (18) makes the pressurizing means (50) pressurize the second tank means (T2) and makes the pressure reducing means (60) reduce a pressure of the first tank means (T1) during a second pressure state.

In the thirtieth solution, the respective tank means (T1, T2) are not disposed at positions lower than that of the condenser and the liquid refrigerant is recovered from the condenser to the tank means (T1, T2) in a refrigerant circuit that can continuously perform the heat absorption or radiation

running of the heat exchange means (3) on the application side.

Thus, in accordance the thirtieth solution, the liquid refrigerant can be recovered from the condenser to the tank means (T1, T2) and circulated, without receiving such restriction that the respective tank means (T1, T2) are disposed at positions lower than that of the condenser, in a refrigerant circuit that can continuously perform the heat absorption or radiation running of the heat exchange means (3) on the application side.

In the thirty-first solution provided by the present invention, the sixteenth solution is modified such that, firstly, the pressure regulating means (18) includes: pressurizing means (50) for performing a pressurizing operation of pushing the liquid refrigerant in one of the first tank means (T1) and the second tank means (T2) to the liquid pipe (7) by raising the internal pressure of the one tank means (T1 or T2); and pressure reducing means (60) for performing a pressure reducing operation of recovering the liquid refrigerant from the liquid pipe (7) to the other tank means (T2 or T1) by lowering the internal pressure of the other tank means (T2 or T1), as shown in Figure 25.

And, the pressurizing means (50) includes a circulating evaporator (51), which is connected to the respective tank means (T1, T2) and which raises the internal pressure of each

said tank means (**T1, T2**) by evaporating the refrigerant. An evaporating pressure of the circulating evaporator (**51**) is set higher than an evaporating pressure of the heat exchange means to be the evaporator.

Furthermore, the pressure regulating means (**18**) makes the pressurizing means (**50**) pressurize the first tank means (**T1**) and makes the pressure reducing means (**60**) reduce a pressure of the second tank means (**T2**) during a first pressure state. And the pressure regulating means (**18**) makes the pressurizing means (**50**) pressurize the second tank means (**T2**) and makes the pressure reducing means (**60**) reduce a pressure of the first tank means (**T1**) during a second pressure state.

In the thirty-first solution, the liquid refrigerant is supplied from the tank means (**T1, T2**) to the evaporator without receiving any restriction on the installation positions of the respective tank means (**T1, T2**) with respect to the evaporator in a refrigerant circuit that can continuously perform the heat absorption or radiation running of the heat exchange means (**3**) on the application side.

Thus, in accordance the thirty-first solution, the liquid refrigerant can be supplied from the tank means (**T1, T2**) to the evaporator, without receiving any restriction on the installation positions of the respective tank means (**T1, T2**) with respect to the evaporator, in a refrigerant circuit that can continuously perform the heat absorption or radiation run-

ning of the heat exchange means (3) on the application side.

In the thirty-second solution provided by the present invention, the thirty-first solution is modified such that, firstly, at least one first auxiliary tank means (**ST1**) and at least one second auxiliary tank means (**ST2**) are provided above the circulating evaporator (**51**) as shown in Figures **30**, **33**, **36**, etc.

And selector means (**I**) is also provided for selecting a first selection state in which the liquid refrigerant in the liquid pipe (**7**) is recovered to the first auxiliary tank means (**ST1**) by making the first auxiliary tank means (**ST1**) communicate with the pressure reducing means (**60**) and the liquid pipe (**7**) and in which the liquid refrigerant in the second auxiliary tank means (**ST2**) is dripped and supplied to the circulating evaporator (**51**) by making the second auxiliary tank means (**ST2**) communicate with the pressurizing means (**50**) or a second selection state in which the liquid refrigerant in the liquid pipe (**7**) is recovered to the second auxiliary tank means (**ST2**) by making the second auxiliary tank means (**ST2**) communicate with the pressure reducing means (**60**) and the liquid pipe (**7**) and in which the liquid refrigerant in the first auxiliary tank means (**ST1**) is dripped and supplied to the circulating evaporator (**51**) by making the first auxiliary tank means (**ST1**) communicate with the pressurizing means (**50**).

Thus, in accordance with the thirty-second solution, by providing a plurality of auxiliary tank means (**ST1**, **ST2**), the same effects as those attained by the twenty-ninth solution can be attained. Accordingly, in a refrigerant circuit that can continuously perform the heat absorption or radiation running of the heat exchange means (**3**) on the application side, the number of times by which the operations of recovering/supplying the liquid refrigerant are repeatedly performed on the respective auxiliary tank means (**ST1**, **ST2**) can be reduced, and the lifetimes thereof can be lengthened.

In the thirty-third solution provided by the present invention, the twenty-sixth or thirtieth solution is modified such that, firstly, the heat source means (**A**) includes first heat exchange means (**12**) for exchanging heat with the heat exchange means (**1**) on the heat source side and second heat exchange means (**72**) for exchanging heat with the circulating condenser (**61**) as shown in Figure **22**.

And, during the heat absorption running of the heat exchange means (**3**) on the application side, an evaporating temperature of the first heat exchange means (**12**) and an evaporating temperature of the second heat exchange means (**72**) are equal to each other but a ratio of a capacity of the circulating condenser (**61**) to a flow rate of the refrigerant flowing through the second heat exchange means (**72**) is set larger than a ratio of a capacity of the heat exchange means (**1**) on

the heat source side to a flow rate of the refrigerant flowing through the first heat exchange means (12).

In accordance with the thirty-third solution, a specific arrangement for setting the condensing pressure of the circulating condenser (61) to be lower than the condensing pressure of the heat exchange means (1) on the heat source side to be the condenser can be obtained.

In the thirty-fourth solution provided by the present invention, the twenty-seventh or twenty-first solution is modified such that, firstly, the heat source means (A) includes first heat exchange means (12) for exchanging heat with the heat exchange means (1) on the heat source side and second heat exchange means (71) for exchanging heat with the circulating evaporator (51) as shown in Figure 23.

And, during the heat radiation running of the heat exchange means (3) on the application side, a condensing temperature of the first heat exchange means (12) and a condensing temperature of the second heat exchange means (71) are equal to each other but a ratio of a capacity of the circulating evaporator (51) to a flow rate of the refrigerant flowing through the second heat exchange means (71) is set larger than a ratio of a capacity of the heat exchange means (1) on the heat source side to a flow rate of the refrigerant flowing through the first heat exchange means (12).

In accordance with the thirty-fourth solution, a speci-

fic arrangement for setting the evaporating pressure of the circulating evaporator (**51**) to be higher than the evaporating pressure of the heat exchange means (**1**) on the heat source side to be the evaporator can be obtained.

Thus, in accordance with the thirty-third solution, a specific arrangement for setting the condensing pressure of the circulating condenser (**61**) to be lower than the condensing pressure of the heat exchange means on the heat source side to be the condenser can be obtained. On the other hand, in accordance with the thirty-fourth solution, a specific arrangement for setting the evaporating pressure of the circulating evaporator (**51**) to be higher than the evaporating pressure of the heat exchange means on the heat source side to be the evaporator can be obtained. As a result, a predetermined pressure can be applied onto the tank means (**T**) with certainty and the reliability of the system can be improved.

In the thirty-fifth solution provided by the present invention, the twenty-sixth or the thirtieth solution is modified such that, firstly, the pressure reducing means (**60**) includes: a gas recovering pipe (**62**) for connecting an upper end of the tank means (**T**) to a gas side of the circulating condenser (**61**); and a liquid supplying pipe (**63**) for connecting a lower end of the tank means (**T**) to a liquid side of the circulating condenser (**1**) as shown in Figure **23**. And the

liquid supplying pipe (63) is connected to the lower end of the tank means (T) independent of the liquid pipe (7).

In accordance with the thirty-fifth solution, the liquid supplying pipe (63) of the pressure reducing means (60) is connected to the lower end of the tank means (T) independent of the liquid pipe (7) and heat exchangers to be the circulating condenser (61) and the condenser are respectively connected to the tank means (T). Thus, the pipes (63, 7) can be provided with check valves having diameters respectively corresponding to the pipe diameters thereof. In particular, a check valve allowing for the reduction in pressure loss is applicable to the liquid supplying pipe (63). As a result, the refrigerant can be circulated smoothly in the pressure reducing means (60).

In the thirty-sixth solution provided by the present invention, the first or the sixteenth solution is modified such that, firstly, the pressure regulating means (18) includes: pressurizing means (50) for performing a pressurizing operation of pushing the liquid refrigerant in the tank means (T) to the liquid pipe (7) by raising the internal pressure of the tank means (T); and pressure reducing means (60) for performing a pressure reducing operation of recovering the liquid refrigerant from the liquid pipe (7) to the tank means (T) by lowering the internal pressure of the tank means (T) as shown in Figures 30 and 33.

And, the pressure reducing means (60) includes a circulating condenser (61), which is connected to the tank means (T) and which lowers the internal pressure of the tank means (T) by condensing the refrigerant. The pressurizing means (50) includes a circulating evaporator (51), which is connected to the tank means (T) and which raises the internal pressure of the tank means (T) by evaporating the refrigerant.

Furthermore, the heat source means (A) includes: first heat exchange means (12) for exchanging heat with the compressor (11) and the heat exchange means (1) on the heat source side; second heat exchange means (72) for exchanging heat with the circulating condenser (61); and third heat exchange means (71) for exchanging heat with the circulating evaporator (51). During the heat radiation running of the heat exchange means (3) on the application side, the heat source means (A) makes the third heat exchange means (71) exchange heat of the gaseous refrigerant discharged from the compressor (11) with the circulating evaporator (51) so as to change sensible heat of the refrigerant, makes the first heat exchange means (12) exchange the heat with the heat exchange means (1) on the heat source side so as to condense the refrigerant, and then makes the second heat exchange means (72) exchange the heat with the circulating condenser (61) so as to evaporate the refrigerant.

On the other hand, in the thirty-seventh solution provided by the present invention, the first or the sixteenth solution is modified such that, firstly, the pressure regulating means (18) includes: pressurizing means (50) for performing a pressurizing operation of pushing the liquid refrigerant in the tank means (T) to the liquid pipe (7) by raising the internal pressure of the tank means (T); and pressure reducing means (60) for performing a pressure reducing operation of recovering the liquid refrigerant from the liquid pipe (7) to the tank means (T) by lowering the internal pressure of the tank means (T) as shown in Figure 36.

And, the pressure reducing means (60) includes a circulating condenser (61), which is connected to the tank means (T) and which lowers the internal pressure of the tank means (T) by condensing the refrigerant. The pressurizing means (50) includes a circulating evaporator (51), which is connected to the tank means (T) and which raises the internal pressure of the tank means (T) by evaporating the refrigerant.

Furthermore, the heat source means (A) includes: first heat exchange means (12) for exchanging heat with the compressor (11) and the heat exchange means (1) on the heat source side; second heat exchange means (72) for exchanging heat with the circulating condenser (61); and third heat exchange means (71) for exchanging heat with the circulating

evaporator (51). During the heat radiation running of the heat exchange means (3) on the application side, the heat source means (A) distributes the gaseous refrigerant discharged from the compressor (11) to the third heat exchange means (71) and the first heat exchange means (12), makes the third heat exchange means (71) exchange heat with the circulating evaporator (51) so as to condense the refrigerant, makes the first heat exchange means (12) exchange heat with the heat exchange means (1) on the heat source side so as to condense the refrigerant, and then makes the second heat exchange means (72) exchange heat of the condensed refrigerant with the circulating condenser (61) so as to evaporate the refrigerant.

Thus, in accordance with the thirty-sixth and the thirty-seventh solutions, a refrigerant circuit applicable as heat source means (A) to a circuit for circulating a refrigerant by making the pressurizing means (50) and the pressure reducing means (60) pressurize/reduce the pressure of the tank means (T) can be obtained and the overall arrangement of the system can be specified.

BRIEF DESCRIPTION OF THE DRAWINGS

Figure 1 is a diagram showing an overall arrangement of refrigerant circuitry in the first embodiment.

Figure 2 is a diagram showing a pressurizing/pressure

reducing mechanism.

Figure 3 is a diagram showing a first variant of the pressurizing/pressure reducing mechanism.

Figure 4 is a diagram showing a second variant of the pressurizing/pressure reducing mechanism.

Figure 5 is a diagram showing a third variant of the pressurizing/pressure reducing mechanism.

Figure 6 is a diagram showing a fourth variant of the pressurizing/pressure reducing mechanism.

Figure 7 is a diagram, corresponding to Figure 1, in the second embodiment.

Figure 8 is a diagram showing a variant of refrigerant flow control means.

Figure 9 is a diagram, corresponding to Figure 1, in the third embodiment.

Figure 10 is a diagram, corresponding to Figure 2, in the fourth embodiment.

Figure 11 is a diagram corresponding to Figure 3 and showing a variant of the fourth embodiment.

Figure 12 is a diagram showing a secondary refrigerant circuit in the fifth embodiment.

Figure 13 is a diagram showing a secondary refrigerant circuit in the sixth embodiment.

Figure 14 is a diagram showing a secondary refrigerant circuit in the seventh embodiment.

Figure 15 is a diagram, corresponding to Figure 1, in the eighth embodiment.

Figure 16 is a diagram, corresponding to Figure 1, in the ninth embodiment.

Figure 17 is a diagram showing a cooling running operation in the ninth embodiment.

Figure 18 is a diagram showing a heating running operation in the ninth embodiment.

Figure 19 is a diagram, corresponding to Figure 1, in the tenth embodiment.

Figure 20 is a diagram showing a cooling running operation in the tenth embodiment.

Figure 21 is a diagram showing a heating running state in the tenth embodiment.

Figure 22 is a diagram, corresponding to Figure 1, in the eleventh embodiment.

Figure 23 is a diagram, corresponding to Figure 1, in the twelfth embodiment.

Figure 24 is a diagram showing a secondary refrigerant circuit in the thirteenth embodiment.

Figure 25 is a diagram showing a secondary refrigerant circuit in the fourteenth embodiment.

Figure 26 is a diagram showing a secondary refrigerant circuit in the fifteenth embodiment.

Figure 27 is a diagram showing a secondary refrigerant

circuit in the sixteenth embodiment.

Figure 28 is a diagram, corresponding to Figure 1, in the seventeenth embodiment.

Figure 29 is a diagram showing a secondary refrigerant circuit in the eighteenth embodiment.

Figure 30 is a diagram, corresponding to Figure 1, in the nineteenth embodiment.

Figure 31 is a diagram showing a cooling running operation in the nineteenth embodiment.

Figure 32 is a diagram showing a heating running state in the nineteenth embodiment.

Figure 33 is a diagram, corresponding to Figure 1, in the twentieth embodiment.

Figure 34 is a diagram showing a cooling running operation in the twentieth embodiment.

Figure 35 is a diagram showing a heating running state in the twentieth embodiment.

Figure 36 is a diagram, corresponding to Figure 1, in the twenty-first embodiment.

Figure 37 is a diagram showing a cooling running operation in the twenty-first embodiment.

Figure 38 is a diagram showing a heating running state in the twenty-first embodiment.

BEST MODE FOR CARRYING OUT THE INVENTION

Hereinafter, embodiments of the present invention will be described with reference to the drawings. In each of the embodiments, two-system refrigerant circuitry including a primary refrigerant circuit and a secondary refrigerant circuit is provided. And, in each of the embodiments, the present invention is applied to refrigerant circuitry for an air conditioning system for conditioning the indoor air by exchanging heat between the primary refrigerant circuit and the secondary refrigerant circuit.

(First Embodiment)

First, the first embodiment will be described with reference to Figure 1.

In this embodiment, an air-conditioning system exclusively used for cooling is constructed and a primary refrigerant circuit (A) constitutes heat source means (A). And, Figure 1 shows refrigerant circuitry as an entire heat transport system according to this embodiment.

First, a secondary refrigerant circuit (B) for cooling the indoor air by exchanging heat with the indoor air will be described.

The secondary refrigerant circuit (B) is constructed such that indoor heat exchangers (3, 3) disposed in a room to be air-conditioned as heat exchange means on the application

side and a heat exchanger (1) on the secondary heat source side functioning as heat exchange means on the heat source side for applying/extracting heat with the primary refrigerant circuit (A) are connected through a gas pipe (6) and a liquid pipe (7), and is formed as a closed circuit in which a refrigerant can circulate. The indoor heat exchangers (3, 3) are connected in parallel to each other and indoor electrically motorized expansion valves (EV1, EV1) are provided for the liquid pipe (7) so as to correspond to the respective indoor heat exchangers (3, 3).

This embodiment is characterized in that a tank (T), in which a liquid refrigerant is reserved, is connected to the liquid pipe (7). A lower end of the tank (T) is connected to the liquid pipe (7) through a connecting pipe (17).

A first check valve (CV1) allowing only the liquid refrigerant to flow from the heat exchanger (1) on the secondary heat source side to the tank (T) is provided between the connecting point of the connecting pipe (17) to the liquid pipe (7) and the heat exchanger (1) on the secondary heat source side. In addition, a second check valve (CV2) allowing only the liquid refrigerant to flow from the tank (T) to the indoor heat exchangers (3, 3) is provided between the connecting point of the connecting pipe (17) to the liquid pipe (7) and the indoor heat exchangers (3, 3). Refrigerant control means (H) is constituted by these check valves (CV1, CV2).

A pressurizing/pressure-reducing mechanism (**18**) functioning as pressure regulating means is connected to an upper end of the tank (**T**) through a pressurizing/pressure-reducing pipe (**19**), which is a pressure pipe. The pressurizing/pressure-reducing mechanism (**18**) is constituted, for example, by a reservoir container (**18a**) functioning as pressure generating means for reserving the liquid refrigerant therein and a heat exchange unit (**18b**) functioning as heat exchange means for heating or cooling the reservoir container (**18a**) as shown in Figure **2**. More specifically, if the reservoir container (**18a**) is heated by the heat exchange unit (**18b**), then the inner pressure is raised by the refrigerant evaporating in the reservoir container (**18a**). On the other hand, if the reservoir container (**18a**) is cooled by the heat exchange unit (**18b**), then the inner pressure is lowered by the refrigerant condensing in the reservoir container (**18a**).

Next, the primary refrigerant circuit (**A**) for exchanging heat with the secondary refrigerant circuit (**B**) will be described.

The primary refrigerant circuit (**A**) is constituted by connecting a compressor (**11**), an outdoor heat exchanger (**14**), an outdoor electrically motorized expansion valve (**EV2**) and a heat exchanger (**12**) on the primary heat source side in this order through a refrigerant pipe (**16**). The outlet side of compressor (**11**) is connected to the outdoor heat exchanger

(14) and the inlet side thereof is connected to the heat exchanger (12) on the primary heat source side.

Moreover, the opening/closing states of the respective electrically motorized expansion valves (EV1, EV2) are controlled by a controller (C). In Figure 1, (F) denotes an indoor fan.

Next, the room cooling running of the refrigerant circuits (A, B) having the above-described arrangements will be described.

When the cooling running is started, the compressor (11) is driven in the primary refrigerant circuit (A). As indicated by the solid-line arrows in Figure 1, a high-temperature, high-pressure gaseous refrigerant discharged from the compressor (11) exchanges heat with the outdoor air and is condensed in the outdoor heat exchanger (14). Then, the pressure of the refrigerant is reduced in the outdoor electrically motorized expansion valve (EV2). In the heat exchanger (12) on the primary heat source side, the refrigerant exchanges heat with the heat exchanger (1) on the secondary heat source side, extracts heat from the refrigerant in the heat exchanger (1) on the secondary heat source side and is evaporated to be sucked into the compressor (11). This circulation operation is repeated.

On the other hand, in the secondary refrigerant circuit (B), the reservoir container (18a) of the pressuriz-

ing/pressure-reducing mechanism (18) is heated by the heat exchange unit (18b). As a result, the refrigerant is evaporated in the reservoir container (18a) and the internal pressure thereof rises. As indicated by the broken-line arrows in Figure 1, the pressure is applied into the tank (T) through the pressurizing/pressure-reducing pipe (19) and pushes out the liquid refrigerant in the tank (T) to the liquid pipe (7) through the connecting pipe (17) while pushing down the water level of the liquid refrigerant. The pushed-out liquid refrigerant flows through the liquid pipe (7) toward the indoor heat exchangers (3, 3). The pressure of the liquid refrigerant is reduced in the indoor electrically motorized expansion valves (EV1, EV1). Then, in the respective indoor heat exchangers (3, 3), the liquid refrigerant exchanges heat with the indoor air and is evaporated, thereby cooling the indoor air. The gaseous refrigerant flows through the gas pipe (6) to the heat exchanger (1) on the secondary heat source side, and then exchanges heat with the heat exchanger (12) on the primary heat source side so as to be condensed.

If the reservoir container (18a) of the pressurizing/pressure-reducing mechanism (18) is cooled by the heat exchange unit (18b) after such an operation has been performed, the refrigerant is condensed in the reservoir container (18a) and the internal pressure thereof falls. As indicated by the one-dot-chain arrows in Figure 1, the pressure

is applied to the inside of the tank (T) through the pressurizing/pressure-reducing pipe (19), and the internal pressure of the tank (T) falls. As a result, the liquid refrigerant condensed in the heat exchanger (1) on the secondary heat source side is recovered into the tank (T) through the liquid pipe (7).

The pressurizing/pressure-reducing mechanism (18) repeatedly pressurizes/reduces the pressure of the tank (T) in the above-described manner. As a result, the liquid refrigerant is pushed out from the tank (T) during pressurizing, while the liquid refrigerant is recovered into the tank (T) during pressure reducing. Consequently, the refrigerant is circulated in the secondary refrigerant circuit (B), thereby cooling the indoor air.

Accordingly, the heat transport system of this embodiment can transport heat in the secondary refrigerant circuit (B) without providing any mechanical driving source such as a pump for the secondary refrigerant circuit (B). As a result, the power consumption can be reduced, the number of parts having such factors as to cause some failure can also be reduced, and reliability can be ensured for the entire system.

In addition, since the liquid refrigerant in the heat exchanger (1) on the secondary heat source side is recovered by utilizing the suction force generated in the tank (T), it is no longer necessary to dispose the tank (T) at a position

lower than that of the heat exchanger (1) on the secondary heat source side. As a result, the restrictions on the positions where units are disposed can be alleviated, and the universality thereof can be improved.

Moreover, the refrigerant can be circulated stably in the secondary refrigerant circuit (**B**) by utilizing the pressure of the refrigerant. Thus, even when the secondary refrigerant circuit (**B**) is formed in a large size, the refrigerant can be circulated satisfactorily. As a result, the system can be enlarged and high reliability can be attained.

(Variants of Pressurizing/pressure-reducing Mechanism)

Next, variants of the pressurizing/pressure-reducing mechanism (**18**) applicable to the above-described secondary refrigerant circuit (**B**) will be described.

Figure **3** shows a first variant. The pressurizing/pressure-reducing mechanism (**18**) includes a pressurizing/pressure-reducing compressor (**18c**). More specifically, the pressurizing/pressure-reducing pipe (**19**) is branched into two (first and second) branch pipes (**19a, 19b**). The first branch pipe (**19a**) is connected to the outlet side of the compressor (**18c**) and the second branch pipe (**19b**) is connected to the inlet side of the compressor (**18c**). A first and a second solenoid valve (**SV1, SV2**) are provided for the branch pipes (**19a, 19b**), respectively.

Selector means (I) is constituted by these branch pipes (19a, 19b) and the solenoid valves (SV1, SV2). It is noted that the compressor (11) of the primary refrigerant pipe (A) may also be used as the pressurizing/pressure-reducing compressor (18c).

In pushing out the liquid refrigerant from the tank (T) into the liquid pipe (7), the first solenoid valve (SV1) is opened and the second solenoid valve (SV2) is closed, thereby applying a high pressure into the tank (T). On the other hand, in recovering the liquid refrigerant into the tank (T) through the liquid pipe (7), the second solenoid valve (SV2) is opened and the first solenoid valve (SV1) is closed, thereby establishing a low-pressure state in the tank (T).

By repeatedly pressurizing and reducing the pressure of the tank (T), a state where the liquid refrigerant is pushed out from the tank (T) and a state where the liquid refrigerant is recovered into the tank (T) are alternately established in the same way as in the above-described embodiment. The indoor air is cooled while circulating the refrigerant in the secondary refrigerant circuit (B).

Figure 4 shows a second variant of the pressurizing/pressure-reducing mechanism (18). The pressurizing/pressure-reducing mechanism (18) changes the internal pressure of the tank (T) by directly heating and cooling the tank (T).

Specifically, a similar heat exchange unit (**18b**) to that of the above-described embodiment is disposed adjacent to the tank (**T**). If the tank (**T**) is heated by the heat exchange unit (**18b**), the internal pressure of the tank is raised by the refrigerant evaporating in the tank (**T**). On the other hand, if the tank (**T**) is cooled, the internal pressure of the tank is lowered by the refrigerant condensing therein.

By repeatedly heating and cooling the tank (**T**), the indoor air is cooled while circulating the refrigerant in the secondary refrigerant circuit (**B**).

Figure **5** shows a third variant of the pressurizing/pressure-reducing mechanism (**18**). The pressurizing/pressure-reducing mechanism (**18**) includes a pressurizing/pressure-reducing refrigerant circuit (**D**) for directly heating and cooling the tank (**T**).

The refrigerant circuit (**D**) is constructed by connecting a compressor (**D1**), a four-position selector valve (**D2**), a first heat exchanger (**D3**), an expansion valve (**D4**) and a second heat exchanger (**D5**) to each other through a refrigerant pipe (**D6**). The gas side of the first heat exchanger (**D3**) is switchably connected to the inlet side and the outlet side of the compressor (**D1**) through the four-position selector valve (**D2**). The first heat exchanger (**D3**) is disposed adjacent to the tank (**T**) and exchanges heat with the tank (**T**).

In pushing out the liquid refrigerant from the tank (**T**)

to the liquid pipe (7), the four-position selector valve (D2) is switched to the direction indicated by the solid lines. The gaseous refrigerant discharged from the compressor (D1) flows through the first heat exchanger (D3) and is condensed while applying heat to the refrigerant in the tank (T). Thereafter, the pressure of the refrigerant is reduced in the expansion valve (D4). Then, the refrigerant is evaporated in the second heat exchanger (D5) so as to return to the compressor (D1). The refrigerant in the tank (T), which has received heat from the refrigerant in the first heat exchanger (D3), is evaporated and a high pressure is built up in the tank (T). Owing to the pressure, the liquid refrigerant is pushed out from the tank (T) to the liquid pipe (7).

Conversely, in recovering the liquid refrigerant into the tank (T) through the liquid pipe (7), the four-position selector valve (D2) is switched to the direction indicated by the broken lines. The gaseous refrigerant discharged from the compressor (D1) is condensed in the second heat exchanger (D5). Thereafter, the pressure of the refrigerant is reduced in the expansion valve (D4). Then, the refrigerant flows through the first heat exchanger (D3), extracts heat from the refrigerant in the tank (T) and is evaporated to return to the compressor (D1). The refrigerant in the tank (T), the heat of which has been extracted by the refrigerant in the first heat exchanger (D3), is condensed and a low pressure is

built up in the tank (T). Owing to the pressure, the liquid refrigerant is recovered into the tank (T) through the liquid pipe (7).

By repeatedly heating and cooling the tank (T), the indoor air is cooled while circulating the refrigerant in the secondary refrigerant circuit (B).

Figure 6 shows a fourth variant of the pressurizing/pressure-reducing mechanism (18). The pressurizing/pressure-reducing mechanism (18) also includes the above-described pressurizing/pressure-reducing refrigerant circuit (D). A heat exchanger (19c) is connected to the pressurizing/pressure-reducing pipe (19) connected to the tank (T). Heat is exchanged between the heat exchanger (19c) and the first heat exchanger (D3) of the refrigerant circuit (D).

If the selection states of the four-position selector valve (D2) are alternately switched, then a high-pressure state and a low-pressure state are alternately switched in the heat exchanger (19c). As a result, the pressure of the heat exchanger (19c) is applied to the tank (T) and push-out of the liquid refrigerant from the tank (T) and recovery of the liquid refrigerant into the tank (T) are alternately performed.

By repeatedly pressurizing and reducing the pressure of the tank (T), the indoor air is cooled while circulating the refrigerant in the secondary refrigerant circuit (B).

(Second Embodiment)

Next, the second embodiment of the present invention will be described. It is noted that only the difference from the foregoing first embodiment will be described hereinafter.

The refrigerant circuitry of this embodiment constitutes an air conditioning system exclusively used for heating. The arrangement of a primary refrigerant circuit and check valves provided for the liquid pipe (7) are different from the counterparts of the first embodiment.

As shown in Figure 7, the primary refrigerant circuit (A) is constituted by connecting a compressor (11), a heat exchanger (12) on the primary heat source side, an outdoor electrically motorized expansion valve (EV2) and an outdoor heat exchanger (14) in this order through a refrigerant pipe (16). The outlet side of compressor (11) is connected to the heat exchanger (12) on the primary heat source side and the inlet side thereof is connected to the outdoor heat exchanger (14).

On the other hand, a third check valve (CV3) and a fourth check valve (CV4) are provided for the secondary refrigerant circuit (B). The third check valve (CV3) is provided between a connecting point of the connecting pipe (17) to the liquid pipe (7) and the heat exchanger (1) on the secondary heat source side, and allows only the liquid refrigerant to flow from the tank (T) to the heat exchanger (1) on the

secondary heat source side. The fourth check valve (**CV4**) is provided between the connecting point of the connecting pipe (**17**) to the liquid pipe (**7**) and the indoor heat exchangers (**3, 3**), and allows only the liquid refrigerant to flow from the indoor heat exchangers (**3**) to the tank (**T**).

It is noted that the pressurizing/pressure-reducing mechanism (**18**) of this embodiment is the same as that of the first embodiment described above.

Next, the room heating running of the refrigerant circuits (**A, B**) having the above-described arrangements will be described.

When the heating running is started, the compressor (**11**) is driven in the primary refrigerant circuit (**A**). As indicated by the solid-line arrows in Figure **7**, a high-temperature, high-pressure gaseous refrigerant discharged from the compressor (**11**) exchanges heat with the heat exchanger (**1**) on the secondary heat source side in the heat exchanger (**12**) on the primary heat source side. The refrigerant applies heat to the refrigerant in the heat exchanger (**1**) on the secondary heat source side and is condensed. Then, the pressure of the refrigerant is reduced in the outdoor electrically motorized expansion valve (**EV2**). In the outdoor heat exchanger (**14**), the refrigerant exchanges heat with the outdoor air and is evaporated to return to the compressor (**11**). This circulation operation is repeated.

On the other hand, in the secondary refrigerant circuit (B), the reservoir container (18a) of the pressurizing/pressure-reducing mechanism (18) is heated by the heat exchange unit (18b). As a result, the refrigerant in the reservoir container (18a) is evaporated and the internal pressure thereof rises (see Figure 2). As indicated by the broken-line arrows in Figure 7, the pressure is applied into the tank (T) through the pressurizing/pressure-reducing pipe (19) and pushes out the liquid refrigerant in the tank (T) to the liquid pipe (7) through the connecting pipe (17) while pushing down the water level of the liquid refrigerant. The pushed-out liquid refrigerant flows through the liquid pipe (7) toward the heat exchanger (1) on the secondary heat source side. In the heat exchanger (1) on the secondary heat source side, the liquid refrigerant exchanges heat with the refrigerant in the heat exchanger (12) on the primary heat source side and is evaporated. Thereafter, the refrigerant passes through the gas pipe (71) and exchanges heat with the indoor air in the indoor heat exchangers (3, 3). As a result, the refrigerant is condensed, thereby heating the indoor air.

If the reservoir container (18a) of the pressurizing/pressure-reducing mechanism (18) is cooled by the heat exchange unit (18b) after such an operation has been performed, the refrigerant is condensed in the reservoir container (18a) and the internal pressure thereof falls. As in-

dicated by the one-dot-chain arrows in Figure 7, the pressure is applied to the tank (T) through the pressurizing/pressure-reducing pipe (19), and the internal pressure of the tank (T) falls. As a result, the liquid refrigerant condensed in the indoor heat exchangers (3) is recovered into the tank (T) through the liquid pipe (7).

The pressurizing/pressure-reducing mechanism (18) repeatedly pressurizes/reduces the pressure of the tank (T). As a result, the refrigerant is circulated in the secondary refrigerant circuit (B), thereby heating the indoor air. Accordingly, the heat transport system of this embodiment can also transport heat in the secondary refrigerant circuit (B) without providing any driving source such as a pump for the secondary refrigerant circuit (B).

In addition, the arrangements shown in the above-described respective variants are also applicable to the pressurizing/pressure-reducing mechanism (18) of the refrigerant circuitry exclusively used for heating in the second embodiment.

It is noted that an arrangement, in which freely opening/closing solenoid valves (SV1-A, SV-B) such as those shown in Figure 8 are provided, instead of the check valves (CV1 to CV4), for the liquid pipe (7) in the first and the second embodiments and in which the opening/closing states of the respective solenoid valves (SV1-A, SV-B) are switched in accor-

dance with the states of the pressure applied from the pressurizing/pressure-reducing mechanism (**18**), may also be employed.

(Third Embodiment)

Next, the third embodiment of the present invention will be described. It is noted that only the difference from the foregoing second embodiment will be described hereinafter.

The refrigerant circuitry of this embodiment constitutes an air conditioning system of a so-called "heat pump type" which selectively cool or heat the indoor air.

Specifically, as shown in Figure **9**, the secondary refrigerant circuit (**B**) is provided with a third solenoid valve (**SV3**) and a fourth solenoid valve (**SV4**). The third solenoid valve (**SV3**) is provided between the fourth check valve (**CV4**) of the liquid pipe (**7**) and the indoor heat exchangers (**3, 3**), opens during the room heating running and closes during the cooling running. The fourth solenoid valve (**SV4**) is provided between the third check valve (**CV3**) of the liquid pipe (**7**) and the heat exchanger (**1**) on the secondary heat source side, opens during the room heating running and closes during the cooling running.

One end of a cooling liquid pipe (**34**) on the supply side is connected between the third solenoid valve (**SV3**) of the liquid pipe (**7**) and the indoor heat exchangers (**3, 3**). And

the other end of the cooling liquid pipe (**34**) on the supply side is connected between the third check valve (**CV3**) and the fourth solenoid valve (**SV4**) of the liquid pipe (**7**). A fifth solenoid valve (**SV5**) opening during the cooling running and closing during the heating running is provided for the cooling liquid pipe (**34**) on the supply side.

One end of a cooling liquid pipe (**35**) on the recovery side is connected between the fourth check valve (**CV4**) and the third solenoid valve (**SV3**) of the liquid pipe (**7**). And the other end of the cooling liquid pipe (**35**) on the recovery side is connected between the fourth solenoid valve (**SV4**) of the liquid pipe (**7**) and the heat exchanger (**1**) on the secondary heat source side. A sixth solenoid valve (**SV6**) opening during the cooling running and closing during the heating running is provided for the cooling liquid pipe (**35**) on the recovery side.

On the other hand, the primary refrigerant circuit (**A**) makes the heat exchanger (**12**) on the primary heat source side heat/cool the heat exchanger (**1**) on the secondary heat source side. Specifically, a compressor (**11**), a four-position selector valve (**22**), an outdoor heat exchanger (**14**), an outdoor electrically motorized expansion valve (**EV2**) and the heat exchanger (**12**) on the primary heat source side are connected to each other through a refrigerant pipe (**16**). The gas side of the heat exchanger (**12**) on the primary heat source side is

switched between the inlet side and the outlet side of the compressor (11) via the four-position selector valve (22).

Next, the room cooling and heating running will be described.

In the cooling running, first, the four-position selector valve (22) of the primary refrigerant circuit (A) is switched to the direction indicated by the solid lines, the fifth solenoid valve (SV5) and the sixth solenoid valve (SV6) are opened and the third solenoid valve (SV3) and the fourth solenoid valve (SV4) are closed in the secondary refrigerant circuit (B). In such a state, in the primary refrigerant circuit (A), as indicated by the solid-line arrows in Figure 9, a high-temperature, high-pressure gaseous refrigerant discharged from the compressor (11) exchanges heat with the outdoor air and is condensed in the outdoor heat exchanger (14) in the same way as in the first embodiment. Then, the pressure of the refrigerant is reduced in the outdoor electrically motorized expansion valve (EV2). In the heat exchanger (12) on the primary heat source side, the refrigerant exchanges heat with the heat exchanger (1) on the secondary heat source side, extracts heat from the refrigerant in the heat exchanger (1) on the secondary heat source side and is evaporated to return to the compressor (11). This circulation operation is repeated.

On the other hand, in the secondary refrigerant circuit

(B), when a high pressure is applied from the pressurizing/pressure-reducing mechanism (18) to the tank (T) through the pressurizing/pressure-reducing pipe (19) as indicated by the solid-line arrows in Figure 9, the water level of the liquid refrigerant in the tank (T) is pushed down and the liquid refrigerant is pushed out to the liquid pipe (7) through the connecting pipe (17). The pushed-out liquid refrigerant flows through the liquid pipe (7) and the cooling liquid pipe (34) on the supply side toward the indoor heat exchangers (3, 3). The pressure of the liquid refrigerant is reduced in the indoor electrically motorized expansion valves (EV1, EV1). Then, in the respective indoor heat exchangers (3, 3), the liquid refrigerant exchanges heat with the indoor air and is evaporated, thereby cooling the indoor air. Thereafter, the gaseous refrigerant flows through the gas pipe (6) toward the heat exchanger (1) on the secondary heat source side, exchanges heat with the heat exchanger (12) on the primary heat source side, and is condensed.

After this operation has been performed, a low pressure is applied from the pressurizing/pressure-reducing mechanism (18) to the tank (T). When the internal pressure of the tank (T) falls, the liquid refrigerant in the heat exchanger (1) on the secondary heat source side is recovered through the liquid pipe (7) and the cooling liquid pipe (35) on the recovery side to the tank (T) as indicated by the broken-line arrows in Fig-

ure 9.

By repeatedly pressurizing and reducing the pressure of the tank (T) by the pressurizing/pressure-reducing mechanism (18), the refrigerant circulates in the secondary refrigerant circuit (B), thereby cooling the indoor air.

Next, the room heating running will be described.

In the heating running, first, the four-position selector valve (22) of the primary refrigerant circuit (A) is switched to the direction indicated by the broken lines, the third solenoid valve (SV3) and the fourth solenoid valve (SV4) are opened and the fifth solenoid valve (SV5) and the sixth solenoid valve (SV6) are closed in the secondary refrigerant circuit (B). In such a state, in the primary refrigerant circuit (A), as indicated by the one-dot-chain arrows in Figure 9, a high-temperature, high-pressure gaseous refrigerant discharged from the compressor (11) exchanges heat with the heat exchanger (1) on the secondary heat source side in the heat exchanger (12) on the primary heat source side in the same way as in the second embodiment described above. The refrigerant is condensed while applying heat to the refrigerant in the heat exchanger (1) on the secondary heat source side. Thereafter, the pressure of the refrigerant is reduced in the outdoor electrically motorized expansion valve (EV2). Then, in the outdoor heat exchanger (14), the refrigerant exchanges heat with the outdoor air and is evaporated to returns to the

compressor (11). This circulation operation is repeated.

On the other hand, in the secondary refrigerant circuit (B), when a high pressure is applied from the pressurizing/pressure-reducing mechanism (18) to the tank (T) through the pressurizing/pressure-reducing pipe (19) as indicated by the one-dot-chain arrows in Figure 9, the water level of the liquid refrigerant in the tank (T) is pushed down and the liquid refrigerant is pushed out to the liquid pipe (7) through the connecting pipe (17). The pushed-out liquid refrigerant flows through the liquid pipe (7) toward the heat exchanger (1) on the secondary heat source side. Then, in the heat exchanger (1) on the secondary heat source side, the refrigerant exchanges heat with the refrigerant in the heat exchanger (12) on the primary heat source side and is evaporated. Thereafter, the refrigerant passes through the gas pipe (6), exchanges heat with the indoor air and is condensed in the respective indoor heat exchangers (3, 3), thereby heating the indoor air.

After this operation has been performed, a low pressure is applied from the pressurizing/pressure-reducing mechanism (18) to the tank (T). When the internal pressure of the tank (T) falls, the liquid refrigerant condensed in the indoor heat exchangers (3) is recovered through the liquid pipe (7) into the tank (T) as indicated by the two-dot-chain arrows in Figure 9.

By repeatedly pressurizing and reducing the pressure of the tank (T) by the pressurizing/pressure-reducing mechanism (18), the refrigerant circulates in the secondary refrigerant circuit (B), thereby heating the indoor air.

(Fourth Embodiment)

Next, the fourth embodiment of the present invention will be described. It is noted that only the difference from the foregoing first embodiment will be described hereinafter.

The refrigerant circuitry of this embodiment modifies the secondary refrigerant circuit (B) so as to continuously perform indoor air conditioning running, and is applicable to the secondary refrigerant circuit (B) in any of the foregoing first to third embodiments.

Specifically, as shown in Figure 10, a part of the liquid pipe (7) is branched into a first and a second branch pipe (7a, 7b). A first and a second tank (T1, T2) are connected to the respective pipes through connecting pipes (17a, 17b), respectively. That is to say, the tanks (T1, T2) are connected in parallel to the liquid pipe (7).

A first and a second pressurizing/pressure-reducing mechanism (18A, 18B) are individually connected to the respective upper ends of the tanks (T1, T2) through pressurizing/pressure-reducing pipes (19d, 19e), respectively. In the pressurizing/pressure-reducing mechanisms (18A, 18B), while

one pressurizing/pressure-reducing mechanism (**18A**) applies a high pressure to the tank (**T1**) connected thereto, the other pressurizing/pressure-reducing mechanism (**18B**) applies a low pressure to the tank (**T2**) connected thereto. Such high-pressure and low-pressure application states are alternately switched.

In addition, a pair of solenoid valves (**SV7** and **SV8**, **SV9** and **SV10**) to be switchably controlled in accordance with the states of the pressure applied from the associated pressurizing/pressure-reducing mechanism (**18A**, **18B**) are provided on both sides of the connecting point of the connecting pipe (**17a**, **17b**) to each branch pipe (**7a**, **7b**).

Next, the air conditioning running operations will be described.

For example, during the room heating running, when the liquid refrigerant is pushed out from the first tank (**T1**) and is recovered to the second tank (**T2**), the solenoid valve (**SV7**) located closer to the heat exchanger (**1**) on the heat source side is opened and the solenoid valve (**SV8**) located closer to the indoor heat exchangers (**3**) is closed on the first branch pipe (**7a**). On the other hand, on the second branch pipe (**7b**), the solenoid valve (**SV9**) located closer to the heat exchanger (**1**) on the heat source side is closed and the solenoid valve (**SV10**) located closer to the indoor heat exchangers (**3**) is opened. In such a state, a high pressure is applied from the

first pressurizing/pressure-reducing mechanism (**18A**) to the first tank (**T1**) and a low pressure is applied from the second pressurizing/pressure-reducing mechanism (**18B**) to the second tank (**T2**), thereby circulating the refrigerant in the secondary refrigerant circuit (**B**).

After such a running condition has lasted for a predetermined time and almost all the liquid refrigerant in the first tank (**T1**) has been drained, switching is performed to establish such a running condition that the liquid refrigerant is drained from the second tank (**T2**) and is recovered to the first tank (**T1**). During this running, the solenoid valve (**SV7**) located closer to the heat exchanger (**1**) on the heat source side is closed and the solenoid valve (**SV8**) located closer to the indoor heat exchangers (**3**) is opened on the first branch pipe (**7a**). On the other hand, on the second branch pipe (**7b**), the solenoid valve (**SV9**) located closer to the heat exchanger (**1**) on the heat source side is opened and the solenoid valve (**SV10**) located closer to the indoor heat exchangers (**3**) is closed. In such a state, a high pressure is applied from the second pressurizing/pressure-reducing mechanism (**18B**) to the second tank (**T2**) and a low pressure is applied from the first pressurizing/pressure-reducing mechanism (**18A**) to the first tank (**T1**), thereby circulating the refrigerant in the secondary refrigerant circuit (**B**).

The states of the pressures applied from the respective

pressurizing/pressure-reducing mechanisms (**18A, 18B**) to the corresponding tanks (**T1, T2**) and the opening/closing states of the respective solenoid valves (**SV7** to **SV10**) are alternately switched, thereby continuously performing the indoor air conditioning running. It is noted that the opening/closing states of the respective solenoid valves (**SV7** to **SV10**) are inverted during the room cooling running.

(Variant of Pressurizing/Pressure-reducing Mechanism)

Next, a variant of the pressurizing/pressure-reducing mechanism (**18**) applicable to the secondary refrigerant circuit (**B**) of the foregoing fourth embodiment will be described with reference to Figure **11**.

In this variant, a pressurizing/pressure-reducing compressor (**D1**) is provided. Specifically, the compressor (**D1**) is connected to pressurizing/pressure-reducing pipes (**19d, 19e**) extending from the respective tanks (**T1, T2**) via a four-position selector valve (**D2**). In accordance with the switching operations of the four-position selector valve (**D2**), a state (i.e., a state indicated by the broken lines in Figure **11**) in which the first tank (**T1**) is connected to the outlet side of the compressor (**D1**) and the second tank (**T2**) is connected to the inlet side of the compressor (**D1**), and a state (i.e., a state indicated by the solid lines in Figure **11**) in which the first tank (**T1**) is connected to the inlet side of

the compressor (**D1**) and the second tank (**T2**) is connected to the outlet side of the compressor (**D1**) are switched.

In accordance with the switching operations of the four-position selector valve (**D2**), the pressure application states with respect to the respective tanks (**T1, T2**) are alternately switched in the same way as in the fourth embodiment described above. As a result, the indoor air conditioning running can be performed continuously.

It is noted that, in this variant, check valves (**CV7** to **CV10**) are used instead of the respective solenoid valves (**SV7** to **SV10**) of the fourth embodiment described above. That is to say, the circuit shown in Figure **11** is a circuit for heating. In a circuit exclusively used for cooling, these check valves are replaced by the counterparts in which the refrigerant is allowed in flow in an inverted direction.

(Fifth Embodiment)

Next, a specific secondary refrigerant circuit (**B**), which is provided with a plurality of tanks (**T1, T2**) in the same way as in the fourth embodiment described above, for continuously performing air conditioning running, will be described.

The circuitry shown in Figure **12** constitutes an air conditioning system exclusively used for cooling. In this secondary refrigerant circuit (**B**), a first and a second driving heat exchanger (**E1, E2**) are connected to pressuriz-

ing/pressure-reducing pipes (**19d, 19e**) extending from the respective tanks (**T1, T2**) and a pressurizing/pressure-reducing refrigerant circuit (**D**) is also provided. By exchanging heat between the refrigerant circuit (**D**) and the driving heat exchangers (**E1, E2**), pressure for circulating the refrigerant is applied to the respective tanks (**T1, T2**).

The refrigerant circuit (**D**) will be described. A compressor (**D1**), a first heat exchanger (**D3**) that can exchange heat with the first driving heat exchanger (**E1**), an expansion valve (**D4**) and a second heat exchanger (**D5**) that can exchange heat with the second driving heat exchanger (**E2**) are connected to each other through a refrigerant pipe (**D6**). Specifically, a gas-side pipe (**D3-G**) of the first heat exchanger (**D3**) is branched into two, which are connected to the outlet side and the inlet side of the compressor (**D1**), respectively. Of these branch pipes, a pipe (**D3-G1**) on the outlet side of the compressor (**D1**) is provided with a first outlet-side solenoid valve (**SV-O1**) and a pipe (**D3-G2**) on the inlet side thereof is provided with a first inlet-side solenoid valve (**SV-I1**).

Similarly, a gas-side pipe (**D5-G**) of the second heat exchanger (**D5**) is also branched into two, which are connected to the outlet side and the inlet side of the compressor (**D1**), respectively. A pipe (**D5-G1**) on the outlet side of the compressor (**D1**) is provided with a second outlet-side solenoid valve (**SV-O2**) and a pipe (**D5-G2**) on the inlet side thereof is

provided with a second inlet-side solenoid valve (**SV-I2**). Also, the liquid sides of the respective heat exchangers (**D3, D5**) are coupled to each other through a liquid pipe (**D6-L**) via the expansion valve (**D4**).

Reservoirs (**20, 21**) for reserving a refrigerant for driving are connected to the respective driving heat exchangers (**E1, E2**). A check valve (**CV1-A**) allowing only the liquid refrigerant to flow from the heat exchanger (**1**) on the secondary heat source side to the first tank (**T1**) is provided between the connecting point of a connecting pipe (**17a**) to the first branch pipe (**7a**) of the liquid pipe (**7**) and the heat exchanger (**1**) on the secondary heat source side. A check valve (**CV2-A**) allowing only the liquid refrigerant to flow from the first tank (**T1**) to the indoor heat exchangers (**3, 3**) is provided between the connecting point of the connecting pipe (**17a**) to the first branch pipe (**7a**) and the indoor heat exchangers (**3, 3**).

On the other hand, a check valve (**CV1-B**) allowing only the liquid refrigerant to flow from the heat exchanger (**1**) on the secondary heat source side to the second tank (**T2**) is provided between the connecting point of a connecting pipe (**17b**) to the second branch pipe (**7b**) of the liquid pipe (**7**) and the heat exchanger (**1**) on the secondary heat source side. A check valve (**CV2-B**) allowing only the liquid refrigerant to flow from the second tank (**T2**) to the indoor heat exchangers (**3, 3**) is provided between the connecting point of the con-

necting pipe (**17b**) to the second branch pipe (**7b**) and the indoor heat exchangers (**3, 3**).

The other arrangements are the same as those of the primary and the secondary refrigerant circuits (**A, B**) of the first embodiment described above.

Next, the room cooling running operation will be described.

First, in the pressurizing/pressure-reducing refrigerant circuit (**D**), the first outlet-side solenoid valve (**SV-O1**) and the second inlet-side solenoid valve (**SV-I2**) are opened, the first inlet-side solenoid valve (**SV-I1**) and the second outlet-side solenoid valve (**SV-O2**) are closed and the compressor (**D1**) is driven. The high-temperature, high-pressure gaseous refrigerant discharged from the compressor (**D1**) flows through the gas-side pipe (**D3-G1**) and the first heat exchanger (**D3**) as indicated by the solid-line arrows in Figure **12**. In the first heat exchanger (**D3**), the refrigerant exchanges heat with the first driving heat exchanger (**E1**), applies heat to the refrigerant in the first driving heat exchanger (**E1**) and is condensed. Thereafter, the pressure of the liquid refrigerant is reduced by the expansion valve (**D4**) in the liquid pipe (**D6-L**), exchanges heat with the second driving heat exchanger (**E2**) in the second heat exchanger (**D5**), extracts heat from the refrigerant in the second driving heat exchanger (**E2**) and is evaporated to return to the compressor (**D1**)

through the gas-side pipe (**D5-G2**). This circulation operation is repeated.

In accordance with this refrigerant circulation operation, the refrigerant is evaporated and a high pressure is built up in the first driving heat exchanger (**E1**), while the refrigerant is condensed and a low pressure is built up in the second driving heat exchanger (**E2**). Thus, a high pressure is applied to the first tank (**T1**) and a low pressure is applied to the second tank (**T2**). In the same way as in the fourth embodiment described above, the operation of draining the liquid refrigerant from the first tank (**T1**) and the operation of recovering the liquid refrigerant into the second tank (**T2**) are simultaneously performed, thereby circulating the refrigerant in the secondary refrigerant circuit (**B**).

After such a running state has lasted for a predetermined period of time, the opening/closing states of the respective solenoid valves in the pressurizing/pressure-reducing refrigerant circuit (**D**) are switched. Specifically, the first outlet-side solenoid valve (**SV-O1**) and the second inlet-side solenoid valve (**SV-I2**) are closed and the first inlet-side solenoid valve (**SV-I1**) and the second outlet-side solenoid valve (**SV-O2**) are opened. As a result, the refrigerant flows as indicated by the broken-line arrows in Figure **12**. The refrigerant is condensed and a low pressure is built up in the first driving heat exchanger (**E1**), while the refrigerant is

evaporated and a high pressure is built up in the second driving heat exchanger (**E2**). Thus, a low pressure is applied to the first tank (**T1**) and a high pressure is applied to the second tank (**T2**). As a result, the operation of draining the liquid refrigerant from the second tank (**T2**) and the operation of recovering the liquid refrigerant into the first tank (**T1**) are simultaneously performed, thereby circulating the refrigerant in the secondary refrigerant circuit (**B**).

The pressure application states with respect to the respective tanks (**T1, T2**) are alternately switched, thereby continuously performing room cooling running. Also, the gaseous refrigerant recovered from the tank (**T2**), to which a low pressure is applied, into the driving heat exchanger (**E2**) is condensed in the driving heat exchanger (**E2**) so as to be temporarily reserved in the reservoir (**21**). And, when the opening/closing states of the respective solenoid valves (**SV-O1** to **SV-I2**) are switched, the refrigerant is evaporated in the driving heat exchanger (**E2**), thereby applying a high pressure to the tank (**T2**).

(Sixth Embodiment)

Next, an air conditioning system, which is provided with a plurality of tanks (**T1, T2**) for performing continuous running and which is an air conditioning system exclusively used for heating, will be described. It is noted that, in this

embodiment, only the difference from the foregoing fifth embodiment will be described hereinafter.

As shown in Figure 13, this secondary refrigerant circuit (B) is different from that of the fifth embodiment in the arrangements of the check valves provided for the respective branch pipes (7a, 7b) of the liquid pipe (7).

Specifically, a check valve (CV3-A) allowing only the liquid refrigerant to flow from the first tank (T1) to the heat exchanger (1) on the secondary heat source side is provided between the connecting point of the connecting pipe (17a) to the first branch pipe (7a) of the liquid pipe (7) and the heat exchanger (1) on the secondary heat source side. A check valve (CV4-A) allowing only the liquid refrigerant to flow from the indoor heat exchangers (3, 3) to the first tank (T1) is provided between the connecting point of the connecting pipe (17a) to the first branch pipe (7a) and the indoor heat exchangers (3, 3).

On the other hand, a check valve (CV3-B) allowing only the liquid refrigerant to flow from the second tank (T2) to the heat exchanger (1) on the secondary heat source side is provided between the connecting point of a connecting pipe (17b) to the second branch pipe (7b) of the liquid pipe (7) and the heat exchanger (1) on the secondary heat source side. A check valve (CV4-B) allowing only the liquid refrigerant to flow from the indoor heat exchangers (3, 3) to the second tank

(T2) is provided between the connecting point of the connecting pipe (17b) to the second branch pipe (7b) and the indoor heat exchangers (3, 3).

The other arrangements are the same as the counterparts of the secondary refrigerant circuit of the fifth embodiment described above.

Next, the room heating running operation will be described.

In the same way as in the fifth embodiment described above, the respective solenoid valves of the pressurizing/pressure-reducing refrigerant circuit (D) are switched. Specifically, a state where the first outlet-side solenoid valve (SV-O1) and the second inlet-side solenoid valve (SV-I2) are opened and the first inlet-side solenoid valve (SV-I1) and the second outlet-side solenoid valve (SV-O2) are closed, and a state where the first outlet-side solenoid valve (SV-O1) and the second inlet-side solenoid valve (SV-I2) are closed and the first inlet-side solenoid valve (SV-I1) and the second outlet-side solenoid valve (SV-O2) are opened, are alternately and repeatedly established. Thus, a running condition where a high pressure is applied to the first tank (T1) and a low pressure is applied to the second tank (T2), and a running condition where a low pressure is applied to the first tank (T1) and a high pressure is applied to the second tank (T2), are alternately and repeatedly estab-

lished. And, the liquid refrigerant drained from one tank (T1) to the liquid pipe (7) is evaporated in the heat exchanger (1) on the secondary heat source side, and then condensed in the indoor heat exchangers (3, 3), thereby heating the indoor air. Thereafter, the liquid refrigerant is recovered into the other tank (T2). This refrigerant circulation operation is repeated, thereby continuously performing the room heating running.

(Seventh Embodiment)

Next, the seventh embodiment of the present invention will be described. It is noted that only the difference from the foregoing sixth embodiment will be described hereinafter.

The secondary refrigerant circuit (B) of this embodiment is applied to an air conditioning system of a heat pump type.

Specifically, as shown in Figure 14, a third solenoid valve (SV3) opening during the room heating running and closing during the cooling running is provided between a branch point of the branch pipes (7a, 7b) of the liquid pipe (7) on the side of the indoor heat exchangers (3) and the indoor heat exchangers (3, 3). A fourth solenoid valve (SV4) opening during the room heating running and closing during the cooling running is provided between a branch point of the branch pipes (7a, 7b) of the liquid pipe (7) on the side of the heat exchanger (1) on the secondary heat source side and the heat

exchanger (1) on the secondary heat source side.

In addition, one end of a cooling liquid pipe (34) on the supply side is connected between the third solenoid valve (SV3) of the liquid pipe (7) and the indoor heat exchangers (3, 3). And the other end of the cooling liquid pipe (34) on the supply side is connected to the downstream of the third check valve (CV3-A) of the first branch pipe (7a). The cooling liquid pipe (34) on the supply side is provided with a fifth solenoid valve (SV5) opening during the cooling running and closing during the heating running.

Moreover, one end of a cooling liquid pipe (35) on the recovery side is connected between the fourth solenoid valve (SV4) of the liquid pipe (7) and the heat exchanger (1) on the secondary heat source side. And the other end of the cooling liquid pipe (35) on the recovery side is connected to the upstream of the fourth check valve (CV4-B) of the second branch pipe (7b). The cooling liquid pipe (35) on the recovery side is provided with a sixth solenoid valve (SV6) opening during the cooling running and closing during the heating running.

The other arrangements are the same as the counterparts of the sixth embodiment. Also, in this embodiment, the primary refrigerant circuit of the third embodiment described above is used as the primary refrigerant circuit.

Hereinafter, the room cooling and heating running will

be described.

During the cooling running, in accordance with the switching operations of the respective solenoid valves in the pressurizing/pressure-reducing refrigerant circuit (D), as indicated by the solid-line arrows in Figure 14, the liquid refrigerant drained from one tank (T1) to the branch pipe (7a) is passed through the cooling liquid pipe (34) on the supply side in the same way as in the fifth embodiment described above. The pressure of the refrigerant is reduced in the indoor electrically motorized expansion valves (EV1, EV1). Then, the refrigerant is evaporated in the indoor heat exchangers (3, 3), thereby cooling the indoor air. Thereafter, the gaseous refrigerant is condensed in the heat exchanger (1) on the secondary heat source side, passed through the cooling liquid pipe (35) on the recovery side and then recovered into the other tank (T2). This refrigerant circulation operation is repeated and a tank draining the liquid refrigerant and a tank recovering the refrigerant are alternately switched, thereby continuously performing the room cooling running.

On the other hand, during the room heating running, in accordance with the switching operations of the respective solenoid valves in the pressurizing/pressure-reducing refrigerant circuit (D), as indicated by the broken-line arrows in Figure 14, the liquid refrigerant drained from one tank (T1) to the liquid pipe (7) flows through the liquid pipe (7) to-

ward the heat exchanger (1) on the secondary heat source side in the same way as in the sixth embodiment described above. In the heat exchanger (1) on the secondary heat source side, the refrigerant exchanges heat with the refrigerant in the heat exchanger (12) on the primary heat source side and is evaporated. Thereafter, the refrigerant is passed through the gas pipe (6), exchanges heat with the indoor air and is condensed in the indoor heat exchangers (3, 3), thereby heating the indoor air. Thereafter, the liquid refrigerant condensed in the indoor heat exchanger (3) is passed through the liquid pipe (7) and recovered into the other tank (T2). This refrigerant circulation operation is repeated and a tank draining the liquid refrigerant and a tank recovering the refrigerant are alternately switched, thereby continuously performing the room heating running.

(Eighth Embodiment)

Next, the eighth embodiment of the present invention will be described. It is noted that only the difference from the foregoing sixth embodiment will be described hereinafter.

The secondary refrigerant circuit (B) of this embodiment is also applied to an air conditioning system of a heat pump type. The arrangement of the primary refrigerant circuit (A) is the same as that of the third embodiment described above and thus the description thereof will be omitted herein.

As shown in Figure 15, the secondary refrigerant circuit (B) includes a four-position selector valve (10) in the liquid pipe (7). The four-position selector valve (10) is connected to a first liquid pipe (7A) extending from the liquid side of the heat exchanger (1) on the secondary heat source side, to a second and a third liquid pipe (7B, 7C) respectively extending from a first and a second branch points (X, Y) (where X is a branch point closer to the heat exchanger (1) on the secondary heat source side and Y is a branch point closer to the indoor heat exchanger (3)) on both ends of the respective branch pipes (7a, 7b) of the liquid pipe (7) and to a fourth liquid pipe (7D) extending from the indoor heat exchangers (3, 3, 3) to the tanks (T1, T2).

The four-position selector valve (10) switches the connection states between the liquid side of the heat exchanger (1) on the secondary heat source side and the respective branch points (X, Y) of the branch pipes (7a, 7b) and between the liquid sides of the indoor heat exchangers (3, 3, 3) and the respective branch points (X, Y) of the branch pipes (7a, 7b). Specifically, a state where the liquid side of the heat exchanger (1) on the secondary heat source side is connected to the first branch point (X) and the liquid sides of the indoor heat exchangers (3, 3, 3) are connected to the second branch point (Y) (i.e., a selected state indicated by the solid lines in Figure 15) and a state where the liquid side of

the heat exchanger (1) on the secondary heat source side is connected to the second branch point (Y) and the liquid sides of the indoor heat exchangers (3, 3, 3) are connected to the first branch point (X) (i.e., a selected state indicated by the broken lines in Figure 15) are switched.

Also, the secondary refrigerant circuit (B) of this embodiment includes three indoor heat exchangers (3). The other arrangements are the same as the counterparts of the sixth embodiment described above. In Figure 15, (28) denotes an accumulator.

Hereinafter, the room cooling and heating running will be described.

First, during the cooling running, the four-position selector valve (22) is switched to the direction indicated by the solid lines in the primary refrigerant circuit (A) and the four-position selector valve (10) is switched to the direction indicated by the broken lines in the secondary refrigerant circuit (B). In such a state, the compressor (11) of the primary refrigerant circuit (A) and the compressor (D1) of the pressurizing/pressure-reducing refrigerant circuit (D) in the secondary refrigerant circuit (B) are both driven.

As a result, in accordance with the switching operations of the respective solenoid valves in the pressurizing/pressure-reducing refrigerant circuit (D), as indicated by the solid-line arrows in Figure 15, the liquid refrigerant

drained from one tank (T1) is passed through the second liquid pipe (7B) and flows through the four-position selector valve (10) and the fourth liquid pipe (7D) in the same way as in the fifth embodiment. Then, the pressure of the refrigerant is reduced in the indoor electrically motorized expansion valves (EV1, EV1). The refrigerant is evaporated in the indoor heat exchangers (3, 3), thereby cooling the indoor air. Thereafter, the gaseous refrigerant is passed through the gas pipe (6), condensed in the heat exchanger (1) on the secondary heat source side, passed through the first liquid pipe (7A), the four-position selector valve (10) and the third liquid pipe (7C) and then recovered into the other tank (T2). This refrigerant circulation operation is repeated and a tank draining the liquid refrigerant and a tank recovering the refrigerant are alternately switched, thereby continuously performing the room cooling running.

On the other hand, during the room heating running, the four-position selector valve (22) is switched to the direction indicated by the broken lines in the primary refrigerant circuit (A) and the four-position selector valve (10) is switched to the direction indicated by the solid lines in the secondary refrigerant circuit (B). In such a state, the compressor (11) of the primary refrigerant circuit (A) and the compressor (D1) of the pressurizing/pressure-reducing refrigerant circuit (D) in the secondary refrigerant circuit (B) are

both driven.

As a result, in accordance with the switching operations of the respective solenoid valves in the pressurizing/pressure-reducing refrigerant circuit (D), as indicated by the broken-line arrows in Figure 15, the liquid refrigerant drained from one tank (T1) flows through the second liquid pipe (7B), the four-position selector valve (10) and the first liquid pipe (7A) in the same way as in the sixth embodiment described above. Then, in the heat exchanger (1) on the secondary heat source side, the refrigerant exchanges heat with the refrigerant in the heat exchanger (12) on the primary heat source side and is evaporated. The gaseous refrigerant is passed through the gas pipe (6), is introduced into the indoor heat exchangers (3, 3), exchanges heat with the indoor air and is condensed, thereby heating the indoor air. Thereafter, the liquid refrigerant condensed in the indoor heat exchanger (3) is passed through the fourth liquid pipe (7D), the four-position selector valve (10) and the third liquid pipe (7C) and is recovered into the other tank (T2). This refrigerant circulation operation is repeated and a tank draining the liquid refrigerant and a tank recovering the refrigerant are alternately switched, thereby continuously performing the room heating running.

(Ninth Embodiment)

Next, the ninth embodiment of the present invention will be described. It is noted that only the difference from the foregoing eighth embodiment will be described hereinafter. The secondary refrigerant circuit (**B**) of this embodiment is also applied to an air conditioning system of a heat pump type. The arrangement of the primary refrigerant circuit (**A**) is the same as that of the third embodiment described above and thus the description thereof will be omitted herein. This embodiment is different from the eighth embodiment in the arrangement of the pressurizing/pressure-reducing mechanism (**18**). Thus, only the pressurizing/pressure-reducing mechanism (**18**) will be described.

As shown in Figure **16**, in the secondary refrigerant circuit (**B**), a first and a second driving heat exchanger (**E1**, **E2**) are connected to a first and a second pressurizing/pressure-reducing pipes (**19d**, **19e**) extending from the respective tanks (**T1**, **T2**), and a pressurizing/pressure-reducing refrigerant circuit (**D**) is also provided. By exchanging heat between the refrigerant circuit (**D**) and the driving heat exchangers (**E1**, **E2**), pressures for circulating the refrigerant are applied to the respective tanks (**T1**, **T2**).

Specifically, the first pressurizing/pressure-reducing pipe (**19d**) connected to the upper end of the first tank (**T1**) is branched into a first pressurizing/pressure-reducing branch

pipe (**19d-A**) leading to the first driving heat exchanger (**E1**) and a second pressurizing/pressure-reducing branch pipe (**19d-B**) leading to the second driving heat exchanger (**E2**). A first pressurizing/pressure-reducing solenoid valve (**SV-1**) and a second pressurizing/pressure-reducing solenoid valve (**SV-2**) are provided for the first pressurizing/pressure-reducing branch pipe (**19d-A**) and the second pressurizing/pressure-reducing branch pipe (**19d-B**), respectively.

On the other hand, the second pressurizing/pressure-reducing pipe (**19e**) connected to the upper end of the second tank (**T2**) is branched into a third pressurizing/pressure-reducing branch pipe (**19e-A**) leading to a point between the first pressurizing/pressure-reducing solenoid valve (**SV-1**) of the first pressurizing/pressure-reducing branch pipe (**19d-A**) and the first driving heat exchanger (**E1**) and a fourth pressurizing/pressure-reducing branch pipe (**19e-B**) leading to a point between the second pressurizing/pressure-reducing solenoid valve (**SV-2**) of the second pressurizing/pressure-reducing branch pipe (**19d-B**) and the second driving heat exchanger (**E2**). A third pressurizing/pressure-reducing solenoid valve (**SV-3**) and a fourth pressurizing/pressure-reducing solenoid valve (**SV-4**) are provided for the third pressurizing/pressure-reducing branch pipe (**19e-A**) and the fourth pressurizing/pressure-reducing branch pipe (**19e-B**), respectively.

The refrigerant circuit (**D**) will be described. A com-

pressor (**D1**), a second heat exchanger (**D3**) that can exchange heat with the second driving heat exchanger (**E2**), an expansion valve (**D4**) and a first heat exchanger (**D5**) that can exchange heat with the first driving heat exchanger (**E1**) are connected in this order through a refrigerant pipe (**D6**).

Reservoirs (**20, 21**) for reserving a refrigerant for driving are connected to the respective driving heat exchangers (**E1, E2**) via check valves (**CV-1, CV-2**) and solenoid valves (**SV-5** to **SV-8**). Specifically, the lower end of the first driving heat exchanger (**E1**) is connected to the lower ends of the respective reservoirs (**20, 21**) via check valves (**CV1, CV1**) allowing only the liquid refrigerant to flow from the first driving heat exchanger (**E1**) to the respective reservoirs (**20, 21**). The lower end of the second driving heat exchanger (**E2**) is connected to the lower ends of the respective reservoirs (**20, 21**) via check valves (**CV2, CV2**) allowing only the liquid refrigerant to flow from the respective reservoirs (**20, 21**) to the second driving heat exchanger (**E2**).

The upper end of the first reservoir (**20**) is connected to the first pressurizing/pressure-reducing branch pipe (**19d-A**) via the fifth solenoid valve (**SV-5**) and to the second pressurizing/pressure-reducing branch pipe (**19d-B**) via the sixth solenoid valve (**SV-6**). The upper end of the second reservoir (**21**) is connected to the first pressurizing/pressure-reducing branch pipe (**19d-A**) via the seventh solenoid valve (**SV-7**) and

to the second pressurizing/pressure-reducing branch pipe (19d-B) via the eighth solenoid valve (SV-8).

The other arrangements are the same as those of the primary and the secondary refrigerant circuits (A, B) of the eighth embodiment described above.

Next, the room cooling and heating running will be described.

First, during the cooling running, the four-position selector valve (22) is switched to the direction indicated by the solid lines in the primary refrigerant circuit (A). On the other hand, in the secondary refrigerant circuit (B), the four-position selector valve (10) is switched to the direction indicated by the broken lines. Furthermore, in the pressurizing/pressure-reducing mechanism (18), the second solenoid valve (SV-2), the third solenoid valve (SV-3), the sixth solenoid valve (SV-6) and the seventh solenoid valve (SV-7) are opened, while the first solenoid valve (SV-1), the fourth solenoid valve (SV-4), the fifth solenoid valve (SV-5) and the eighth solenoid valve (SV-8) are closed.

In such a state, the compressor (11) of the primary refrigerant circuit (A) and the compressor (D1) of the pressurizing/pressure-reducing refrigerant circuit (D) in the secondary refrigerant circuit (B) are both driven. And, in the pressurizing/pressure-reducing refrigerant circuit (D), as indicated by the solid-line arrows in Figure 17, the refriger-

ant discharged from the compressor (**D1**) flows through the first heat exchanger (**D3**). In the first heat exchanger (**D3**), the refrigerant exchanges heat with the second driving heat exchanger (**E2**), applies heat to the refrigerant in the second driving heat exchanger (**E2**) and is condensed. Thereafter, the pressure of the refrigerant is reduced in the expansion valve (**D4**). And, in the second heat exchanger (**D5**), the refrigerant exchanges heat with the first driving heat exchanger (**E1**), extracts heat from the refrigerant in the first driving heat exchanger (**E1**) and is evaporated to return to the compressor (**D1**). This circulation operation is repeated.

As a result of the refrigerant circulation operation, the refrigerant is evaporated and a high pressure is built up in the second driving heat exchanger (**E2**), while the refrigerant is condensed and a low pressure is built up in the first driving heat exchanger (**E1**). Thus, a high pressure is applied through the second pressurizing/pressure-reducing branch pipe (**19d-B**) to the first tank (**T1**) and a low pressure is applied through the third pressurizing/pressure-reducing branch pipe (**19e-A**) to the second tank (**T2**). In the same way as in the fourth embodiment, the operation of draining the liquid refrigerant from the first tank (**T1**) and the operation of recovering the liquid refrigerant into the second tank (**T2**) are simultaneously performed, thereby circulating the refrigerant in the secondary refrigerant circuit (**B**).

Specifically, as indicated by the broken-line arrows in Figure 17, the liquid refrigerant drained from the first tank (T1) passes through the second liquid pipe (7B) and flows through the four-position selector valve (10) and the fourth liquid pipe (7D). Thereafter, the pressure of the refrigerant is reduced in the indoor electrically motorized expansion valves (EV1, EV1, ...). And the refrigerant is evaporated in the indoor heat exchangers (3, 3, ...), thereby cooling the indoor air. Thereafter, the gaseous refrigerant is passed through the gas pipe (6), condensed in the heat exchanger (1) on the secondary heat source side and passed through the first liquid pipe (7A), the four-position selector valve (10) and the third liquid pipe (7C) so as to be recovered into the second tank (T2). This refrigerant circulation operation is repeated.

In this case, the gaseous refrigerant, which has been recovered from the second tank (T2) into the first driving heat exchanger (E1) through the third pressurizing/pressure-reducing branch pipe (19e-A), is condensed and reserved in the second reservoir (21). The pressure of the second driving heat exchanger (E2) is equalized with that of the first reservoir (20), and the liquid refrigerant in the reservoir (20) is supplied to the second driving heat exchanger (E2).

After such a refrigerant circulation operation has been performed for a predetermined period of time, the switching

operations are performed for the respective solenoid valves. Specifically, the first solenoid valve (**SV-1**), the fourth solenoid valve (**SV-4**), the fifth solenoid valve (**SV-5**) and the eighth solenoid valve (**SV-8**) are opened, while the second solenoid valve (**SV-2**), the third solenoid valve (**SV-3**), the sixth solenoid valve (**SV-6**) and the seventh solenoid valve (**SV-7**) are closed.

When the respective solenoid valves are switched in such a manner, a high pressure is applied through the fourth pressurizing/pressure-reducing branch pipe (**19e-B**) to the second tank (**T2**) and a low pressure is applied through the first pressurizing/pressure-reducing branch pipe (**19d-A**) to the first tank (**T1**). As a result, the operation of draining the liquid refrigerant from the second tank (**T2**) and the operation of recovering the liquid refrigerant into the first tank (**T1**) are simultaneously performed, thereby circulating the refrigerant in the secondary refrigerant circuit (**B**).

Specifically, as indicated by the one-dot-chain arrows in Figure **17**, the liquid refrigerant drained from the second tank (**T2**) passes through the second liquid pipe (**7B**) and flows through the four-position selector valve (**10**) and the fourth liquid pipe (**7D**). Thereafter, the pressure of the refrigerant is reduced in the indoor electrically motorized expansion valves (**EV1, EV1, …**). And the refrigerant is evaporated in the indoor heat exchangers (**3, 3, …**), thereby cooling the in-

door air. Thereafter, the gaseous refrigerant is passed through the gas pipe (6), condensed in the heat exchanger (1) on the secondary heat source side and passed through the first liquid pipe (7A), the four-position selector valve (10) and the third liquid pipe (7C) so as to be recovered into the first tank (T1). This refrigerant circulation operation is repeated.

In this case, the gaseous refrigerant, which has been recovered from the first tank (T1) into the first driving heat exchanger (E1) through the first pressurizing/pressure-reducing branch pipe (19d-A), is condensed and reserved in the first reservoir (20). The pressure of the second driving heat exchanger (E2) is equalized with that of the second reservoir (21), and the liquid refrigerant in the reservoir (21) is supplied to the second driving heat exchanger (E2).

A tank draining the liquid refrigerant and a tank recovering the refrigerant are alternately switched in such a manner, thereby continuously performing the room cooling running.

On the other hand, during the room heating running, the four-position selector valve (22) is switched to the direction indicated by the broken lines in the primary refrigerant circuit (A). In the secondary refrigerant circuit (B), the four-position selector valve (10) is switched to the direction indicated by the solid lines. Furthermore, in the pres-

surizing/pressure-reducing mechanism (**18**), the second solenoid valve (**SV-2**), the third solenoid valve (**SV-3**), the sixth solenoid valve (**SV-6**) and the seventh solenoid valve (**SV-7**) are opened, while the first solenoid valve (**SV-1**), the fourth solenoid valve (**SV-4**), the fifth solenoid valve (**SV-5**) and the eighth solenoid valve (**SV-8**) are closed in the same way as in the case of the cooling running described above. As a result, a high pressure is applied to the first tank (**T1**) and a low pressure is applied to the second tank (**T2**).

Alternatively, the first solenoid valve (**SV-1**), the fourth solenoid valve (**SV-4**), the fifth solenoid valve (**SV-5**) and the eighth solenoid valve (**SV-8**) are opened and the second solenoid valve (**SV-2**), the third solenoid valve (**SV-3**), the sixth solenoid valve (**SV-6**) and the seventh solenoid valve (**SV-7**) are closed. As a result, a high pressure is applied to the second tank (**T2**) and a low pressure is applied to the first tank (**T1**). These two states are alternately switched.

As a result, as indicated by the broken-line and one-dot-chain arrows in Figure **18**, the liquid refrigerant drained from one tank (**T1**) passes through the second liquid pipe (**7B**) and flows through the four-position selector valve (**10**) and the first liquid pipe (**7A**). Thereafter, in the heat exchanger (**1**) on the secondary heat source side, the refrigerant exchanges heat with the refrigerant in the heat exchanger (**12**)

on the primary heat source side and is evaporated. The gaseous refrigerant is passed through the gas pipe (6) and introduced into the indoor heat exchangers (3, 3), in which the refrigerant exchanges heat with the indoor air and is condensed, thereby heating the indoor air. Then, the liquid refrigerant condensed in the indoor heat exchanger (3) is passed through the fourth liquid pipe (7D), the four-position selector valve (10) and the third liquid pipe (7C) and recovered into the other tank (T2). This refrigerant circulation operation is repeated and a tank draining the liquid refrigerant and a tank recovering the refrigerant are alternately switched, thereby continuously performing the room heating running.

(Tenth Embodiment)

Next, the tenth embodiment of the present invention will be described with reference to Figures 19 to 21. This embodiment is a variant of the primary refrigerant circuit (A) to be combined with substantially the same circuit as the secondary refrigerant circuit (B) of the ninth embodiment described above. Also, this embodiment is a heat pump circuit for cooling and heating.

First, the primary refrigerant circuit (A) will be described.

The primary refrigerant circuit (A) is constituted by connecting a compressor (11), a four-position selector valve

(22), an outdoor heat exchanger (14), an outdoor electrically motorized expansion valve (EVW) and a heat exchanger (12) on the primary heat source side through a refrigerant pipe (16). The gas side of heat exchanger (12) on the primary heat source side is switched between the inlet side and the outlet side of the compressor (11) via the four-position selector valve (22).

A check valve (CV-1) allowing only the refrigerant to flow from the outdoor electrically motorized expansion valve (EVW) to the outdoor heat exchanger (14) is provided between the outdoor heat exchanger (14) and the outdoor electrically motorized expansion valve (EVW). The primary refrigerant circuit (A) includes a heating heat exchanger (D3) and a cooling heat exchanger (D5) both for exchanging heat with the secondary refrigerant circuit (B).

One end (i.e., the lower end in Figure 19) of the heating heat exchanger (D3) is connected between the outdoor heat exchanger (14) and the check valve (CV-1) of the refrigerant pipe (16) through a first cooling liquid line (CL-1) including a check valve (CV-2). The other end (i.e., the upper end in Figure 19) thereof is connected between the outdoor electrically motorized expansion valve (EVW) and the check valve (CV-1) in the refrigerant pipe (16) through a second cooling liquid line (CL-2).

A first solenoid valve (SV-1) opening during cooling and closing during heating is provided for the second cooling

liquid line (CL-2). One end of a heating gas line (WGL) is connected between the heating heat exchanger (D3) and the first solenoid valve (SV-1) in the second cooling liquid line (CL-2) and the other end of the heating gas line (WGL) is connected to the outlet side of the compressor (11). A second solenoid valve (SV-2) opening during heating and closing during cooling is provided for the heating gas line (WGL).

A heating liquid line (WLL) is connected between the first cooling liquid line (CL-1) and the heat exchanger (12) on the primary heat source side. A third solenoid valve (SV-3) opening during heating and closing during cooling is provided for the heating liquid line (WLL). Furthermore, one end of the cooling heat exchanger (D5)(i.e., the lower end in Figure 19) is connected to the inlet side of the compressor (11) through an inlet gas line (IGL) and the other end thereof (i.e., the upper end in Figure 19) is connected to the first cooling liquid line (CL-1) through the cooling liquid line (CLL). A motor operated valve (D4) is provided for the cooling liquid line (CLL).

On the other hand, in the secondary refrigerant circuit (B), a heat exchanger (1) on the secondary heat source side for exchanging heat with the heat exchanger (12) on the primary heat source side and indoor heat exchangers (3, 3, 3) are connected to each other through a gas pipe (6) and a liquid pipe (7). A part of the liquid pipe (7) is branched into a

first and a second branch pipe (**7a, 7b**), to which a first and a second tank (**T1, T2**) reserving the liquid refrigerant therein are respectively connected through connection pipes (**17a, 17b**). A check valve (**CV3-A**) allowing only the liquid refrigerant to flow from the first tank (**T1**) to the heat exchanger (**1**) on the secondary heat source side is provided between the connecting point of the connecting pipe (**17a**) to the first branch pipe (**7a**) of the liquid pipe (**7**) and the heat exchanger (**1**) on the secondary heat source side. A check valve (**CV4-A**) allowing only the liquid refrigerant to flow from the indoor heat exchangers (**3, 3**) to the first tank (**T1**) is provided between the connecting point of the connecting pipe (**17a**) to the first branch pipe (**7a**) and the indoor heat exchangers (**3, 3**).

On the other hand, a check valve (**CV3-B**) allowing only the liquid refrigerant to flow from the second tank (**T2**) to the heat exchanger (**1**) on the secondary heat source side is provided between the connecting point of a connecting pipe (**17b**) to the second branch pipe (**7b**) of the liquid pipe (**7**) and the heat exchanger (**1**) on the secondary heat source side. A check valve (**CV4-B**) allowing only the liquid refrigerant to flow from the indoor heat exchangers (**3, 3**) to the second tank (**T2**) is provided between the connecting point of the connecting pipe (**17b**) to the second branch pipe (**7b**) and the indoor heat exchangers (**3, 3**).

Of the two connecting points (X, Y) of each of the connecting pipes (17a, 17b), a fourth solenoid valve (SV-4) is provided between the connecting point (X) closer to the heat exchanger (1) on the secondary heat source side and the heat exchanger (1) on the secondary heat source side, and a fifth solenoid valve (SV-5) is provided between the connecting point (Y) closer to the indoor heat exchangers (3, 3, 3) and the indoor heat exchangers (3, 3, 3).

One end of a liquid refrigerant recovery pipe (LR1) is connected between the heat exchanger (1) on the secondary heat source side and the fourth solenoid valve (SV-4). The liquid refrigerant recovery pipe (LR1) is connected to the upstream of the check valve (CV4-B) at the other end, and is provided with a sixth solenoid valve (SV-6). One end of a liquid refrigerant supply pipe (LS1) is connected between the indoor heat exchangers (3, 3, 3) and the fifth solenoid valve (SV-5). The liquid refrigerant supply pipe (LS1) is connected to the downstream of the check valve (CV3-A) at the other end, and is provided with a seventh solenoid valve (SV-7).

A pressurizing/pressure-reducing mechanism (18) is connected to the respective upper ends of the tanks (T1, T2) through a first and a second pressurizing/pressure-reducing pipes (19d, 19e). The pressurizing/pressure-reducing mechanism (18) is constructed such that while a high pressure is applied to one tank (T1), a low pressure is applied to the other tank

(T2) and alternately switches these states. Specifically, a first and a second driving heat exchanger (E1, E2) are connected to the first and the second pressurizing/pressure-reducing pipes (19d, 19e). These driving heat exchangers (E1, E2) exchange heat with the heating and the cooling heat exchangers (D3, D5), thereby applying pressure for circulating the refrigerant onto the respective tanks (T1, T2).

That is to say, the first pressurizing/pressure-reducing pipe (19d) connected to the upper end of the first tank (T1) is branched into a first branch pipe (19d-A) leading to the first driving heat exchanger (E1) and a second branch pipe (19d-B) leading to the second driving heat exchanger (E2). An eighth solenoid valve (SV-8) is provided for the first branch pipe (19d-A). A check valve (CV-5) allowing only the refrigerant to flow from the first tank (T1) to the second driving heat exchanger (E2) and a ninth solenoid valve (SV-9) are provided for the second branch pipe (19d-B).

On the other hand, the second pressurizing/pressure-reducing pipe (19e) connected to the upper end of the second tank (T2) is also branched into two. One of them or the first branch pipe (19e-A) is connected between the eighth solenoid valve (SV-8) and the first driving heat exchanger (E1) of the first branch pipe (19d-A) via a tenth solenoid valve (SV-10). The other of them or the second branch pipe (19e-B) is connected between the ninth solenoid valve (SV-9) and the second

driving heat exchanger (**E2**) of the second branch pipe (**19d-B**) through a check valve (**CV-6**).

In addition, the secondary refrigerant circuit (**B**) further includes a pair of reservoirs (**20, 21**). A pipe connected to the upper end of the first reservoir (**20**) is branched into two. One of them is connected between the check valve (**CV-5**) and the ninth solenoid valve (**SV-9**) of the second branch pipe (**19d-B**), and the other is connected between the eighth solenoid valve (**SV-8**) and the first driving heat exchanger (**E1**) of the first branch pipe (**19d-A**) via an eleventh solenoid valve (**SV-11**). A pipe connected to the lower end of the first reservoir (**20**) is also branched into two, which are connected to the first driving heat exchanger (**E1**) and the second driving heat exchanger (**E2**), respectively. Each of the branch pipes is provided with a check valve (**CV-8**) allowing only the refrigerant to flow from the reservoir (**20**) into the first driving heat exchanger (**E1**) and a check valve (**CV-7**) allowing only the refrigerant to flow from the second driving heat exchanger (**E2**) into the reservoir (**20**).

A pipe connected to the upper end of the second reservoir (**21**) is also branched into two. One of them is connected between the ninth solenoid valve (**SV-9**) of the second branch pipe (**19d-B**) and the second driving heat exchanger (**E2**) via a twelfth solenoid valve (**SV-12**), and the other is connected between the eighth solenoid valve (**SV-8**) of the first branch

pipe (**19d-A**) and the first driving heat exchanger (**E1**) via a thirteenth solenoid valve (**SV-13**). A pipe connected to the lower end of the second reservoir (**21**) is also branched into two, which are connected to the first driving heat exchanger (**E1**) and the second driving heat exchanger (**E2**), respectively. Each of the branch pipes is provided with the check valve (**CV-8**) allowing only the refrigerant to flow from the reservoir (**21**) into the first driving heat exchanger (**E1**) and the check valve (**CV-7**) allowing only the refrigerant to flow from the second driving heat exchanger (**E2**) into the reservoir (**21**).

A first by-pass pipe (**BPL-1**) is connected between the gas pipe (**6**) and the second branch pipe (**19d-B**). The first by-pass pipe (**BPL-1**) includes a fourteenth solenoid valve (**SV-14**), thereby by-passing a part of the gaseous refrigerant to the gas pipe (**6**). A second by-pass pipe (**BPL-2**) is connected between the gas pipe (**6**) and the second pressurizing/pressure-reducing pipe (**19e**). The second by-pass pipe (**BPL-2**) includes a fifteenth solenoid valve (**SV-15**), thereby by-passing a part of the gaseous refrigerant to the gas pipe (**6**). A third by-pass pipe (**BPL-3**) is connected between the liquid pipe (**7**) and the respective driving heat exchangers (**E1, E2**). The third by-pass pipe (**BPL-3**) includes a check valve (**CV-9**), thereby by-passing the liquid refrigerant to the respective driving heat exchangers (**E1, E2**).

Next, the cooling running operation will be described.

In the primary refrigerant circuit (**A**), the four-position selector valve (**D2**) is switched to the direction indicated by the solid lines, the first solenoid valve (**SV-1**) is opened and the second and the third solenoid valves (**SV-2**, **SV-3**) are closed. Then, as indicated by the solid-line arrows in Figure **20**, the refrigerant discharged from the compressor (**11**) is condensed in the outdoor heat exchanger (**14**). Thereafter, a part of the refrigerant is supplied to the heating heat exchanger (**D3**) and the other part is supplied to the cooling heat exchanger (**D5**) after the pressure thereof has been reduced in the pressure reducing valve (**D4**). The refrigerant supplied to the heating heat exchanger (**D3**) applies heat to the first driving heat exchanger (**E1**) and is excessively cooled. Then, in the heating heat exchanger (**12**), the refrigerant extracts heat from the refrigerant in the heat exchanger (**1**) on the secondary heat source side and is evaporated to return to the compressor (**11**). On the other hand, the refrigerant supplied to the cooling heat exchanger (**D5**) extracts heat from the second driving heat exchanger (**E2**) and is evaporated to return to the compressor (**11**).

As a result of the refrigerant circulation operation, the refrigerant is evaporated and a high pressure is built up in the first driving heat exchanger (**E1**). On the other hand, the refrigerant is condensed and a low pressure is built up in the second driving heat exchanger (**E2**). In such a state,

the sixth, seventh, eighth, eleventh and twelfth solenoid valves (**SV-6, SV-7, SV-8, SV-11, SV-12**) are opened and the fourth, fifth, ninth, tenth and thirteenth solenoid valves (**SV-4, SV-5, SV-9, SV-10, SV-13**) are closed.

As a result, a high pressure is applied to the first tank (**T1**) and a low pressure is applied to the second tank (**T2**), and the operation of draining the liquid refrigerant from the first tank (**T1**) and the operation of recovering the liquid refrigerant into the second tank (**T2**) are simultaneously performed, thereby circulating the refrigerant in the secondary refrigerant circuit (**B**) as indicated by the broken-line arrows in Figure **20**. The gaseous refrigerant, which has been recovered from the second tank (**T2**) into the second driving heat exchanger (**E2**), is condensed and reserved in the second reservoir (**21**). The pressure of the first reservoir (**20**) is equalized with that of the first driving heat exchanger (**E1**), thereby supplying the liquid refrigerant to the first driving heat exchanger (**E1**).

After this refrigerant circulation operation has been performed for a predetermined time, the respective solenoid valves are switched, thereby opening the sixth, seventh, ninth, tenth and thirteenth solenoid valves (**SV-6, SV-7, SV-9, SV-10, SV-13**) and closing the fourth, fifth, eighth, eleventh and twelfth solenoid valves (**SV-4, SV-5, SV-8, SV-11, SV-12**).

As a result, a high pressure is applied to the second tank (**T2**) and a low pressure is applied to the first tank (**T1**), and the operation of draining the liquid refrigerant from the second tank (**T2**) and the operation of recovering the liquid refrigerant into the first tank (**T1**) are simultaneously performed, thereby circulating the refrigerant in the secondary refrigerant circuit (**B**) as indicated by the one-dot-chain arrows in Figure **20**. The gaseous refrigerant, which has been recovered from the first tank (**T1**) into the second driving heat exchanger (**E2**), is condensed and reserved in the first reservoir (**20**). The pressure of the second reservoir (**21**) is equalized with that of the first driving heat exchanger (**E1**), thereby supplying the liquid refrigerant to the first driving heat exchanger (**E1**).

In such a manner, a tank draining the liquid refrigerant and a tank recovering the liquid refrigerant are alternately switched, thereby continuously performing the room cooling running.

Moreover, since the liquid refrigerant condensed in the second driving heat exchanger (**E2**) is recovered into the reservoirs (**20, 21**), a large heat exchange area can be secured for the second driving heat exchanger (**E2**) and the quantity of heat exchanged with the cooling heat exchanger (**D5**) is increased. As a result, the performance of the entire system can be improved.

On the other hand, during the room heating running, the four-position selector valve (**D2**) is switched to the direction indicated by the broken lines, the first solenoid valve (**SV-1**) is closed and the second and the third solenoid valves (**SV-2, SV-3**) are opened in the primary refrigerant circuit (**A**). Then, as indicated by the solid-line arrows in Figure **21**, a part of the refrigerant discharged from the compressor (**11**) is supplied to the heat exchanger (**12**) on the primary heat source side and the other refrigerant is supplied to the heating heat exchanger (**D3**).

The refrigerant supplied to the heat exchanger (**12**) on the primary heat source side exchanges heat with the heat exchanger (**1**) on the secondary heat source side and is condensed. Thereafter, a part of the refrigerant is supplied to the outdoor heat exchanger (**14**) and the other part is supplied to the cooling heat exchanger (**D5**). The refrigerant supplied to the outdoor heat exchanger (**14**) exchanges heat with the outdoor air, is evaporated and then is recovered into the compressor (**11**). The refrigerant supplied to the cooling heat exchanger (**D5**) extracts heat from the second driving heat exchanger (**E2**), is evaporated and then returns to the compressor (**11**).

On the other hand, the refrigerant supplied to the heating heat exchanger (**D3**) applies heat to the first driving heat exchanger (**E1**) and is condensed. Thereafter, in the

heat exchanger (12) on the primary heat source side, the refrigerant applies heat to the refrigerant in the heat exchanger (1) on the secondary heat source side and is excessively cooled. Then, the refrigerant is evaporated in the outdoor heat exchanger (14) and the cooling heat exchanger (D5) so as to return to the compressor (11).

As a result of the refrigerant circulation operation, the refrigerant is evaporated and a high pressure is built up in the first driving heat exchanger (E1). On the other hand, the refrigerant is condensed and a low pressure is built up in the second driving heat exchanger (E2). In such a state, the fourth, fifth, eighth, eleventh and twelfth solenoid valves (SV-4, SV-5, SV-8, SV-11, SV-12) are opened and the sixth, seventh, ninth, tenth and thirteenth solenoid valves (SV-6, SV-7, SV-9, SV-10, SV-13) are closed. As a result, a high pressure is applied to the first tank (T1) and a low pressure is applied to the second tank (T2), and the operation of draining the liquid refrigerant from the first tank (T1) and the operation of recovering the liquid refrigerant into the second tank (T2) are simultaneously performed, thereby circulating the refrigerant in the secondary refrigerant circuit (B) as indicated by the broken-line arrows in Figure 21.

After this refrigerant circulation operation has been performed for a predetermined time, the respective solenoid

valves are switched, thereby opening the fourth, fifth, ninth, tenth and thirteenth solenoid valves (**SV-4, SV-5, SV-9, SV-10, SV-13**) and closing the sixth, seventh, eighth, eleventh and twelfth solenoid valves (**SV-6, SV-7, SV-8, SV-9, SV-11, SV-12**). As a result, a high pressure is applied to the second tank (**T2**) and a low pressure is applied to the first tank (**T1**), and the operation of draining the liquid refrigerant from the second tank (**T2**) and the operation of recovering the liquid refrigerant into the first tank (**T1**) are simultaneously performed, thereby circulating the refrigerant in the secondary refrigerant circuit (**B**) as indicated by the one-dot-chain arrows in Figure **21**.

A tank draining the liquid refrigerant and a tank recovering the liquid refrigerant are alternately switched, thereby continuously performing the room heating running. During this heating running, the quantity of heat exchanged between the second driving heat exchanger (**E2**) and the cooling heat exchanger (**D5**) can be increased and the performance of the entire system can be improved by recovering the liquid refrigerant into the reservoirs (**20, 21**).

Thus, in this embodiment, since the liquid refrigerant condensed in the outdoor heat exchanger (**14**) can be cooled in the heating heat exchanger (**D3**) to reach an excessively cooled state during the room cooling running, a large quanti-

primary heat source side and the heat exchanger (1) on the secondary heat source side can be secured, and the performance of the entire system can be improved.

(Eleventh Embodiment)

Next, the eleventh embodiment of the present invention will be described with reference to Figure 22.

This embodiment provides a pressurizing circuit (50) and a pressure reducing circuit (60) for a pressurizing/pressure-reducing mechanism (18) and is applied to an air conditioning system exclusively used for cooling.

First, the secondary refrigerant circuit (B) will be described.

In the secondary refrigerant circuit (B), an indoor heat exchanger (3) and a heat exchanger (1) on the secondary heat source side are connected to each other through a gas pipe (6) and a liquid pipe (7).

A tank (T) is connected to the liquid pipe (7). A first check valve (CV1) allowing only the liquid refrigerant to flow from the heat exchanger (1) on the secondary heat source side to the tank (T) is provided between the tank (T) and the heat exchanger (1) on the secondary heat source side in the liquid pipe (7). A second check valve (CV2) allowing only the liquid refrigerant to flow from the tank (T) to the indoor heat exchanger (3) is provided between the tank (T) and the indoor

heat exchanger (3) in the liquid pipe (7). An indoor electrically motorized expansion valve (EV1) is further provided between the second check valve (CV2) and the indoor heat exchanger (3) in the liquid pipe (7).

The pressurizing circuit (50) and the pressure reducing circuit (60) are connected to the tank (T). First, the pressurizing circuit (50) will be described.

The pressurizing circuit (50) includes a circulating evaporator (51). The circulating evaporator (51) is disposed at a position lower than the installation position of the tank (T). The circulating evaporator (51) is connected to the upper part of the tank (T) through a gas supply pipe (52) and to the lower part of the tank (T) through a liquid recovery pipe (53).

A first solenoid valve (SV1), which is opened in applying a high pressure to the inside of the tank (T), is provided for the gas supply pipe (52). A third check valve (CV3) allowing only the refrigerant to flow from the tank (T) into the circulating evaporator (51) is provided for the liquid recovery pipe (53).

Next, the pressure reducing circuit (60) will be described.

The pressure reducing circuit (60) includes a circulating condenser (61). The circulating condenser (61) is disposed at a position higher than the installation position of

the tank (T). The circulating condenser (61) is connected to the upper part of the tank (T) through a gas recovery pipe (62) and to the lower part of the tank (T) through a liquid supply pipe (63).

A second solenoid valve (SV2), which is opened in applying a low pressure to the inside of the tank (T), is provided for the gas recovery pipe (62). A fourth check valve (CV4) allowing only the refrigerant to flow from the circulating condenser (61) to the tank (T) is provided for the liquid supply pipe (63).

Next, the primary refrigerant circuit (A) for exchanging heat with the secondary refrigerant circuit (B) will be described.

The primary refrigerant circuit (A) is constituted by connecting: a compressor (11); an outdoor heat exchanger (14) for exchanging heat with the outdoor air; a heating heat exchanger (71) that can exchange heat with the circulating evaporator (51); a cooling heat exchanger (72) that can exchange heat with the circulating condenser (61); and a heat exchanger (12) on the primary heat source side that can exchange heat with the heat exchanger (1) on the secondary heat source side to each other through a refrigerant pipe (16).

Specifically, the outdoor heat exchanger (14) and the heating heat exchanger (71) are connected in this order to the outlet side of the compressor (11). The liquid side of

the heating heat exchanger (**71**) is branched into a first branch pipe (**16a**) and a second branch pipe (**16b**). The first branch pipe (**16a**) is connected to the cooling heat exchanger (**72**) and the second branch pipe (**16b**) is connected to the heat exchanger (**12**) on the primary heat source side. A first outdoor electrically motorized expansion valve (**EV-A**) and a second outdoor electrically motorized expansion valve (**EV-B**) are provided for the first branch pipe (**16a**) and the second branch pipe (**16b**), respectively. The gas sides of the cooling heat exchanger (**72**) and the heat exchanger (**12**) on the primary heat source side are combined with each other and connected to the inlet side of the compressor (**11**).

In this embodiment, the condensing temperature in the circulating condenser (**61**) is set lower than the condensing temperature in the heat exchanger (**1**) on the secondary heat source side. Specifically, the first branch pipe (**16a**) and the second branch pipe (**16b**) have respectively different pipe diameters, and the flow rate of the first branch pipe (**16a**) is set smaller than the flow rate of the second branch pipe (**16b**) by a predetermined ratio. On the other hand, the heat exchange area between the cooling heat exchanger (**72**) and the circulating condenser (**61**) is set smaller than the heat exchange area between the heat exchanger (**12**) on the primary heat source side and the heat exchanger (**1**) on the secondary heat source side, and the ratio is set smaller than the above-

described predetermined ratio.

Specifically, for example, if the ratio of the flow rate of the first branch pipe (**16a**) to the flow rate of the second branch pipe (**16b**) is 1:10, then the ratio of the heat exchange area between the cooling heat exchanger (**72**) and the circulating condenser (**61**) to the heat exchange area between the heat exchanger (**12**) on the primary heat source side and the heat exchanger (**1**) on the secondary heat source side is set at 2:10. Thus, in respect of the power as a heat exchanger with respect to the flow rate of a refrigerant, the circulating condenser (**61**) is made superior to the heat exchanger (**1**) on the secondary heat source side. As a result, the condensing temperature of the circulating condenser (**61**) becomes lower than the condensing temperature of the heat exchanger (**1**) on the secondary heat source side.

Next, the room cooling running operation in this embodiment will be described.

During the cooling running, the compressor (**11**) is driven in the primary refrigerant circuit (**A**). As indicated by the solid-line arrows in Figure **22**, a high-temperature, high-pressure gaseous refrigerant discharged from the compressor (**11**) sequentially flows through the outdoor heat exchanger (**14**) and the heating heat exchanger (**71**), exchanges heat with the outdoor air and the refrigerant in the circulating evaporator (**51**) and is condensed, thereby applying

heat to the refrigerant in the circulating evaporator (51). Thereafter, the liquid refrigerant is distributed into the respective branch pipes (16a, 16b) and the pressure of the refrigerant is reduced in the respective outdoor electrically motorized expansion valves (EV-A, EV-B). Then, the refrigerant flows through the cooling heat exchanger (72) and the heat exchanger (12) on the primary heat source side. Here, the liquid refrigerant exchanges heat with the refrigerant in the circulating condenser (61) and with the refrigerant in the heat exchanger (1) on the secondary heat source side and is evaporated, thereby extracting heat from the refrigerant in the circulating condenser (61) and the refrigerant in the heat exchanger (1) on the secondary heat source side. Thereafter, the gaseous refrigerants, which have flowed through the cooling heat exchanger (72) and the heat exchanger (12) on the primary heat source side, are combined to return to the compressor (11). This circulation operation is repeated.

On the other hand, in the secondary refrigerant circuit (B), a refrigerant evaporation operation and a refrigerant condensation operation are caused in the circulating evaporator (51) and in the circulating condenser (61), respectively, as a result of the above-described heat exchange operation. And, a high pressure and a low pressure are respectively generated in the circulating evaporator (51) and in the circulating condenser (61).

In such a state, first, the first solenoid valve (**SV1**) is opened and the second solenoid valve (**SV2**) is closed. As a result, the high pressure in the circulating evaporator (**51**) is applied to the inside of the tank (**T**) through the gas supply pipe (**52**), the water level of the liquid refrigerant in the tank (**T**) is pushed down and the liquid refrigerant is pushed out into the liquid pipe (**7**) as indicated by the broken-line arrows in Figure **22**. The pushed-out liquid refrigerant flows through the liquid pipe (**7**) toward the indoor heat exchanger (**3**). The pressure of the liquid refrigerant is reduced in the indoor electrically motorized expansion valve (**EV1**). Then, in the indoor heat exchanger (**3**), the liquid refrigerant exchanges heat with the indoor air, and is evaporated, thereby cooling the indoor air. The evaporated gaseous refrigerant flows through the gas pipe (**6**) to the heat exchanger (**1**) on the secondary heat source side, exchanges heat with the heat exchanger (**12**) on the primary heat source side, and is condensed.

After this operation has been performed, the first solenoid valve (**SV1**) is closed and the second solenoid valve (**SV2**) is opened. As a result, the low pressure in the circulating condenser (**61**) is applied to the inside of the tank (**T**) through the gas recovery pipe (**62**). The condensing temperature of the circulating condenser (**61**) is lower than the condensing temperature of the heat exchanger (**1**) on the sec-

ondary heat source side and the internal pressure of the circulating condenser (61) is lower than the internal pressure of the heat exchanger (1) on the secondary heat source side. Thus, the internal pressure of the tank (T) becomes lower than the internal pressure of the heat exchanger (1) on the secondary heat source side. As indicated by the one-dot-chain arrows in Figure 22, the liquid refrigerant in the heat exchanger (1) on the secondary heat source side is recovered into the tank (T) through the liquid pipe (7).

In this case, the gaseous refrigerant in the upper part in the tank (T) is sucked into the circulating condenser (61) and then condensed to be a liquid refrigerant, which is recovered into the tank (T) through the liquid supply pipe (63). When the pressurizing operation is started by the above-described pressurizing circuit (50) from such a state, the pressure in the entire pressurizing circuit (50) is equalized. As a result, a part of the liquid refrigerant in the tank (T) is recovered into the circulating evaporator (51) and is used as a refrigerant for generating a high pressure.

By alternately repeating the pressurizing operation by the pressurizing circuit (50) and the pressure reducing operation by the pressure reducing circuit (60), the liquid refrigerant is pushed out from the tank (T) during the pressurizing operation, recovered into the tank (T) during the

pressure reducing operation and circulated in the secondary refrigerant circuit (B), thereby cooling the indoor air.

Thus, in this embodiment, since the liquid refrigerant in the heat exchanger (1) on the secondary heat source side is recovered by utilizing the suction force generated in the tank (T), it is no longer necessary to dispose the tank (T) at a position lower than that of the heat exchanger (1) on the secondary heat source side. As a result, the restriction on the installation positions of units can be alleviated and the universality can be improved.

(Twelfth Embodiment)

Next, the twelfth embodiment of the present invention will be described. It is noted that only the difference from the foregoing eleventh embodiment will be described hereinafter.

The refrigerant circuit of this embodiment is applied to an air conditioning system exclusively used for heating. The arrangement of the primary refrigerant circuit (A) and the arrangement of a check valve provided for the liquid pipe (7) are different from the counterparts of the eleventh embodiment described above.

Specifically, as shown in Figure 23, the heating heat exchanger (71) and the heat exchanger (12) on the primary heat source side are connected in this order to the outlet

side of the compressor (11) in the primary refrigerant circuit (A). The liquid side of the heat exchanger (12) on the primary heat source side is branched into a first branch pipe (16c) and a second branch pipe (16d). The first branch pipe (16c) is connected to the outdoor heat exchanger (14) and the second branch pipe (16d) is connected to the cooling heat exchanger (72). A first outdoor electrically motorized expansion valve (EV-C) and a second outdoor electrically motorized expansion valve (EV-D) are provided for the first branch pipe (16c) and the second branch pipe (16d), respectively. The gas sides of the outdoor heat exchanger (14) and the cooling heat exchanger (72) are combined with each other and connected to the inlet side of the compressor (11).

On the other hand, a first check valve (CV3) allowing only the liquid refrigerant to flow from the tank (T) into the heat exchanger (1) on the secondary heat source side is provided between the tank (T) and the heat exchanger (1) on the secondary heat source side in the liquid pipe (7) in the secondary refrigerant circuit (B). A second check valve (CV4) allowing only the liquid refrigerant to flow from the indoor heat exchanger (3) into the tank (T) is provided between the tank (T) and the indoor heat exchanger (3) in the liquid pipe (7).

It is noted that the pressurizing circuit (50) and the pressure reducing circuit (60) of this embodiment are the same

as those of the eleventh embodiment described above.

By serially connecting the heating heat exchanger (71) for exchanging heat with the circulating evaporator (51) to the heat exchanger (12) on the primary heat source side for exchanging heat with the heat exchanger (1) on the secondary heat source side in this manner, the evaporating temperature in the circulating evaporator (51) becomes higher than the evaporating temperature in the heat exchanger (1) on the secondary heat source side. That is to say, owing to the difference in evaporating temperatures, the internal pressure of the circulating evaporator (51) becomes higher than the internal pressure of the heat exchanger (1) on the secondary heat source side.

Next, the room heating running operation in this embodiment will be described.

During the heating running, the compressor (11) is driven in the primary refrigerant circuit (A). As indicated by the solid-line arrows in Figure 23, a high-temperature, high-pressure gaseous refrigerant discharged from the compressor (11) sequentially flows through the heating heat exchanger (71) and the heat exchanger (12) on the primary heat source side, exchanges heat with the refrigerant in the circulating evaporator (51) and the refrigerant in the heat exchanger (1) on the secondary heat source side, and is condensed, thereby applying heat to the refrigerant in the

circulating evaporator (51) and the refrigerant in the heat exchanger (1) on the secondary heat source side. Thereafter, the liquid refrigerant is distributed into the respective branch pipes (16c, 16d) and the pressure of the refrigerant is reduced in the respective outdoor electrically motorized expansion valves (EV-C, EV-D). Then, the refrigerant flows through the cooling heat exchanger (72) and the outdoor heat exchanger (14). The liquid refrigerant exchanges heat with the circulating condenser (61) in the cooling heat exchanger (72) and with the outdoor air in the outdoor heat exchanger (14) and is evaporated, thereby extracting heat from the refrigerant in the circulating condenser (61). Thereafter, the gaseous refrigerants, which have flowed through the cooling heat exchanger (72) and the outdoor heat exchanger (14), are combined to return to the compressor (11). This circulation operation is repeated.

On the other hand, in the secondary refrigerant circuit (B), the switching operations of the respective solenoid valves (SV1, SV2) are performed in the pressurizing circuit (50) and the pressure reducing circuit (60) in the same way as in the eleventh embodiment described above, thereby alternately switching a state where a high pressure is applied to the inside of the tank (T) and a state where a low pressure is applied to the inside of the tank (T). In the state where a high pressure is applied to the tank (T), since the evapo-

rating temperature of the circulating evaporator (51) is higher than the evaporating temperature of the heat exchanger (1) on the secondary heat source side as described above, the internal pressure of the circulating evaporator (51) becomes higher than the internal pressure of the heat exchanger (1) on the secondary heat source side. Thus, a high pressure is built up in the tank (T). As indicated by the broken-line arrows in Figure 23, the liquid refrigerant pushed out from the tank (T) exchanges heat with the heat exchanger (12) on the primary heat source side and is evaporated in the heat exchanger (1) on the secondary heat source side. The evaporated gaseous refrigerant flows into the indoor heat exchanger (3) through the gas pipe (6), exchanges heat with the indoor air and is condensed, thereby heating the indoor air.

In the state where a low pressure is applied to the inside of the tank (T), the liquid refrigerant in the indoor heat exchanger (3) is recovered into the tank (T) through the liquid pipe (7) as indicated by the one-dot-chain arrows in Figure 23. In this operation, during the pressure reducing operation of the pressure reducing circuit (60), the gaseous refrigerant sucked into the circulating condenser (61) is condensed to be a liquid refrigerant, which is recovered into the tank (T) through the liquid supply pipe (63). On the other hand, during the pressurizing operation of the pressurizing circuit (50), a part of the liquid refrigerant in the

tank (T) is recovered into the circulating evaporator (51) and is used as a refrigerant for generating a high pressure. This operation is repeated, thereby heating the indoor air.

Thus, in this embodiment, since the liquid refrigerant in the indoor heat exchanger (3) is recovered by utilizing the suction force generated in the tank (T), it is no longer necessary to dispose the tank (T) at a position lower than that of the indoor heat exchanger (3). As a result, the restriction on the installation positions of units can be alleviated and the universality can be improved.

(Thirteenth Embodiment)

Next, the thirteenth embodiment of the present invention will be described. It is noted that only the difference from the foregoing eleventh embodiment will also be described hereinafter.

The refrigerant circuit of this embodiment is applied to an air conditioning system exclusively used for cooling. The primary refrigerant circuit (A) is the same as that of the eleventh embodiment described above. Thus, the description thereof will be omitted herein.

And, the secondary refrigerant circuit (B) of this embodiment is characterized in that two tanks (T1, T2) are provided in the same way as in the fifth embodiment described above and that the respective tanks (T1, T2) are connected in

parallel to the pressurizing circuit (**50**) and the pressure reducing circuit (**60**).

Specifically, as shown in Figure **24**, the gas supply pipe (**52**) of the pressurizing circuit (**50**) is branched into branch pipes (**52a, 52b**), which are connected to the upper ends of the respective tanks (**T1, T2**) and are provided with solenoid valves (**SV1-1, SV1-2**), respectively. The liquid recovery pipe (**53**) of the pressurizing circuit (**50**) is branched into branch pipes (**53a, 53b**), which are connected to the lower ends of the respective tanks (**T1, T2**) and are provided with check valves (**CV3-1, CV3-2**), respectively.

On the other hand, the gas recovery pipe (**62**) of the pressure reducing circuit (**60**) is branched into branch pipes (**62a, 62b**), which are connected to the upper ends of the respective tanks (**T1, T2**) and are provided with solenoid valves (**SV2-1, SV2-2**), respectively. The liquid supply pipe (**63**) of the pressure reducing circuit (**60**) is branched into branch pipes (**63a, 63b**), which are connected to the lower ends of the respective tanks (**T1, T2**) and are provided with check valves (**CV4-1, CV4-2**), respectively.

The liquid pipe (**7**) coupled to the heat exchanger (**1**) on the secondary heat source side is branched into branched liquid pipes (**7a, 7b**), which are connected to the lower ends of the respective tanks (**T1, T2**) and are provided with check valves (**CV1-1, CV1-2**). The liquid pipe (**7**) coupled to the

indoor heat exchanger (3) is branched into branched liquid pipes (7c, 7d), which are connected to the lower ends of the respective tanks (T1, T2) and are provided with check valves (CV2-1, CV2-2).

In this embodiment, while the secondary refrigerant circuit (B) is operated during the cooling running, the operations of opening one of the solenoid valves (SV1-1, SV1-2) provided for the respective branch pipes (52a, 52b) of the pressurizing circuit (50) and closing the other are alternately repeated. In addition, the operations of opening one of the solenoid valves (SV2-1, SV2-2) provided for the respective branch pipes (62a, 62b) of the pressure reducing circuit (60) and closing the other are also alternately repeated. In such a manner, the operations of making one tank push out the liquid refrigerant toward the indoor heat exchanger (3) and making the other tank recover the liquid refrigerant from the heat exchanger (1) on the secondary heat source side are repeated.

Specifically, in the case where the solenoid valve (SV1-1) of the branch pipe (52a) in the pressurizing circuit (50) is opened and the solenoid valve (SV2-2) of the branch pipe (62a) in the pressure reducing circuit (60) is also opened, the upper tank (T1) pushes out the liquid refrigerant toward the indoor heat exchanger (3) and the lower tank (T2) recovers the liquid refrigerant from the heat exchanger (1) on the

secondary heat source side as indicated by the solid lines in Figure **24**. Conversely, in the case where the solenoid valve (**SV1-2**) of the branch pipe (**52b**) in the pressurizing circuit (**50**) is opened and the solenoid valve (**SV2-1**) of the branch pipe (**62a**) in the pressure reducing circuit (**60**) is also opened, the lower tank (**T2**) pushes out the liquid refrigerant toward the indoor heat exchanger (**3**) and the upper tank (**T1**) recovers the liquid refrigerant from the heat exchanger (**1**) on the secondary heat source side as indicated by the broken-line arrows in Figure **24**. By alternately repeating these operations, the room cooling is performed continuously.

(Fourteenth Embodiment)

Next, the fourteenth embodiment of the present invention will be described. It is noted that only the difference from the foregoing thirteenth embodiment will be described hereinafter.

The refrigerant circuit of this embodiment is applied to an air conditioning system exclusively used for heating. The primary refrigerant circuit (**A**) is the same as that of the thirteenth embodiment described above. Thus, the description thereof will be omitted herein. Also, the secondary refrigerant circuit (**B**) is different from that of the thirteenth embodiment in the arrangement of the check valve provided for the liquid pipe (**7**).

Specifically, as shown in Figure **25**, check valves allowing the refrigerant to flow in different directions are employed as the check valves (**CV1-1**, **CV1-2**, **CV2-1**, **CV2-2**) provided for the liquid pipe (**7**).

Thus, during the heating running, the opening/closing operations of the solenoid valves (**SV1-1**, **SV1-2**) provided for the respective branch pipes (**52a**, **52b**) of the pressurizing circuit (**50**) and the solenoid valves (**SV2-1**, **SV2-2**) provided for the respective branch pipes (**62a**, **62b**) of the pressure reducing circuit (**60**) are alternately repeated in the same way as in the thirteenth embodiment described above. As a result, the operations of making one tank push out the liquid refrigerant toward the heat exchanger (**1**) on the secondary heat source side and making the other tank recover the liquid refrigerant from the indoor heat exchanger (**3**) are alternately repeated (i.e., the state indicated by the solid-line arrows in Figure **25** and the state indicated by the broken-line arrows therein are alternately repeated) whereby the room heating is performed continuously.

(Fifteenth Embodiment)

Next, the fifteenth embodiment of the present invention will be described. It is noted that only the difference from the foregoing eleventh embodiment will be described hereinafter.

The refrigerant circuit of this embodiment is applied to an air conditioning system exclusively used for cooling. The primary refrigerant circuit (**A**) is the same as that of the eleventh embodiment described above. Thus, the description thereof will be omitted herein.

The secondary refrigerant circuit (**B**) of this embodiment is characterized by including a sub-tank (**ST**) of a small size, separately from the above-described tank (**T**) (hereinafter, referred to as a main tank) as shown in Figure **26**, and by temporarily reserving a liquid refrigerant in the sub-tank (**ST**).

Hereinafter, the circuit configuration of the secondary refrigerant circuit (**B**) of this embodiment will be described.

A gas recovery pipe (**62**) coupled to the circulating condenser (**61**) is branched into branch pipes (**62a, 62b**), one of which is connected to the upper end of the main tank (**T**) via the second solenoid valve (**SV2**) and the other of which is connected to the upper end of the sub-tank (**ST**) via the third solenoid valve (**SV3**).

A gas supply pipe (**52**) coupled to the circulating evaporator (**51**) is branched into branch pipes (**52a, 52b**), one of which is connected to the upper end of the main tank (**T**) via the first solenoid valve (**SV1**) and the other of which is connected to the branch pipe (**62b**) via the fourth solenoid valve (**SV4**).

The sub-tank (**ST**) is disposed at a position higher than that of the circulating evaporator (**51**). The lower end of the circulating evaporator (**51**) and the lower part of the sub-tank (**ST**) are connected to each other through a liquid recovery pipe (**53**). A check valve (**CV3**) allowing only the refrigerant to flow from the sub-tank (**ST**) into the circulating evaporator (**51**) is provided for the liquid recovery pipe (**53**).

The liquid recovery pipe (**53**) and the liquid pipe (**7**) are connected to each other through a liquid suction pipe (**54**) provided with a check valve (**CV5**) allowing only the liquid refrigerant to flow from the liquid pipe (**7**) into liquid recovery pipe (**53**). One end of the liquid suction pipe (**54**) is connected between the sub-tank (**ST**) and the check valve (**CV3**) in the liquid recovery pipe (**53**) and the other end thereof is connected between the indoor electrically motorized expansion valve (**EV1**) and the check valve (**CV2**) in the liquid pipe (**7**). Selector means (**I**) for switching the pressure application states to the sub-tank (**ST**) is constituted in this manner. The other arrangements are the same as those the eleventh embodiment described above.

Next, the room cooling running of this embodiment will be described.

In the primary refrigerant circuit (**A**), the same operations as those of the eleventh embodiment described above are

performed.

In the secondary refrigerant circuit (**B**), first, the first and the fourth solenoid valves (**SV1**, **SV4**) are opened and the second and the third solenoid valves (**SV2**, **SV3**) are closed. As a result, the high pressure in the circulating evaporator (**51**) is applied to the main tank (**T**) and the pressure in the circulating evaporator (**51**) is equalized with the pressure in the sub-tank (**ST**). Consequently, as indicated by the solid-line arrows in Figure **26**, the liquid refrigerant pushed out from the main tank (**T**) is evaporated in the indoor heat exchanger (**3**) and then condensed in the heat exchanger (**1**) on the secondary heat source side. In addition, the liquid refrigerant in the sub-tank (**ST**) is dripped and supplied to the circulating evaporator (**51**) through the liquid recovery pipe (**53**).

Thereafter, the opening/closing operations of the solenoid valves are switched and the first and the fourth solenoid valves (**SV1**, **SV4**) are closed while the second and the third solenoid valves (**SV2**, **SV3**) are opened. As a result, the low pressure in the circulating condenser (**61**) is applied to the main tank (**T**) and the sub-tank (**ST**) through the respective branch pipes (**62a**, **62b**) of the gas recovery pipe (**62**). Consequently, as indicated by the broken-line arrows in Figure **26**, the liquid refrigerant in the heat exchanger (**1**) on the secondary heat source side is recovered into the main tank (**T**)

through the liquid pipe (7). Also, since a low pressure state is established in the sub-tank (ST), a part of the liquid refrigerant in the liquid pipe (7) is recovered into the sub-tank (ST) through the liquid suction pipe (54). When the solenoid valves are switched again and the pressure in the sub-tank (ST) is equalized with the pressure in the circulating evaporator (51), the liquid refrigerant recovered into the sub-tank (ST) is supplied to the circulating evaporator (51) and functions as a refrigerant for driving. This operation is repeated, thereby cooling the indoor air.

Thus, since the liquid refrigerant is reserved in the sub-tank (ST) and is supplied to the circulating evaporator (51) in this embodiment, it is no longer necessary to dispose the main tank (T) above the circulating evaporator (51) unlike the foregoing embodiments. Thus, the flexibility in the installation positions of the main tank (T) and the circulating evaporator (51) can be increased.

It is noted that if a refrigerant circuit including such a sub-tank to an air conditioning system exclusively used for heating, check valves allowing the refrigerant to flow in different directions are employed as the check valves (CV1, CV2) provided for the liquid pipe (7).

(Sixteenth Embodiment)

Next, the sixteenth embodiment of the present invention

will be described. The secondary refrigerant circuit (**B**) of this embodiment is characterized by including two sub-tanks (**ST1, ST2**) similar to that of the fifteenth embodiment described above.

The circuit configuration thereof will be described. As shown in Figure **27**, the respective sub-tanks (**ST1, ST2**) are connected in parallel to each other in the circulating condenser (**61**) and the circulating evaporator (**51**). Specifically, the respective sub-tanks (**ST1, ST2**) are connected to the circulating condenser (**61**) through the gas recovery pipes (**62b-1, 62b-2**), respectively, and are connected to the circulating evaporator (**51**) through the gas supply pipes (**52b-1, 52b-2**), respectively. Solenoid valves (**SV3-A, SV3-B**) are provided for the gas recovery pipes (**62b-1, 62b-2**), respectively and solenoid valves (**SV4-A, SV4-B**) are provided for the gas supply pipes (**52b-1, 52b-2**), respectively. Liquid suction pipes (**54-1, 54-2**) are provided for the lower ends of the respective sub-tanks (**ST1, ST2**) so as to correspond to the liquid recovery pipes (**53-1, 53-2**) connected to the circulating evaporator (**51**).

Selector means (**I**) for switching the pressure application states to the respective sub-tanks (**ST1, ST2**) is constituted in such a manner. The other arrangements are the same as those of the eleventh embodiment described above. The other arrangements are substantially the same as those of the fifteen-

th embodiment described above.

Next, the room cooling running of this embodiment will be described.

In the primary refrigerant circuit (**A**), the same operations as those of the eleventh embodiment described above are performed.

In the secondary refrigerant circuit (**B**), states where one of the two sub-tanks (**ST1**, **ST2**) is connected to the circulating condenser (**61**) and the other is connected to the circulating evaporator (**51**) are alternately and repeatedly established. Specifically, one solenoid valve (**SV3-A**) of the gas recovery pipe (**62b**) is opened and the other solenoid valve (**SV3-B**) is closed while one solenoid valve (**SV4-B**) of the gas supply pipe (**52b**) is opened and the other solenoid valve (**SV4-A**) is closed. As a result, as indicated by the solid-line arrows in Figure **27**, a state, in which one sub-tank (**ST1**) communicates with the circulating condenser (**61**) to recover a part of the liquid refrigerant in the liquid pipe (**7**) and the other sub-tank (**ST2**) communicates with the circulating evaporator (**51**) to drip and supply the liquid refrigerant to the circulating evaporator (**51**), is established.

When these solenoid valves are switched, a state, in which the other sub-tank (**ST2**) recovers a part of the liquid refrigerant in the liquid pipe (**7**) and one sub-tank (**ST1**) drips and supplies the liquid refrigerant to the circulating

evaporator (**51**), is established as indicated by the broken-line arrows in Figure **27**. These states are alternately and repeatedly established.

Thus, since the operation of recovering the liquid refrigerant into one sub-tank and the operation of supplying the liquid refrigerant from the other sub-tank to the circulating evaporator (**51**) are performed simultaneously, the number of times by which the solenoid valves are opened/closed for switching the operations of the sub-tanks (**ST1**, **ST2**) can be reduced as compared with the case of providing only one sub-tank (**ST**) as in the fifteenth embodiment described above. As a result, the durability thereof can be improved.

It is noted that if this embodiment is applied to an air conditioning system exclusively used for heating, check valves allowing the refrigerant to flow in different directions are employed as the check valves (**CV1**, **CV2**) provided for the liquid pipe (**7**).

(Seventeenth Embodiment)

Next, the seventeenth embodiment of the present invention will be described. It is noted that only the difference from the foregoing eleventh embodiment will be described hereinafter.

The refrigerant circuit of this embodiment is applied to an air conditioning system of a heat pump type.

First, the primary refrigerant circuit (**A**) will be described.

As shown in Figure **28**, the primary refrigerant circuit (**A**) makes the heat exchanger (**12**) on the primary heat source side heat/cool the heat exchanger (**1**) on the secondary heat source side. Specifically, the compressor (**11**), the four-position selector valve (**22**), the outdoor heat exchanger (**14**), the heating heat exchanger (**71**), the outdoor electrically motorized expansion valves (**EV-A**, **EV-B**), the cooling heat exchanger (**72**) and the heat exchanger (**12**) on the primary heat source side are connected to each other through the refrigerant pipe (**16**). More specifically, the pipe connected to the heating heat exchanger (**71**) is branched into two, one of which is connected to the cooling heat exchanger (**72**) via the first outdoor electrically motorized expansion valve (**EV-A**) and the other of which is connected to the heat exchanger (**12**) on the primary heat source side via the second electrically motorized expansion valve (**EV-B**).

The gas side of the cooling heat exchanger (**72**) is connected to the inlet side of the compressor (**11**). The gas side of the heat exchanger (**12**) on the primary heat source side is switched between the inlet side and the outlet side of the compressor (**11**) via the four-position selector valve (**22**). A check valve (**CV6**) allowing only the refrigerant to flow from the outdoor heat exchanger (**14**) into heating heat exchanger

(71) is provided for the pipe connecting the outdoor heat exchanger (14) and the heating heat exchanger (71) to each other. A check valve (CV7) allowing only the refrigerant to flow from the heat exchanger (12) on the primary heat source side into the four-position selector valve (22) is provided for the pipe connecting the heat exchanger (12) on the primary heat source side and the four-position selector valve (22) to each other.

A fifth solenoid valve (SV5) is provided between the second outdoor electrically motorized expansion valve (EV-B) and the heating heat exchanger (71). The solenoid valve (SV5) and the heating heat exchanger (71), and the heat exchanger (12) on the primary heat source side and the check valve (CV7) are connected through a gaseous refrigerant by-pass pipe (GBL). A sixth solenoid valve (SV6) and a check valve (CV8) allowing only the refrigerant to flow from the heating heat exchanger (71) into the heat exchanger (12) on the primary heat source side are provided for the gaseous refrigerant by-pass pipe (GBL). The check valve (CV7) and the four-position selector valve (22), and the check valve (CV6) and the heating heat exchanger (71) are connected through an outlet gas by-pass pipe (OGL). A check valve (CV9) allowing only the outlet gas to flow toward the heating heat exchanger (71) is provided for the outlet gas by-pass pipe (OGL).

Next, the secondary refrigerant circuit (B) will be de-

scribed. It is noted that only the difference from the secondary refrigerant circuit (**B**) of the foregoing eleventh embodiment will be described hereinafter.

As shown in Figure **28**, in the secondary refrigerant circuit (**B**), a seventh solenoid valve (**SV7**) opening during the room cooling running and closing during the heating running is provided between the fourth check valve (**CV4**) of the liquid pipe (**7**) and the indoor heat exchanger (**3**), and an eighth solenoid valve (**SV8**) opening during the room cooling running and closing during the heating running is provided between the third check valve (**CV3**) of the liquid pipe (**7**) and the heat exchanger (**1**) on the secondary heat source side.

One end of a heating liquid pipe (**34**) on the recovery side is connected between the seventh solenoid valve (**SV7**) of the liquid pipe (**7**) and the indoor heat exchanger (**3**). The other end of the heating liquid pipe (**34**) on the recovery side is connected between the third check valve (**CV3**) and the eighth solenoid valve (**SV8**) in the liquid pipe (**7**). A ninth solenoid valve (**SV9**) opening during the heating running and closing during the cooling running is provided for the heating liquid pipe (**34**) on the recovery side.

One end of a heating liquid pipe (**35**) on the supply side is connected between the fourth check valve (**CV4**) and the seventh solenoid valve (**SV7**) in the liquid pipe (**7**). The other end of the heating liquid pipe (**35**) on the supply side is

connected between the eighth solenoid valve (**SV8**) of the liquid pipe (**7**) and the heat exchanger (**1**) on the secondary heat source side. A tenth solenoid valve (**SV10**) opening during the heating running and closing during the cooling running is provided for the heating liquid pipe (**35**) on the supply side. The other arrangements are the same as those of the eleventh embodiment described above.

Hereinafter, the room cooling and heating running operations will be described.

During the cooling running, firstly, in the primary refrigerant circuit (**A**), the four-position selector valve (**22**) is switched to the direction indicated by the solid lines, the fifth solenoid valve (**SV5**) is opened and the sixth solenoid valve (**SV6**) is closed. On the other hand, in the secondary refrigerant circuit (**B**), the seventh solenoid valve (**SV7**) and the eighth solenoid valve (**SV8**) are opened and the ninth solenoid valve (**SV9**) and the tenth solenoid valve (**SV10**) are closed.

In such a state, as indicated by the solid-line arrows in Figure **28**, a high-temperature, high-pressure gaseous refrigerant discharged from the compressor (**11**) is condensed in the outdoor heat exchanger (**14**) and the heating heat exchanger (**71**) in the primary refrigerant circuit (**A**). Thereafter, the refrigerant is distributed into the cooling heat exchanger (**72**) and the heat exchanger (**12**) on the primary heat

source side. The pressure of the refrigerant is reduced in the respective outdoor electrically motorized expansion valves (**EV-A, EV-B**). Then, in the cooling heat exchanger (**72**), the refrigerant exchanges heat with the refrigerant in the circulating condenser (**61**). And in the heat exchanger (**12**) on the primary heat source side, the refrigerant exchanges heat with the refrigerant in the heat exchanger (**1**) on the secondary heat source side. Then, these refrigerants are evaporated. The evaporated refrigerants return to the compressor (**11**). This circulation operation is repeated.

On the other hand, in the secondary refrigerant circuit (**B**), a pressurizing operation and a pressure reducing operation are repeatedly performed on the tank (**T**) in the same way as described above. Thus, as a result of the pressurizing operation, the liquid refrigerant is pushed out from the tank (**T**) and the pressure thereof is reduced in the indoor electrically motorized expansion valve (**EV1**) as indicated by the solid line arrows in Figure **28**. Then, the refrigerant exchanges heat with the indoor air and is evaporated in the indoor heat exchanger (**3**), thereby cooling the indoor air. Thereafter, the refrigerant is condensed in the heat exchanger (**1**) on the secondary heat source side. And, as a result of the pressure reducing operation, the condensed liquid refrigerant is recovered into the tank (**T**). By repeating pressurization and pressure reduction in such a manner, the refrigerant is circulated

in the secondary refrigerant circuit (**B**), thereby cooling the indoor air.

Next, the room heating running will be described.

During the heating running, firstly, in the primary refrigerant circuit (**A**), the four-position selector valve (**22**) is switched to the direction indicated by the broken lines, the sixth solenoid valve (**SV6**) is opened and the fifth solenoid valve (**SV5**) is closed. On the other hand, in the secondary refrigerant circuit (**B**), the seventh solenoid valve (**SV7**) and the eighth solenoid valve (**SV8**) are closed and the ninth solenoid valve (**SV9**) and the tenth solenoid valve (**SV10**) are opened.

In such a state, as indicated by the broken-line arrows in Figure **28**, a high-temperature, high-pressure gaseous refrigerant discharged from the compressor (**11**) exchanges heat with the refrigerant in the circulating evaporator (**51**) in the heating heat exchanger (**71**), thereby changing the sensible heat in the primary refrigerant circuit (**A**). Thereafter, the gaseous refrigerant flows through the gaseous refrigerant by-pass pipe (**GBL**) and the heat exchanger (**12**) on the primary heat source side, in which the refrigerant exchanges heat with the refrigerant in the heat exchanger (**1**) on the secondary heat source side and is condensed. A part of the condensed liquid refrigerant is evaporated in the outdoor heat exchanger (**14**) and returns to the compressor (**11**) via the

four-position selector valve (22). The pressure of the other liquid refrigerant is reduced in the first outdoor electrically motorized expansion valve (EV-A). Then, in the cooling heat exchanger (72), the refrigerant exchanges heat with the refrigerant in the circulating condenser (61) and is evaporated to return to the compressor (11). This circulation operation is repeated.

On the other hand, in the secondary refrigerant circuit (B), a pressurizing operation and a pressure reducing operation are repeatedly performed on the tank (T) in the same way as described above. Thus, as a result of the pressurizing operation, the liquid refrigerant is pushed out from the tank (T), flows through the heating liquid pipe (35) on the supply side and the heat exchanger (1) on the secondary heat source side and is evaporated as indicated by the broken-line arrows in Figure 28. Thereafter, in the indoor heat exchanger (3), the refrigerant exchanges heat with the indoor air and is condensed, thereby heating the indoor air. Then, as a result of the pressure reducing operation, the condensed liquid refrigerant is recovered into the tank (T) through the heating liquid pipe (34) on the recovery side. By repeating pressurization and pressure reduction in such a manner, the refrigerant is circulated in the secondary refrigerant circuit (B), thereby heating the indoor air.

(Eighteenth Embodiment)

Next, the eighteenth embodiment of the present invention will be described. It is noted that this embodiment is also applied to an air conditioning system of a heat pump type. Since the primary refrigerant circuit (**A**) is the same as that of the seventeenth embodiment described above, the description thereof will be omitted herein. As for the secondary refrigerant circuit (**B**), only the difference from that of the seventeenth embodiment will be described.

As shown in Figure **29**, the secondary refrigerant circuit (**B**) includes a four-position selector valve (**10**) in the liquid pipe (**7**). Specifically, the four-position selector valve (**10**) is connected to a first liquid pipe (**7A**) extending from the liquid side of the heat exchanger (**1**) on the secondary heat source side, to a second and a third liquid pipe (**7B, 7C**) extending from the tank (**T**) and to a fourth liquid pipe (**7D**) extending from the liquid side of the indoor heat exchanger (**3**). This switches the states how the refrigerant pushed out from the tank (**T**) is supplied to the heat exchanger (**1**) on the secondary heat source side and the indoor heat exchanger (**3**).

Hereinafter, the room cooling and heating running will be described.

First, during the cooling running, the four-position selector valve (**22**) is switched to the direction indicated by the solid lines in the primary refrigerant circuit (**A**) in the

same way as in the seventeenth embodiment described above. On the other hand, in the secondary refrigerant circuit (**B**), the four-position selector valve (**22**) is also switched to the direction indicated by the solid lines. In such a state, pressurization and pressure reduction are repeated by the pressurizing circuit (**50**) and the pressure reducing circuit (**60**). As a result, the refrigerant circulates in the secondary refrigerant circuit (**B**) as indicated by the solid-line arrows in Figure **29**, thereby cooling the indoor air.

On the other hand, during the room heating running, both the four-position selector valves (**22**, **10**) are switched to the direction indicated by the broken lines. In such a state, pressurization and pressure reduction are repeated by the pressurizing circuit (**50**) and the pressure reducing circuit (**60**). As a result, in the secondary refrigerant circuit (**B**), the refrigerant circulates in the opposite direction to that of the cooling running as indicated by the broken-line arrows in Figure **29**, thereby heating the indoor air.

(Nineteenth Embodiment)

Next, the nineteenth embodiment of the present invention will be described with reference to Figures **30** to **32**. It is noted that this embodiment is also applied to an air conditioning system of a heat pump type.

First, the primary refrigerant circuit (**A**) is construct-

ed such that the compressor (11), the four-position selector valve (22), the outdoor heat exchanger (14), the heating heat exchanger (71), the first and the second outdoor electrically motorized expansion valves (EV-A, EV-B), the cooling heat exchanger (72) and the heat exchanger (12) on the primary heat source side are connected to each other through the refrigerant pipe (16) in the same way as in the seventeenth embodiment described above.

More specifically, the gas side of the outdoor heat exchanger (14) is switched between the inlet side and the outlet side of the compressor (11) via the four-position selector valve (22). The outdoor heat exchanger (14) is connected to the heating heat exchanger (71) through a cooling gas supply pipe (CGL). A check valve (CV1) allowing only the refrigerant to flow toward the heating heat exchanger (71) is provided for the cooling gas supply pipe (CGL). The liquid side of the heating heat exchanger (71) is branched into a first and a second liquid branch pipe (LSL-1, LSL-2) via a first solenoid valve (SV1). The first branch pipe (LSL-1) is connected to the cooling heat exchanger (72) via the first outdoor electrically motorized expansion valve (EV-A) and the second branch pipe (LSL-2) is connected to the heat exchanger (12) on the primary heat source side via the second outdoor electrically motorized expansion valve (EV-B).

The gas side of the cooling heat exchanger (72) is con-

nected to the inlet side of the compressor (**11**). The gas side of the heat exchanger (**12**) on the primary heat source side is connected to the four-position selector valve (**22**) via a check valve (**CV2**) and is switched between the inlet side and the outlet side of the compressor (**11**) via the four-position selector valve (**22**).

The second branch pipe (**LSL-2**) and the outdoor heat exchanger (**14**) are connected to each other through a heating liquid pipe (**WLL**), which is provided with a check valve (**CV3**) allowing only the refrigerant to flow toward the outdoor heat exchanger (**14**).

The heating heat exchanger (**71**) and the first solenoid valve (**SV1**), and the heat exchanger (**12**) on the primary heat source side and the check valve (**CV2**) are connected through a heating gas supply pipe (**WGL**). A check valve (**CV4**) allowing only the liquid refrigerant to be supplied to the heat exchanger (**12**) on the primary heat source side and a second solenoid valve (**SV2**) are provided for the heating gas supply pipe (**WGL**). The check valve (**CV2**) and the four-position selector valve (**22**), and the check valve (**CV1**) and the heating heat exchanger (**71**) are connected through an outlet gas by-pass pipe (**GPL**). A check valve (**CV5**) allowing only the refrigerant to flow toward the heating heat exchanger (**71**) is provided for the outlet gas by-pass pipe (**GPL**).

Next, the secondary refrigerant circuit (**B**) will be de-

scribed.

The secondary refrigerant circuit (B) includes: a circulating evaporator (51) for exchanging heat with the heating heat exchanger (71); a circulating condenser (61) for exchanging heat with the cooling heat exchanger (72); a heat exchanger (1) on the secondary heat source side for exchanging heat with the heat exchanger (12) on the primary heat source side; a plurality of indoor heat exchangers (3, 3, 3) and indoor electrically motorized expansion valves (EV1, EV1, EV1) that are connected in parallel to the heat exchanger (1) on the secondary heat source side through a gas pipe (6) and a liquid pipe (7); two main tanks (T1, T2); and two sub-tanks (ST1, ST2).

Specifically, the gas supply pipe (52) connected to the upper end of the circulating evaporator (51) is branched into four branch pipes (52a to 52d), which are individually connected to the upper ends of the respective main tanks (T1, T2) and the respective sub-tanks (ST1, ST2). A first to a fourth tank pressurizing solenoid valve (SV-P1 to SV-P4) are provided for the branch pipes (52a to 52d), respectively.

The liquid recovery pipe (53) connected to the lower end of the circulating evaporator (51) is branched into two branch pipes (53a, 53b), which are individually connected to the lower ends of the respective sub-tanks (ST1, ST2). Check valves (CV6, CV6) allowing only the refrigerant to flow out from the

sub-tanks (**ST1, ST2**) are provided for these branch pipes (**53a, 53b**), respectively.

On the other hand, the gas recovery pipe (**62**) connected to the upper end of the circulating condenser (**61**) is branched into four branch pipes (**62a** to **62d**), which are individually connected to the upper ends of the respective main tanks (**T1, T2**) and the respective sub-tanks (**ST1, ST2**). A first to a fourth tank pressure reducing solenoid valve (**SV-V1** to **SV-V4**) are provided for these branch pipes (**62a** to **62d**), respectively.

The liquid supply pipe (**63**) connected to the lower end of the circulating condenser (**61**) is branched into two branch pipes (**63a, 63b**), which are individually connected to the lower ends of the respective main tanks (**T1, T2**). Check valves (**CV7, CV7**) allowing only the refrigerant to be recovered into the main tanks (**T1, T2**) are provided for these branch pipes (**63a, 63b**), respectively.

The liquid pipe (**7**) extending from the indoor heat exchanger (**3**) is branched into a first and a second liquid pipe (**7A, 7B**). The first branched liquid pipe (**7A**) is connected to the respective branch pipes (**63a, 63b**) via the third solenoid valve (**SV3**). The second branched liquid pipe (**7B**) is connected to the liquid side of the heat exchanger (**1**) on the secondary heat source side via the fourth solenoid valve (**SV4**) and the fifth solenoid valve (**SV5**). The lower ends of the respec-

tive main tanks (**T1**, **T2**) and the respective sub-tanks (**ST1**, **ST2**) are connected between the fourth solenoid valve (**SV4**) and the fifth solenoid valve (**SV5**) in the second branched liquid pipe (**7B**) through the connecting pipes (**17a** to **17d**), respectively.

Check valves (**CV8**, **CV8**, ...) allowing only the refrigerant to flow from the respective main tanks (**T1**, **T2**) and the respective sub-tanks (**ST1**, **ST2**) toward the second branched liquid pipe (**7B**) are provided for the connecting pipes (**17a** to **17d**), respectively. The branch pipes (**63a**, **63b**) are connected between the fifth solenoid valve (**SV5**) and the heat exchanger (**1**) on the secondary heat source side in the second branched liquid pipe (**7B**) through the cooling liquid recovery pipe (**CLL**), which is provided with a sixth solenoid valve (**SV6**).

Hereinafter, the room cooling and heating running operations will be described.

During the cooling running, first, the four-position selector valve (**22**) is switched to the direction indicated by the solid lines, the first solenoid valve (**SV1**) is opened and the second solenoid valve (**SV2**) is closed in the primary refrigerant circuit (**A**). On the other hand, in the secondary refrigerant circuit (**B**), the first and the third tank pressurizing solenoid valves (**SV-P1**, **SV-P3**), the second and the fourth tank pressure reducing solenoid valves (**SV-V2**, **SV-V3**), the fourth solenoid valve (**SV4**) and the sixth solenoid valve (**SV6**)

are opened, and the second and the fourth tank pressurizing solenoid valves (**SV-P2**, **SV-P4**), the first and the third tank pressure reducing solenoid valves (**SV-V1**, **SV-V3**), the third solenoid valve (**SV3**) and the fifth solenoid valve (**SV5**) are closed.

In such a state, in the primary refrigerant circuit (**A**), a high-temperature, high-pressure gaseous refrigerant discharged from the compressor (**11**) sequentially flows through the outdoor heat exchanger (**14**) and the heating heat exchanger (**71**) and is condensed, as indicated by the solid-line arrows in Figure **31**. Thereafter, the refrigerant is distributed into the cooling heat exchanger (**72**) and the heat exchanger (**12**) on the primary heat source side. The pressure of the refrigerant is reduced in the respective outdoor electrically motorized expansion valve (**EV-A**, **EV-B**). Then, in the cooling heat exchanger (**72**), the refrigerant exchanges heat with the refrigerant in the circulating condenser (**61**). And in the heat exchanger (**12**) on the primary heat source side, the refrigerant exchanges heat with the refrigerant in the heat exchanger (**1**) on the secondary heat source side. Then, these refrigerants are evaporated. The evaporated refrigerants return to the compressor (**11**). This circulation operation is repeated.

On the other hand, in the secondary refrigerant circuit (**B**), the internal pressures of the first main tank (**T1**) and

the first sub-tank (**ST1**) become high, whereas the internal pressures of the second main tank (**T2**) and the second sub-tank (**ST2**) become low. Thus, as indicated by the solid-line arrows in Figure **31**, the liquid refrigerant pushed out from the first main tank (**T1**) is passed through the second branched liquid pipe (**7B**). The pressure of the refrigerant is reduced in the indoor electrically motorized expansion valve (**EV1**). Then, in the indoor heat exchanger (**3**), the refrigerant exchanges heat with the indoor air and is evaporated, thereby cooling the indoor air. Thereafter, the refrigerant is condensed in the heat exchanger (**1**) on the secondary heat source side, passed through the cooling liquid recovery pipe (**CLL**) and then recovered into the second main tank (**T2**). On the other hand, since the pressure of the first sub-tank (**ST1**) is equalized with that of the circulating evaporator (**51**), the liquid refrigerant in the first sub-tank (**ST1**) is supplied to the circulating evaporator (**51**) as indicated by the broken-line arrows in Figure **31**. Furthermore, in the meantime, a part of the refrigerant flowing through the second branched liquid pipe (**7B**) is recovered into the second sub-tank (**ST2**).

After such an operation has been performed for a predetermined period of time, the respective solenoid valves in the secondary refrigerant circuit (**B**) are switched. Specifically, the first and the third tank pressurizing solenoid valves (**SV-P1, SV-P3**) and the second and the fourth tank pres-

sure reducing solenoid valves (**SV-V2**, **SV-V3**) are closed, and the second and the fourth tank pressurizing solenoid valves (**SV-P2**, **SV-P4**) and the first and the third tank pressure reducing solenoid valves (**SV-V1**, **SV-V4**) are opened. As a result, the internal pressures of the first main tank (**T1**) and the first sub-tank (**ST1**) become low, whereas the internal pressures of the second main tank (**T2**) and the second sub-tank (**ST2**) become high. Thus, a refrigerant circulation state is established in which the liquid refrigerant pushed out from the second main tank (**T2**) is circulated and then recovered into the first main tank (**T1**). Moreover, the liquid refrigerant in the second sub-tank (**ST2**) is supplied to the circulating evaporator (**51**) and a part of the refrigerant flowing through the second branch pipe (**7B**) is recovered into the first sub-tank (**ST1**).

By repeating the switching operations of the solenoid valves, the refrigerant circulates in the secondary refrigerant circuit (**B**), thereby cooling the indoor air.

Next, the room heating running will be described.

During the heating running, firstly, the four-position selector valve (**22**) is switched to the direction indicated by the broken lines, the second solenoid valve (**SV2**) is opened and the first solenoid valve (**SV1**) is closed in the primary refrigerant circuit (**A**). On the other hand, in the secondary refrigerant circuit (**B**), the third solenoid valve (**SV3**) and

the fifth solenoid valve (**SV5**) are opened and the fourth solenoid valve (**SV4**) and the sixth solenoid valve (**SV6**) are closed. In such a state, the opening/closing operations of the other solenoid valves are repeated.

Specifically, in the same way as in the above-described cooling running, a state where the first and the third tank pressurizing solenoid valves (**SV-P1, SV-P3**) and the second and the fourth tank pressure reducing solenoid valves (**SV-V2, SV-V3**) are opened and the second and the fourth tank pressurizing solenoid valves (**SV-P2, SV-P4**) and the first and the third tank pressure reducing solenoid valves (**SV-V1, SV-V4**) are closed, and an opposite state where the first and the third tank pressurizing solenoid valves (**SV-P1, SV-P3**) and the second and the fourth tank pressure reducing solenoid valves (**SV-V2, SV-V3**) are closed and the second and the fourth tank pressurizing solenoid valves (**SV-P2, SV-P4**) and the first and the third tank pressure reducing solenoid valves (**SV-V1, SV-V4**) are opened, are alternately switched.

As a result, as indicated by the solid-line arrows in Figure **32**, a high-temperature, high-pressure gaseous refrigerant discharged from the compressor (**11**) is passed through the outlet gas by-pass pipe (**GPL**) and exchanges heat with the refrigerant in the circulating evaporator (**51**) in the heating heat exchanger (**71**), thereby changing the sensible heat in the primary refrigerant circuit (**A**). Thereafter, the gaseous

refrigerant flows through the heating gas supply pipe (**WGL**) and the heat exchanger (**12**) on the primary heat source side, in which the refrigerant exchanges heat with the refrigerant in the heat exchanger (**1**) on the secondary heat source side and is condensed. The pressure of the condensed liquid refrigerant is reduced in the second outdoor electrically motorized expansion valve (**EV-B**). Then, the refrigerant is distributed to flow through the cooling heat exchanger (**72**) and the outdoor heat exchanger (**14**). In the cooling heat exchanger (**72**), the refrigerant exchanges heat with the refrigerant in the circulating condenser (**61**). In the outdoor heat exchanger (**14**), the refrigerant exchanges heat with the outdoor air. Then, these refrigerants are evaporated to return to the compressor (**11**). This circulation operation is repeated.

On the other hand, in the secondary refrigerant circuit (**B**), in the state where the internal pressures of the first main tank (**T1**) and the first sub-tank (**ST1**) are high and the internal pressures of the second main tank (**T2**) and the second sub-tank (**ST2**) are low, for example, the liquid refrigerant pushed out from the first main tank (**T1**) flows through the second branched liquid pipe (**7B**) toward the heat exchanger (**1**) on the secondary heat source side, is evaporated in the heat exchanger (**1**) on the secondary heat source side, condensed in the indoor heat exchanger (**3**) and then recovered into the sec-

ond main tank (T2) through the first branched liquid pipe (7A), as indicated by the solid-line arrows in Figure 32. In this case, as indicated by the broken-line arrows in Figure 32, the liquid refrigerant in the first sub-tank (ST1) is also supplied to the circulating evaporator (51) and the refrigerant has been recovered into the second sub-tank (ST2) through the second branched liquid pipe (7A). Then, the switching operations of the solenoid valves are repeated in the above-described manner and the refrigerant circulates in the secondary refrigerant circuit (B), thereby heating the indoor air.

(Twentieth Embodiment)

Next, the twentieth embodiment of the present invention will be described with reference to Figures 33 to 35. It is noted that this embodiment is also applied to an air conditioning system of a heat pump type and that main arrangements are the same as those of the nineteenth embodiment. Thus, only the difference from the nineteenth embodiment will be described hereinafter.

The primary refrigerant circuit (A) includes not only an electrically motorized expansion valve (EV-C) but also a by-pass pipe (BPL) for by-passing the electrically motorized expansion valve (EV-C) between the heating heat exchanger (71) and the first solenoid valve (SV1). A capillary tube (CT) is provided for the by-pass pipe (BPL). The capillary tube (CT)

is provided for the liquid side of the heat exchanger (12) on the primary heat source side, instead of the electrically motorized expansion valve (EV-B). The other arrangements are substantially the same as those of the nineteenth embodiment described above.

On the other hand, in the secondary refrigerant circuit (B), the upper end of the first sub-tank (ST1) is connected to the upper end of the first main tank (T1) and the upper end of the first sub-tank (ST1) is connected to the upper end of the first main tank (T1). Only by performing the switching operations of the solenoid valves (SV-P1, SV-P2, SV-V1, SV-V2) for switching the connection states of the circulating evaporator (51) and the circulating condenser (61) to the respective main tanks (T1, T2), the connection states of the circulating evaporator (51) and the circulating condenser (61) to the respective sub-tanks (ST1, ST2) are switched. The connecting pipes (17a, 17b) coupled to the respective main tanks (T1, T2) are partially branched into the branch pipe (17a-A, 17b-A) and are connected to the first liquid pipe (7A) through the check valves (CV9, CV9). The other arrangements are substantially the same as those of the nineteenth embodiment described above.

The refrigerant circulates as indicated by the arrows in Figure 34 to perform the room cooling during the cooling running of the refrigerant circuit of this embodiment, and cir-

culates as indicated by the arrows in Figure **35** to perform the room heating during the heating running thereof. Since these refrigerant circulation operations are substantially the same as those of the nineteenth embodiment described above, the details thereof will be omitted herein. It is noted that during the cooling running, the electrically motorized expansion valve (**EV-C**) is closed and the pressure of the refrigerant is reduced by the capillary tube (**CT**) in the primary refrigerant circuit (**A**). On the other hand, during the heating running, the electrically motorized expansion valve (**EV-C**) is opened and the refrigerant flows through the heat exchanger (**12**) on the primary heat source side without reducing the pressure thereof.

In the secondary refrigerant circuit (**B**), the refrigerant condensed in the heat exchanger (**1**) on the secondary heat source side is recovered into the main tank (**T2**) through the first liquid pipe (**7A**) and the branch pipe (**17b-A**) during the cooling running. On the other hand, during the heating running, the refrigerant condensed in the indoor heat exchanger (**3**) is recovered into the main tank (**T2**) through the first liquid pipe (**7A**) and the branch pipe (**17b-A**).

(Twenty-first Embodiment)

Next, the twenty-first embodiment of the present invention will be described. It is noted that this embodiment is

also applied to an air conditioning system of a heat pump type and that the main arrangements are the same as those of the twentieth embodiment described above. Thus, only the difference from the twentieth embodiment will be described hereinafter.

The primary refrigerant circuit (**A**) is not provided with the heating gas supply pipe (**WGL**) unlike that of the twentieth embodiment described above. Moreover, no check valve is provided for the pipe connecting the heat exchanger (**12**) on the primary heat source side and the four-position selector valve (**22**).

In the secondary refrigerant circuit (**B**), as for the connection states of the circulating evaporator (**51**) and the circulating condenser (**61**) to the upper ends of the respective main tanks (**T1, T2**) and the respective sub-tanks (**ST1, ST2**), those of the nineteenth embodiment described above are also employed herein.

Furthermore, in this embodiment, the evaporating temperature of the circulating evaporator (**51**) is set higher than the evaporating temperature of the heat exchanger (**1**) on the secondary heat source side during the heating running. That is to say, in the same way as setting the condensing temperature of the circulating condenser (**61**) lower than the condensing temperature of the heat exchanger (**1**) on the secondary heat source side in the first embodiment, the powers

of the heat exchangers are differentiated.

Specifically, the first gas supply pipe (**GL-1**) coupled to the heating heat exchanger (**71**) and the second gas supply pipe (**GL-2**) coupled to the heat exchanger (**12**) on the primary heat source side have respectively different pipe diameters, and the flow rate of the first gas supply pipe (**GL-1**) is set smaller than the flow rate of the second gas supply pipe (**GL-2**) by a predetermined ratio. On the other hand, the heat exchange area between the heating heat exchanger (**71**) and the circulating evaporator (**51**) is set smaller than the heat exchange area between the heat exchanger (**12**) on the primary heat source side and the heat exchanger (**1**) on the secondary heat source side, and the ratio is set smaller than the above-described predetermined ratio.

Specifically, for example, if the ratio of the flow rate of the first gas supply pipe (**GL-1**) to the flow rate of the second gas supply pipe (**GL-2**) is 1:10, then the ratio of the heat exchange area between the heating heat exchanger (**71**) and the circulating evaporator (**51**) to the heat exchange area between the heat exchanger (**12**) on the primary heat source side and the heat exchanger (**1**) on the secondary heat source side is set at 2:10. Thus, in respect of the power as a heat exchanger with respect to the flow rate of a refrigerant, the circulating evaporator (**51**) is superior to the heat exchanger (**1**) on the secondary heat source side. Thus, the evaporating

temperature of the circulating evaporator (51) becomes higher than the evaporating temperature of the heat exchanger (1) on the secondary heat source side. Consequently, the internal pressure of the circulating evaporator (51) becomes higher than the internal pressure of the heat exchanger (1) on the secondary heat source side. During the heating running, the liquid refrigerant is pushed out from the main tanks (T1, T2) toward the heat exchanger (1) on the secondary heat source side. The other arrangements are the same as those of the twentieth embodiment described above.

And, during the cooling running of the refrigerant circuit of this embodiment, the refrigerant circulates in the respective refrigerant circuits (A, B) as indicated by the arrows in Figure 37, thereby cooling the indoor air. Since the refrigerant circulation operation is substantially the same as that of the nineteenth embodiment or the twentieth embodiment described above, the details thereof are omitted herein.

On the other hand, in the primary refrigerant circuit during the heating running, the refrigerant discharged from the compressor (11) is distributed into and condensed in the heating heat exchanger (71) and the heat exchanger (12) on the primary heat source side as indicated by the arrows in Figure 38. And, the refrigerant condensed in the heating heat exchanger (71) flows through the branch pipe (LSL-1) and the cooling heat exchanger (72), exchanges heat with the circulat-

ing condenser (61) and is evaporated to return to the compressor (11). On the other hand, the refrigerant condensed in the heat exchanger (12) on the primary heat source side flows through the heating liquid pipe (WLL) and the outdoor heat exchanger (14), exchanges heat with the outdoor air and is evaporated to return to the compressor (11).

In the secondary refrigerant circuit (B), the refrigerant circulates as indicated by the arrows in Figure 38, thereby heating the indoor air. Since the refrigerant circulation operation in the secondary refrigerant circuit (B) is substantially the same as that of the nineteenth embodiment or the twentieth embodiment described above, the details thereof are omitted herein.

(Other Embodiments)

In the foregoing embodiments, the heat transport system of the present invention has been described as being applied to refrigerant circuitry for an air conditioning system. However, the present invention is not limited thereto, but is applicable to various other types of refrigerating machines.

In the first to the tenth embodiments, the tank (T) is connected to the liquid pipe (7) through the connecting pipe (17). Alternatively, the tank (T) may be directly connected to the liquid pipe (7).

Moreover, in some of the above-described embodiments in

which a plurality of tanks (**T1**, **T2**) and/or a plurality of sub-tanks (**ST1**, **ST2**) are provided, three or more tanks and/or sub-tanks may be provided. That is to say, two or more first tanks (**T1**) and first sub-tanks (**ST1**) and two or more second tanks (**T2**) and second sub-tanks (**ST2**) may be provided such that each first tank (**T1**) and each first sub-tank (**ST1**) perform the same function and each second tank (**T2**) and each second sub-tank (**ST2**) perform the same function.

INDUSTRIAL APPLICABILITY

As described above, the present invention is effectively applicable to a heat transport system used as refrigerant circuitry for an air conditioning system or the like, and more particularly applicable to a heat transport system for transporting heat by circulating a heat transport medium without requiring any driving source such as a pump.



CLAIMS

1. A heat transport system, characterized by comprising:

a refrigerant circuit (B) constituted such that heat exchange means (1) on a heat source side is connected to heat exchange means (3) on an application side through a gas pipe (6) and a liquid pipe (7) so as to circulate a refrigerant therein, the heat exchange means (1) on the heat source side exchanging heat with heat source means (A);

tank means (T) for communicating with the liquid pipe (7) and reserving a liquid refrigerant therein;

pressure regulating means (18) for alternately performing a pressurizing operation for raising an internal pressure of the tank means (T) and a pressure reducing operation for lowering the internal pressure; and

refrigerant control means (H) for allowing only a supply of the liquid refrigerant from the tank means (T) to any of the heat exchange means to be an evaporator during the pressurizing operation of the pressure regulating means (18) and allowing only a recovery of the liquid refrigerant from any of the heat exchange means to be a condenser to the tank means (T) during the pressure reducing operation thereof, thereby circulating the refrigerant of the refrigerant circuit (B) and making the heat exchange means (3) on the application side absorb or radiate heat.

2. The heat transport system of Claim 1, characterized

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in that the heat exchange means (3) on the application side is an evaporator for absorbing heat,

and that the refrigerant control means (H)

allows the liquid refrigerant to be supplied from the tank means (T) to the heat exchange means (3) on the application side and prevents the liquid refrigerant from being supplied from the tank means (T) to the heat exchange means (1) on the heat source side during the pressurizing operation of the pressure regulating means (18), and

allows the liquid refrigerant to be recovered from the heat exchange means (1) on the heat source side to the tank means (T) and prevents the liquid refrigerant from being recovered from the heat exchange means (3) on the application side to the tank means (T) during the pressure reducing operation of the pressure regulating means (18).

3. The heat transport system of Claim 1, characterized in that the heat exchange means (3) on the application side is a condenser for radiating heat,

and that the refrigerant control means (H)

allows the liquid refrigerant to be supplied from the tank means (T) to the heat exchange means (1) on the heat source side and prevents the liquid refrigerant from being supplied from the tank means (T) to the heat exchange means (3) on the application side during the pressurizing operation of the pressure regulating means (18), and

allows the liquid refrigerant to be recovered from the heat exchange means (3) on the application side to the tank means (T) and prevents the liquid refrigerant from being recovered from the heat exchange means (1) on the heat source side to the tank means (T) during the pressure reducing operation of the pressure regulating means (18).

4. The heat transport system of Claim 1, 2 or 3, characterized in that the pressure regulating means (18) applies heat to the refrigerant in the tank means (T) so as to raise the internal pressure of the tank means (T) during the pressurizing operation, and extracts heat from the refrigerant in the tank means (T) so as to lower the internal pressure of the tank means (T) during the pressure reducing operation.

5. The heat transport system of Claim 4, characterized in that the pressure regulating means (18) is heat exchange means (18b), which is adjacent to the tank means (T) and switchably performs a heating operation of applying heat to the refrigerant in the tank means (T) and a cooling operation of extracting heat from the refrigerant in the tank means (T).

6. The heat transport system of Claim 4, characterized in that the pressure regulating means (18) includes a refrigerant circuit (D) including: a compressor (D1); a first heat exchanger (D3); a pressure reducing mechanism (D4); a second heat exchanger (D5); and selector means (D2) for alternately

switching connection states of the first heat exchanger (D3) and the second heat exchanger (D5) to an outlet side of the compressor (D1),

and that the first heat exchanger (D3) exchanges heat with the tank means (T) and heats and cools the refrigerant in the tank means (T) in accordance with a switching operation of the selector means (D2).

7. The heat transport system of Claim 1, 2 or 3, characterized in that the pressure regulating means (18) includes pressure generating means (18a, 18c, 19c) coupled to the tank means (T) through a pressure pipe (19), applies a high pressure from the pressure generating means (18a, 18c, 19c) to the inside of the tank means (T) during a pressurizing operation, and applies a low pressure from the pressure generating means (18a, 18c, 19c) to the inside of the tank means (T) during a pressure reducing operation.

8. The heat transport system of Claim 7, characterized in that the pressure generating means is a reservoir container (18a) in which a liquid refrigerant is reservable,

and that the pressure regulating means (18) applies heat to the liquid refrigerant in the reservoir container (18a) so as to evaporate the liquid refrigerant and to raise an internal pressure of the reservoir container (18a) during a pressurizing operation, and extracts heat from a gaseous refrigerant in the reservoir container (18a) so as to condense the

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gaseous refrigerant and to lower the internal pressure of the reservoir container (18a) during a pressure reducing operation.

9. The heat transport system of Claim 7, characterized in that the pressure generating means is a compressor (18c),

and that a connection state of the pressure pipe (19) to the compressor (18c) is switched to an outlet side and an inlet side of the compressor (18c) by selector means (I),

and that the pressure regulating means (18) connects the pressure pipe (19) to the outlet side of the compressor (18c) during a pressurizing operation and connects the pressure pipe (19) to the inlet side of the compressor (18c) during a pressure reducing operation in accordance with a switching operation of the selector means (I).

10. The heat transport system of Claim 7, characterized in that the pressure generating means is a heat exchanger (19c) in which a refrigerant is reservable,

and that the pressure regulating means (18) applies heat to the refrigerant in the heat exchanger (19c) so as to raise an internal pressure of the heat exchanger (19c) during a pressurizing operation and extracts heat from the refrigerant in the heat exchanger (19c) so as to lower the internal pressure of the heat exchanger (19c) during a pressure reducing operation.

11. The heat transport system of Claim 10, characterized

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in that the pressure regulating means (18) includes a refrigerant circuit (D) including: a compressor (D1); a first heat exchanger (D3); a pressure reducing mechanism (D4); a second heat exchanger (D5); and selector means (D2) for alternately switching connection states of the first heat exchanger (D3) and the second heat exchanger (D5) to an outlet side of the compressor (D1),

and that the first heat exchanger (D3) exchanges heat with the heat exchanger (19c) and heats and cools the refrigerant in the heat exchanger (19c) in accordance with a switching operation of the selector means (D2).

12. The heat transport system of Claim 2, characterized in that the refrigerant control means (H) is constituted by:

a first solenoid valve (SV-A), which is provided between a connecting point of the tank means (T) to the liquid pipe (7) and the heat exchange means (1) on the heat source side, closes during the pressurizing operation of the pressure regulating means (18) and opens during the pressure reducing operation thereof; and

a second solenoid valve (SV-B), which is provided between the connecting point of the tank means (T) to the liquid pipe (7) and the heat exchange means (3) on the application side, opens during the pressurizing operation of the pressure regulating means (18) and closes during the pressure reducing operation thereof.

13. The heat transport system of Claim 3, characterized in that the refrigerant control means (H) is constituted by:

a first solenoid valve (SV-A), which is provided between a connecting point of the tank means (T) to the liquid pipe (7) and the heat exchange means (1) on the heat source side, opens during the pressurizing operation of the pressure regulating means (18) and closes during the pressure reducing operation thereof; and

a second solenoid valve (SV-B), which is provided between the connecting point of the tank means (T) to the liquid pipe (7) and the heat exchange means (3) on the application side, closes during the pressurizing operation of the pressure regulating means (18) and opens during the pressure reducing operation thereof.

14. The heat transport system of Claim 2, characterized in that the refrigerant control means (H) is constituted by:

a first check valve (CV1), which is provided between a connecting point of the tank means (T) to the liquid pipe (7) and the heat exchange means (1) on the heat source side and allows only the liquid refrigerant to flow from the heat exchange means (1) on the heat source side to the tank means (T); and

a second check valve (CV2), which is provided between the connecting point of the tank means (T) to the liquid pipe (7) and the heat exchange means (3) on the application side

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and allows only the liquid refrigerant to flow from the tank means (T) to the heat exchange means (3) on the application side.

15. The heat transport system of Claim 3, characterized in that the refrigerant control means (H) is constituted by:

a first check valve (CV3), which is provided between a connecting point of the tank means (T) to the liquid pipe (7) and the heat exchange means (1) on the heat source side and allows only the liquid refrigerant to flow from the tank means (T) to the heat exchange means (1) on the heat source side; and

a second check valve (CV4), which is provided between the connecting point of the tank means (T) to the liquid pipe (7) and the heat exchange means (3) on the application side and allows only the liquid refrigerant to flow from the heat exchange means (3) on the application side to the tank means (T).

16. A heat transport system, characterized by comprising:

a refrigerant circuit (B) constituted such that heat exchange means (1) on a heat source side is connected to heat exchange means (3) on an application side through a gas pipe (6) and a liquid pipe (7) so as to circulate a refrigerant therein, the heat exchange means (1) on the heat source side exchanging heat with heat source means (A);

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at least one first tank means (T1) and at least one second tank means (T2), which are connected in parallel to the liquid pipe (7) and which reserve a liquid refrigerant therein;

pressure regulating means (18) for alternately switching a first pressure state, in which an internal pressure of the first tank means (T1) is raised and an internal pressure of the second tank means (T2) is lowered, and a second pressure state, in which the internal pressure of the first tank means (T1) is lowered and the internal pressure of the second tank means (T2) is raised; and

refrigerant control means (H) for supplying the liquid refrigerant from the first tank means (T1) to any of the heat exchange means to be an evaporator and recovering the liquid refrigerant from any of the heat exchange means to be a condenser to the second tank means (T2) during the first pressure state of the pressure regulating means (18), and for supplying the liquid refrigerant from the second tank means (T2) to any of the heat exchange means to be an evaporator and recovering the liquid refrigerant from any of the heat exchange means to be a condenser to the first tank means (T1) during the second pressure state, thereby circulating the refrigerant of the refrigerant circuit (B) and making the heat exchange means (3) on the application side continuously absorb or radiate heat.

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17. The heat transport system of Claim 16, characterized in that the heat exchange means (3) on the application side is an evaporator for absorbing heat,

and that the refrigerant control means (H) switches refrigerant flow states in the liquid pipe (7)

so as to supply the liquid refrigerant from the first tank means (T1) to the heat exchange means (3) on the application side and recover the liquid refrigerant from the heat exchange means (1) on the heat source side to the second tank means (T2) during the first pressure state of the pressure regulating means (18), and

so as to supply the liquid refrigerant from the second tank means (T2) to the heat exchange means (3) on the application side and recover the liquid refrigerant from the heat exchange means (1) on the heat source side to the first tank means (T1) during the second pressure state of the pressure regulating means (18).

18. The heat transport system of Claim 16, characterized in that the heat exchange means (3) on the application side is a condenser for radiating heat,

and that the refrigerant control means (H) switches refrigerant flow states in the liquid pipe (7)

so as to supply the liquid refrigerant from the first tank means (T1) to the heat exchange means (1) on the heat source side and recover the liquid refrigerant from the heat

exchange means (3) on the application side to the second tank means (T2) during the first pressure state of the pressure regulating means (18), and

so as to supply the liquid refrigerant from the second tank means (T2) to the heat exchange means (1) on the heat source side and recover the liquid refrigerant from the heat exchange means (3) on the application side to the first tank means (T1) during the second pressure state of the pressure regulating means (18).

19. The heat transport system of Claim 16, 17 or 18, characterized in that the pressure regulating means (18) includes pressure generating means (18A, 18B, D1, E1, E2) coupled to the respective tank means (T1, T2) through pressure pipes (19d, 19e), makes the pressure generating means (18A, 18B, D1, E1, E2) apply a high pressure to the inside of the first tank means (T1) and a low pressure to the inside of the second tank means (T2) during the first pressure state, and makes the pressure generating means (18A, 18B, D1, E1, E2) apply a high pressure to the inside of the second tank means (T2) and a low pressure to the inside of the first tank means (T1) during the second pressure state.

20. The heat transport system of Claim 19, characterized in that the pressure generating means are a first reservoir container (18A), which is connected to the first tank means (T1) and in which a liquid refrigerant is reservable, and a

second reservoir container (**18B**), which is connected to the second tank means (**T2**) and in which a liquid refrigerant is reservable,

and that the pressure regulating means (**18**) applies heat to the liquid refrigerant in the first reservoir container (**18A**) so as to evaporate the liquid refrigerant and to raise an internal pressure of the reservoir container (**18A**) and extracts heat from a gaseous refrigerant in the second reservoir container (**18B**) so as to condense the gaseous refrigerant and to lower the internal pressure of the reservoir container (**18B**) during the first pressure state, and the pressure regulating means (**18**) applies heat to the liquid refrigerant in the second reservoir container (**18B**) so as to evaporate the liquid refrigerant and to raise an internal pressure of the reservoir container (**18B**) and extracts heat from a gaseous refrigerant in the first reservoir container (**18A**) so as to condense the gaseous refrigerant and to lower the internal pressure of the reservoir container (**18A**) during the second pressure state.

21. The heat transport system of Claim 19, characterized in that the pressure generating means is a compressor (**D1**),

and that switching of connection states of the first tank means (**T1**) and the second tank means (**T2**) to the compressor (**D1**) is performed by making selector means (**I**) switch the pressure pipes (**19d, 19e**) to an outlet side and an inlet

side of the compressor (D1),

and that the pressure regulating means (18) connects the outlet side of the compressor (D1) to the first tank means (T1) and the inlet side of the compressor (D1) to the second tank means (T2) during the first pressure state, and connects the outlet side of the compressor (D1) to the second tank means (T2) and the inlet side of the compressor (D1) to the first tank means (T1) during the second pressure state.

22. The heat transport system of Claim 19, characterized in that the pressure generating means are a first heat exchanger (E1), which is connected to the first tank means (T1) and in which a refrigerant is reservable, and a second heat exchanger (E2), which is connected to the second tank means (T2) and in which a refrigerant is reservable,

and that the pressure regulating means (18) applies heat to the refrigerant in the first heat exchanger (E1) so as to raise an internal pressure of the heat exchanger (E1) and extracts heat from the refrigerant in the second heat exchanger (E2) so as to lower the internal pressure of the heat exchanger (E2) during the first pressure state, and the pressure regulating means (18) applies heat to the refrigerant in the second heat exchanger (E2) so as to raise an internal pressure of the heat exchanger (E2) and extracts heat from the refrigerant in the first heat exchanger (E1) so as to lower the internal pressure of the heat exchanger (E1) during

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the second pressure state.

23. The heat transport system of Claim 22, characterized in that the pressure regulating means (18) includes a refrigerant circuit (D) including: a compressor (D1); a first heat exchanger (D3); a pressure reducing mechanism (D4); a second heat exchanger (D5); and selector means (I) for alternately switching connection states of the first heat exchanger (D3) and the second heat exchanger (D5) to an outlet side of the compressor (D1),

and that the first heat exchanger (D3) exchanges heat with the first heat exchanger (E1) connected to the first tank means (T1), the second heat exchanger (D5) exchanges heat with the second heat exchanger (E2) connected to the second tank means (T2), and the heat exchangers (E1, E2) are switched between the first pressure state and the second pressure state in accordance with a switching operation of the selector means (I).

24. The heat transport system of Claim 19, characterized in that the pressure generating means is constituted by a pressurizing heat exchanger (E2) to be heated by a heating heat exchanger (D3) and a pressure reducing heat exchanger (E1) to be cooled by a cooling heat exchanger (D5),

and that the pressure regulating means (18) connects the pressurizing heat exchanger (E2) to the first tank means (T1) and the pressure reducing heat exchanger (E1) to the second

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tank means (T2) during the first pressure state, and connects the pressurizing heat exchanger (E2) to the second tank means (T2) and the pressure reducing heat exchanger (E1) to the first tank means (T1) during the second pressure state.

25. The heat transport system of Claim 24, characterized in that the pressure regulating means (18) includes a refrigerant circuit (D) formed by connecting a compressor (D1), a heating heat exchanger (D3), a pressure reducing mechanism (D4) and a cooling heat exchanger (D5) in this order through a refrigerant pipe.

26. The heat transport system of Claim 1, characterized in that the pressure regulating means (18) includes: pressurizing means (50) for performing a pressurizing operation of pushing the liquid refrigerant in the tank means (T) to the liquid pipe (7) by raising the internal pressure of the tank means (T); and pressure reducing means (60) for performing a pressure reducing operation of recovering the liquid refrigerant from the liquid pipe (7) to the tank means (T) by lowering the internal pressure of the tank means (T),

and that the pressure reducing means (60) includes a circulating condenser (61), which is connected to the tank means (T) and which lowers the internal pressure of the tank means (T) by condensing the refrigerant,

and that a condensing pressure of the circulating condenser (61) is set lower than a condensing pressure of the

heat exchange means to be the condenser.

27. The heat transport system of Claim 1, characterized in that the pressure regulating means (18) includes: pressurizing means (50) for performing a pressurizing operation of pushing the liquid refrigerant in the tank means (T) to the liquid pipe (7) by raising the internal pressure of the tank means (T); and pressure reducing means (60) for performing a pressure reducing operation of recovering the liquid refrigerant from the liquid pipe (7) to the tank means (T) by lowering the internal pressure of the tank means (T),

and that the pressurizing means (50) includes a circulating evaporator (51), which is connected to the tank means (T) and which raises the internal pressure of the tank means (T) by evaporating the refrigerant,

and that an evaporating pressure of the circulating evaporator (51) is set higher than an evaporating pressure of the heat exchange means to be the evaporator.

28. The heat transport system of Claim 27, characterized in that auxiliary tank means (ST) is provided above the circulating evaporator (51),

and that selector means (I) is also provided for recovering the liquid refrigerant in the liquid pipe (7) to the auxiliary tank means (ST) by making the auxiliary tank means (ST) communicate with the pressure reducing means (60) and the liquid pipe (7) during the pressure reducing operation of

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the pressure reducing means (60) and for dripping and supplying the liquid refrigerant in the auxiliary tank means (ST) to the circulating evaporator (51) by making the auxiliary tank means (ST) communicate with the pressurizing means (50) during the pressurizing operation of the pressurizing means (50).

29. The heat transport system of Claim 27, characterized in that at least one first auxiliary tank means (ST1) and at least one second auxiliary tank means (ST2) are provided above the circulating evaporator (51),

and that a selector means (I) is also provided for selecting a first selection state in which the liquid refrigerant in the liquid pipe (7) is recovered to the first auxiliary tank means (ST1) by making the first auxiliary tank means (ST1) communicate with the pressure reducing means (60) and the liquid pipe (7) and in which the liquid refrigerant in the second auxiliary tank means (ST2) is dripped and supplied to the circulating evaporator (51) by making the second auxiliary tank means (ST2) communicate with the pressurizing means (50) or a second selection state in which the liquid refrigerant in the liquid pipe (7) is recovered to the second auxiliary tank means (ST2) by making the second auxiliary tank means (ST2) communicate with the pressure reducing means (60) and the liquid pipe (7) and in which the liquid refrigerant in the first auxiliary tank means (ST1) is dripped and

supplied to the circulating evaporator (51) by making the first auxiliary tank means (ST1) communicate with the pressurizing means (50).

30. The heat transport system of Claim 16, characterized in that the pressure regulating means (18) includes: pressurizing means (50) for performing a pressurizing operation of pushing the liquid refrigerant in one of the first tank means (T1) and the second tank means (T2) to the liquid pipe (7) by raising the internal pressure of the one tank means (T1 or T2); and pressure reducing means (60) for performing a pressure reducing operation of recovering the liquid refrigerant from the liquid pipe (7) to the other tank means (T2 or T1) by lowering the internal pressure of the other tank means (T2 or T1),

and that the pressure reducing means (60) includes a circulating condenser (61), which is connected to the respective tank means (T1, T2) and which lowers the internal pressure of each said tank means (T1, T2) by condensing the refrigerant,

and that a condensing pressure of the circulating condenser (61) is set lower than a condensing pressure of the heat exchange means to be the condenser,

and that the pressure regulating means (18) makes the pressurizing means (50) pressurize the first tank means (T1) and makes the pressure reducing means (60) reduce a pressure

* C L M S *

of the second tank means (T2) during a first pressure state, and the pressure regulating means (18) makes the pressurizing means (50) pressurize the second tank means (T2) and makes the pressure reducing means (60) reduce a pressure of the first tank means (T1) during a second pressure state.

31. The heat transport system of Claim 16, characterized in that the pressure regulating means (18) includes: pressurizing means (50) for performing a pressurizing operation of pushing the liquid refrigerant in one of the first tank means (T1) and the second tank means (T2) to the liquid pipe (7) by raising the internal pressure of the one tank means (T1 or T2); and pressure reducing means (60) for performing a pressure reducing operation of recovering the liquid refrigerant from the liquid pipe (7) to the other tank means (T2 or T1) by lowering the internal pressure of the other tank means (T2 or T1),

and that the pressurizing means (50) includes a circulating evaporator (51), which is connected to the respective tank means (T1, T2) and which raises the internal pressure of each said tank means (T1, T2) by evaporating the refrigerant,

and that an evaporating pressure of the circulating evaporator (51) is set higher than an evaporating pressure of the heat exchange means to be the evaporator,

and that the pressure regulating means (18) makes the pressurizing means (50) pressurize the first tank means (T1)

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and makes the pressure reducing means (60) reduce a pressure of the second tank means (T2) during a first pressure state, and the pressure regulating means (18) makes the pressurizing means (50) pressurize the second tank means (T2) and makes the pressure reducing means (60) reduce a pressure of the first tank means (T1) during a second pressure state.

32. The heat transport system of Claim 31, characterized in that at least one first auxiliary tank means (ST1) and at least one second auxiliary tank means (ST2) are provided above the circulating evaporator (51),

and that selector means (I) is also provided for selecting a first selection state in which the liquid refrigerant in the liquid pipe (7) is recovered to the first auxiliary tank means (ST1) by making the first auxiliary tank means (ST1) communicate with the pressure reducing means (60) and the liquid pipe (7) and in which the liquid refrigerant in the second auxiliary tank means (ST2) is dripped and supplied to the circulating evaporator (51) by making the second auxiliary tank means (ST2) communicate with the pressurizing means (50) or a second selection state in which the liquid refrigerant in the liquid pipe (7) is recovered to the second auxiliary tank means (ST2) by making the second auxiliary tank means (ST2) communicate with the pressure reducing means (60) and the liquid pipe (7) and in which the liquid refrigerant in the first auxiliary tank means (ST1) is dripped and

supplied to the circulating evaporator (51) by making the first auxiliary tank means (ST1) communicate with the pressurizing means (50).

33. The heat transport system of Claim 26 or 30, characterized in that the heat source means (A) includes: first heat exchange means (12) for exchanging heat with the heat exchange means (1) on the heat source side; and second heat exchange means (72) for exchanging heat with the circulating condenser (61),

and that during heat absorption running of the heat exchange means (3) on the application side, an evaporating temperature of the first heat exchange means (12) and an evaporating temperature of the second heat exchange means (72) are equal to each other but a ratio of a capacity of the circulating condenser (61) to a flow rate of the refrigerant flowing through the second heat exchange means (72) is set larger than a ratio of a capacity of the heat exchange means (1) on the heat source side to a flow rate of the refrigerant flowing through the first heat exchange means (12).

34. The heat transport system of Claim 27 or 31, characterized in that the heat source means (A) includes: first heat exchange means (12) for exchanging heat with the heat exchange means (1) on the heat source side; and second heat exchange means (71) for exchanging heat with the circulating evaporator (51),

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and that during heat radiation running of the heat exchange means (3) on the application side, a condensing temperature of the first heat exchange means (12) and a condensing temperature of the second heat exchange means (71) are equal to each other but a ratio of a capacity of the circulating evaporator (51) to a flow rate of the refrigerant flowing through the second heat exchange means (71) is set larger than a ratio of a capacity of the heat exchange means (1) on the heat source side to a flow rate of the refrigerant flowing through the first heat exchange means (12).

35. The heat transport system of Claim 26 or 30, characterized in that the pressure reducing means (60) includes: a gas recovering pipe (62) for connecting an upper end of the tank means (T) to a gas side of the circulating condenser (61); and a liquid supplying pipe (63) for connecting a lower end of the tank means (T) to a liquid side of the circulating condenser (1),

and that the liquid supplying pipe (63) is connected to the lower end of the tank means (T) independent of the liquid pipe (7).

36. The heat transport system of Claim 1 or 16, characterized in that the pressure regulating means (18) includes: pressurizing means (50) for performing a pressurizing operation of pushing the liquid refrigerant in the tank means (T) to the liquid pipe (7) by raising the internal pressure of

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the tank means (T); and pressure reducing means (60) for performing a pressure reducing operation of recovering the liquid refrigerant from the liquid pipe (7) to the tank means (T) by lowering the internal pressure of the tank means (T),

and that the pressure reducing means (60) includes a circulating condenser (61), which is connected to the tank means (T) and which lowers the internal pressure of the tank means (T) by condensing the refrigerant,

and that the pressurizing means (50) includes a circulating evaporator (51), which is connected to the tank means (T) and which raises the internal pressure of the tank means (T) by evaporating the refrigerant,

and that the heat source means (A) includes: first heat exchange means (12) for exchanging heat with the compressor (11) and the heat exchange means (1) on the heat source side; second heat exchange means (72) for exchanging heat with the circulating condenser (61); and third heat exchange means (71) for exchanging heat with the circulating evaporator (51), and that, during heat radiation running of the heat exchange means (3) on the application side, the heat source means (A) makes the third heat exchange means (71) exchange heat of the gaseous refrigerant discharged from the compressor (11) with the circulating evaporator (51) so as to change sensible heat of the refrigerant, makes the first heat exchange means (12) exchange heat with the heat exchange means

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(1) on the heat source side so as to condense the refrigerant, and then makes the second heat exchange means (72) exchange heat with the circulating condenser (61) so as to evaporate the refrigerant.

37. The heat transport system of Claim 1 or 16, characterized in that the pressure regulating means (18) includes: pressurizing means (50) for performing a pressurizing operation of pushing the liquid refrigerant in the tank means (T) to the liquid pipe (7) by raising the internal pressure of the tank means (T); and pressure reducing means (60) for performing a pressure reducing operation of recovering the liquid refrigerant from the liquid pipe (7) to the tank means (T) by lowering the internal pressure of the tank means (T),

and that the pressure reducing means (60) includes a circulating condenser (61), which is connected to the tank means (T) and which lowers the internal pressure of the tank means (T) by condensing the refrigerant,

and that the pressurizing means (50) includes a circulating evaporator (51), which is connected to the tank means (T) and which raises the internal pressure of the tank means (T) by evaporating the refrigerant,

and that the heat source means (A) includes: first heat exchange means (12) for exchanging heat with the compressor (11) and the heat exchange means (1) on the heat source side; second heat exchange means (72) for exchanging heat with the

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circulating condenser (61); and third heat exchange means (71) for exchanging heat with the circulating evaporator (51), and that, during heat radiation running of the heat exchange means (3) on the application side, the heat source means (A) distributes the gaseous refrigerant discharged from the compressor (11) to the third heat exchange means (71) and the first heat exchange means (12), makes the third heat exchange means (71) exchange heat with the circulating evaporator (51) so as to condense the refrigerant, makes the first heat exchange means (12) exchange heat with the heat exchange means (1) on the heat source side so as to condense the refrigerant, and then makes the second heat exchange means (72) exchange heat of the condensed refrigerant with the circulating condenser (61) so as to evaporate the refrigerant.

ABSTRACT

A heat exchanger (1) on the secondary heat source, which exchanges heat with a heat exchanger (12) on the primary heat source in a primary cooling circuit (A), is connected with an indoor heat exchanger (3) through a gas pipe (6) and a liquid pipe (7). A tank (T) storing a liquid cooling medium is connected at its lower end to the liquid pipe (7) and its upper end to a pressure adjustment mechanism (18). Check valves (CV1 and CV2) are disposed on both sides of the connecting point of the tank (T) with respect to the liquid pipe (7). The internal pressure of the tank (T) is changed over alternately between a high pressure state and a low pressure state by the pressure adjustment mechanism (18) so that the liquid cooling medium is supplied to the indoor heat exchanger (3) at the time of the high pressure operation, and the liquid cooling medium is recovered from the heat exchanger (1) on the secondary side to the tank (T) and is circulated by a secondary cooling circuit (B) at the time of the low pressure operation.

Fig. 1

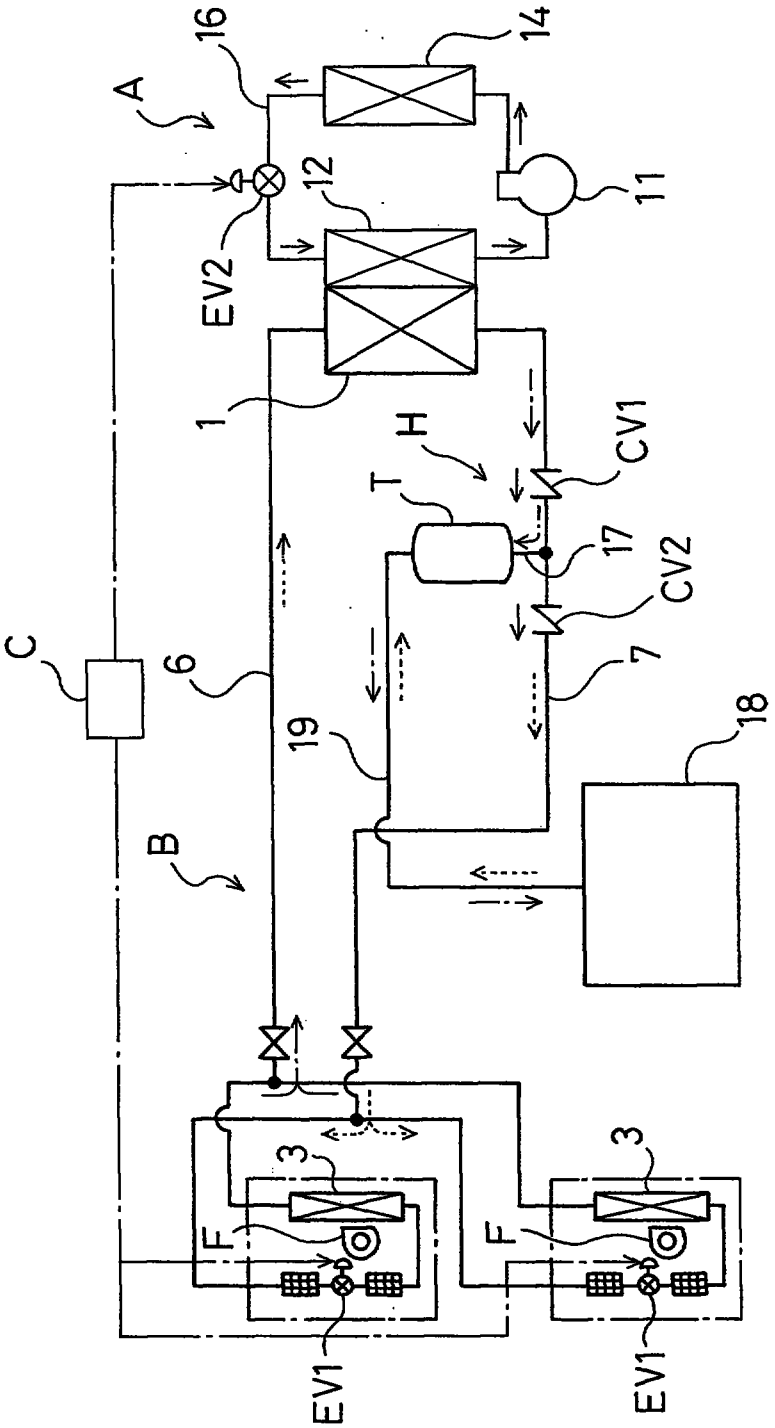


Fig. 2

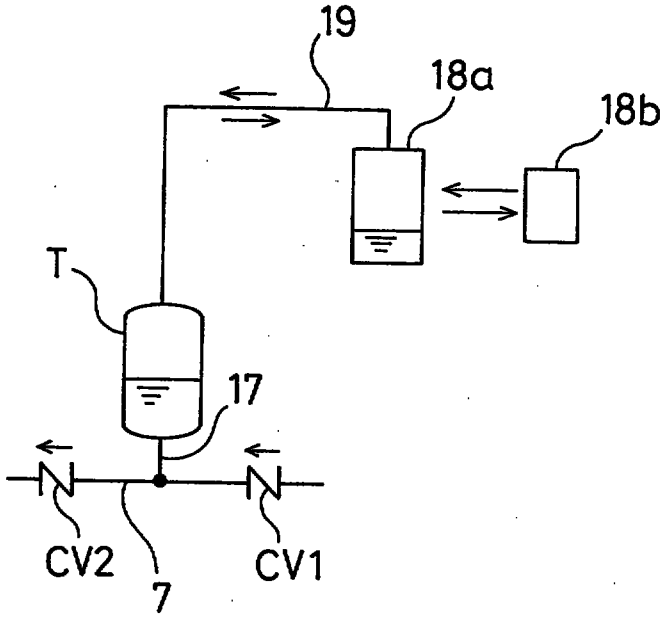


Fig. 3

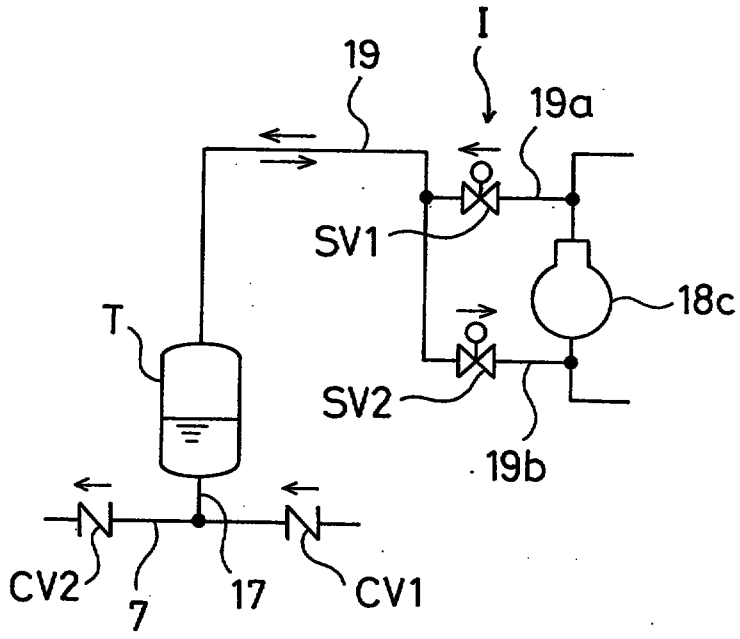


Fig. 4

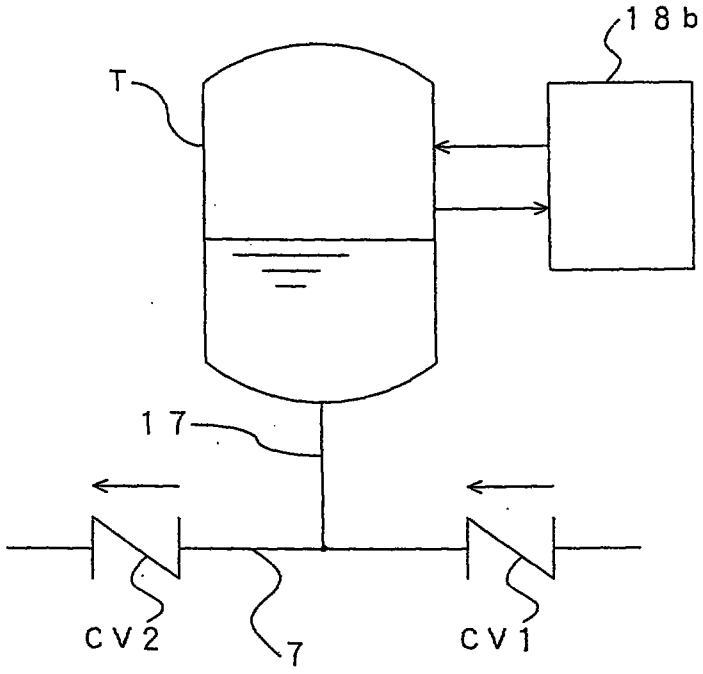


Fig. 5

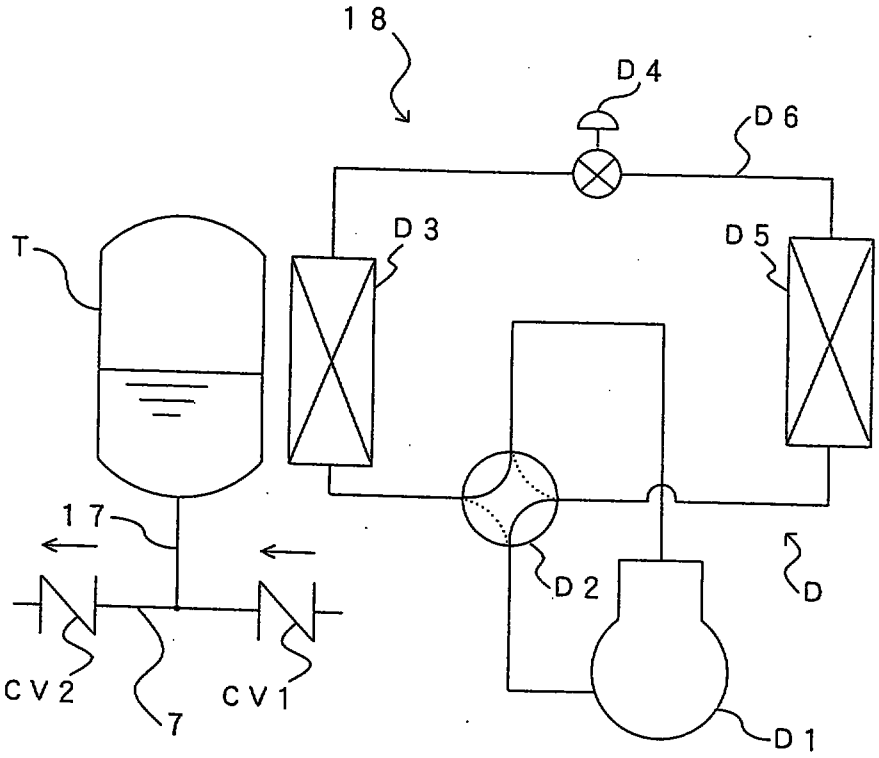


Fig. 6

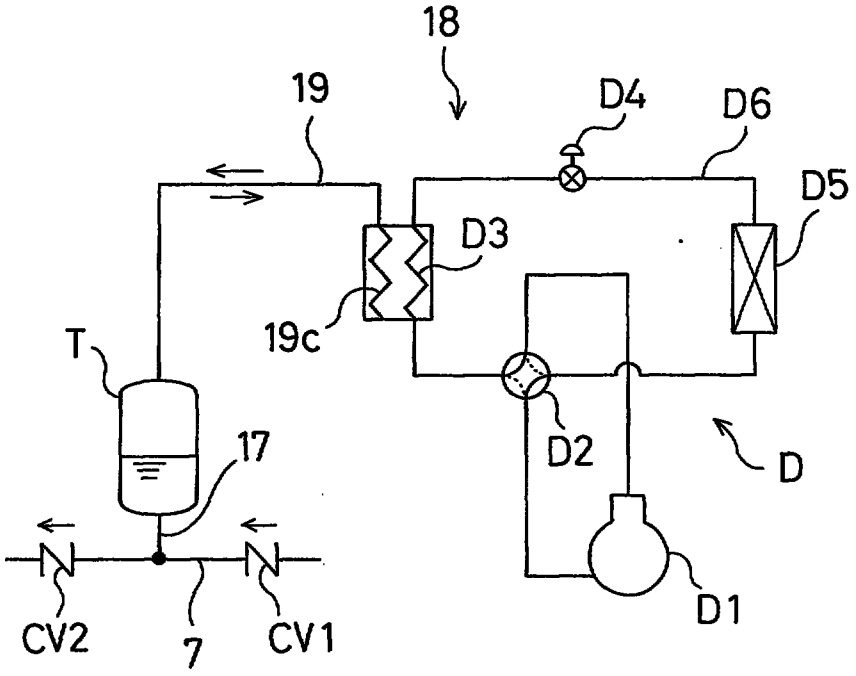


Fig. 7

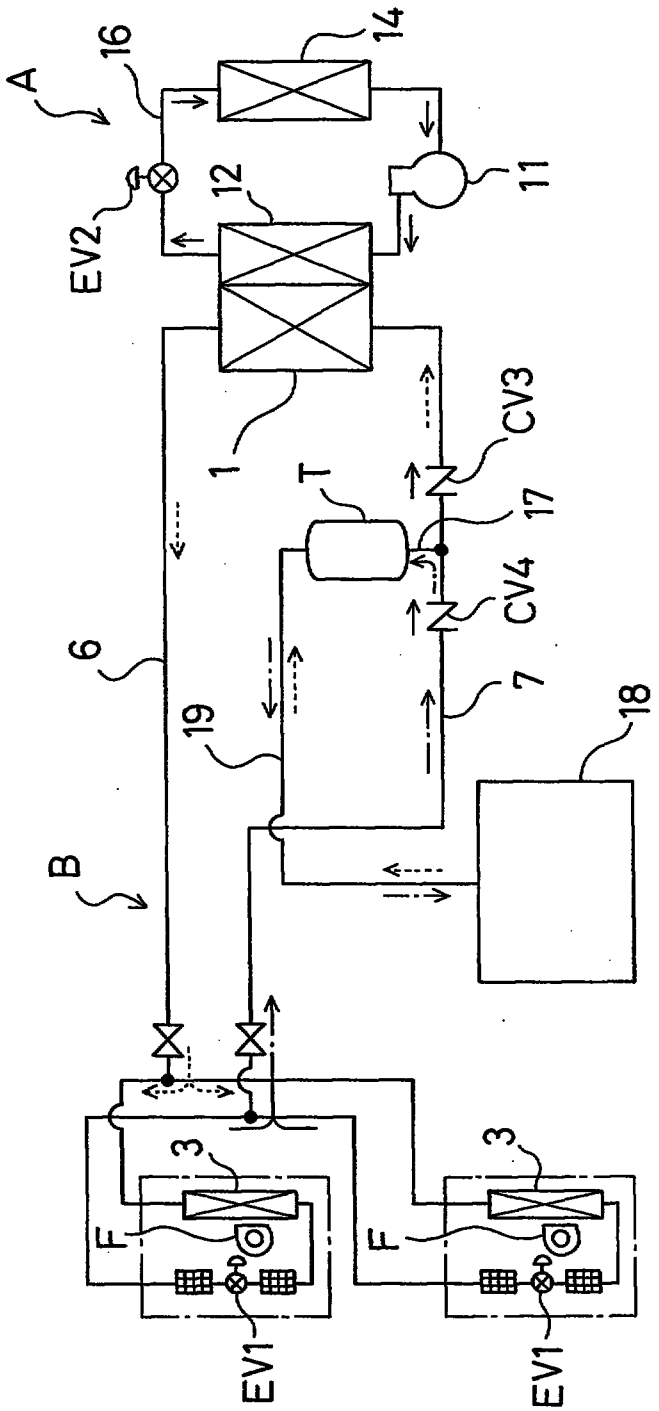


Fig. 8

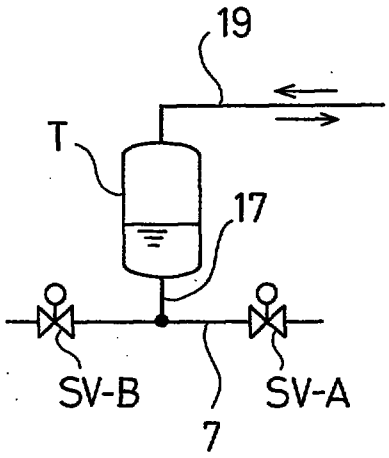


Fig. 9

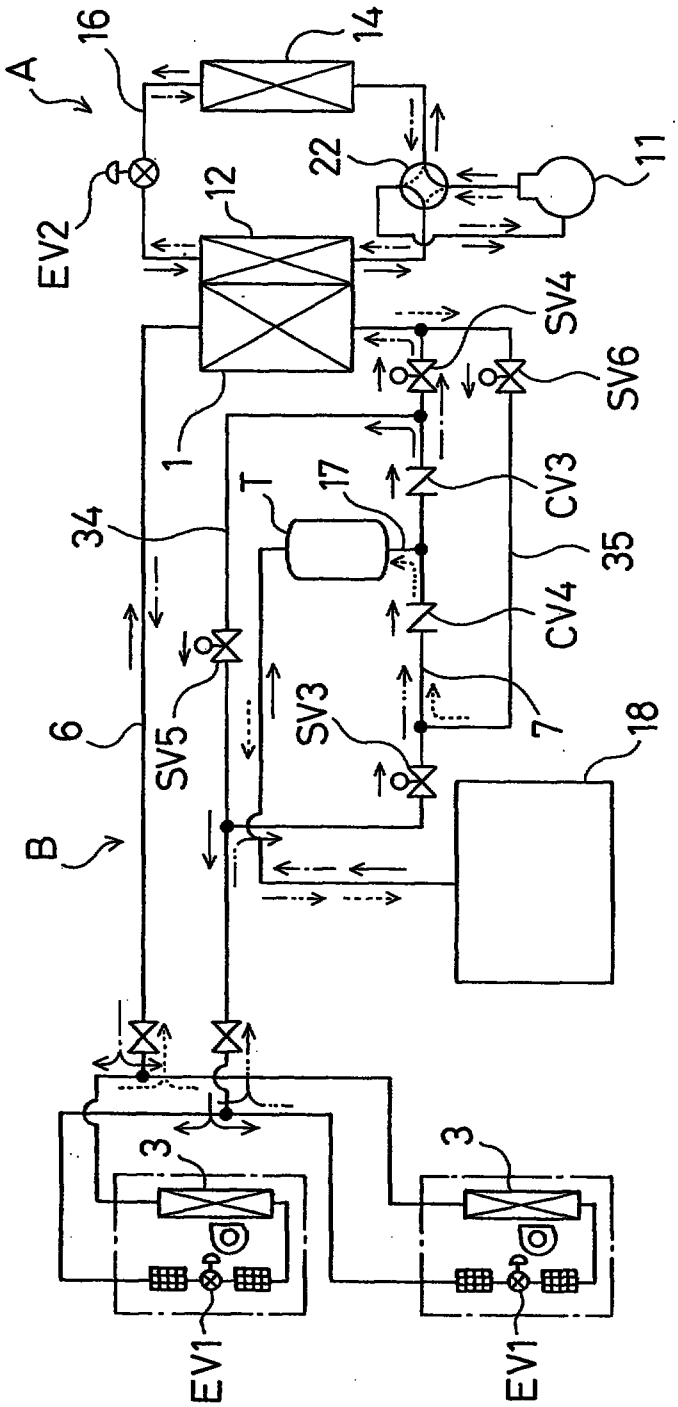

A
B
16
14
22
11
EV2
12
SV4
SV6
1
17
T
CV3
34
35
CV4
SV5
SV3
6
7
18
3
3
EV1
EV1

Fig. 10

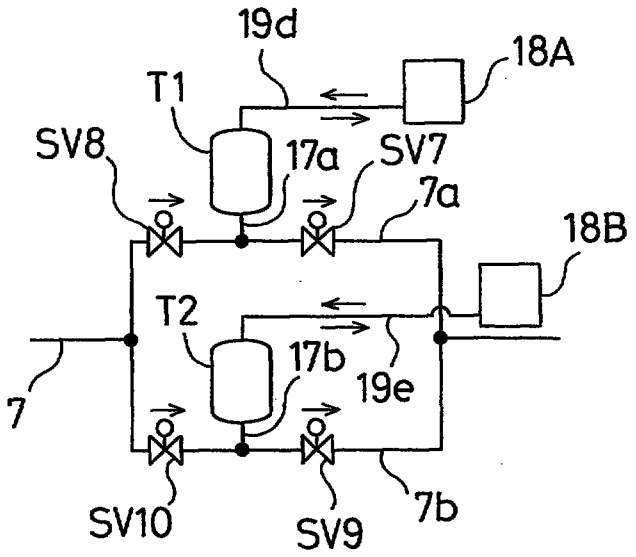

19d
18A
T1
SV8
17a
SV7
7a
18B
T2
17b
19e
7
SV10
SV9
7b

Fig. 11

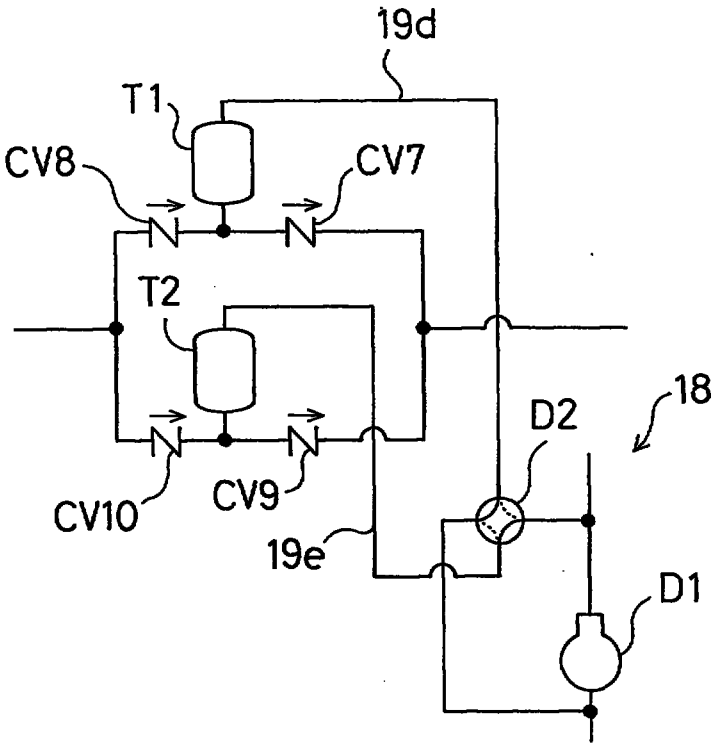

19d
T1
CV8
CV7
T2
18
D2
CV10
CV9
19e
D1

Fig. 12

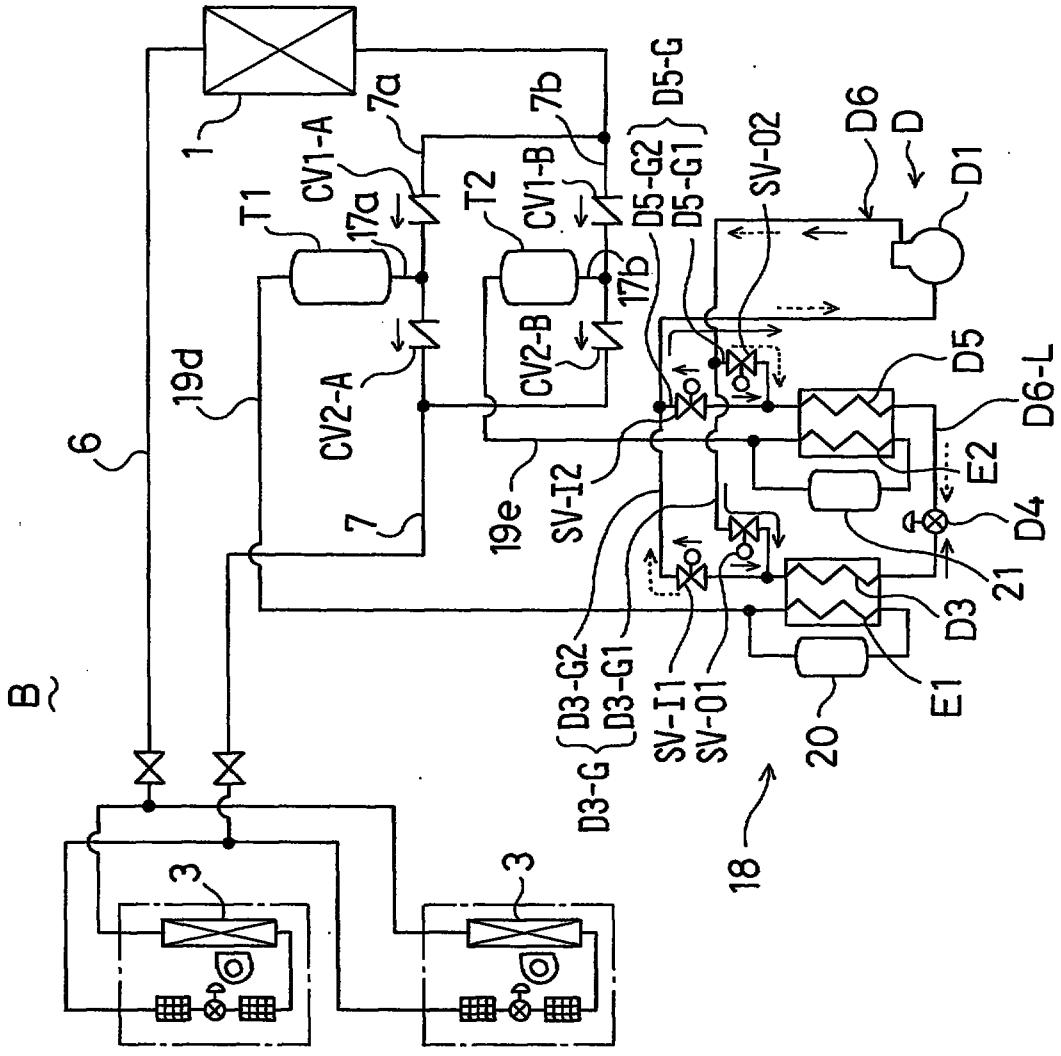

B
6
19d
1
7a
7b
CV1-A
CV1-B
T1
T2
17a
17b
CV2-A
CV2-B
7
19e
SV-I2
D5-G
D5-G2
D5-G1
SV-02
D6
D
D1
D5
D6-L
E2
D4
21
D3
E1
20
18
D3-G
D3-G2
D3-G1
SV-I1
SV-01
3
3

Fig. 13

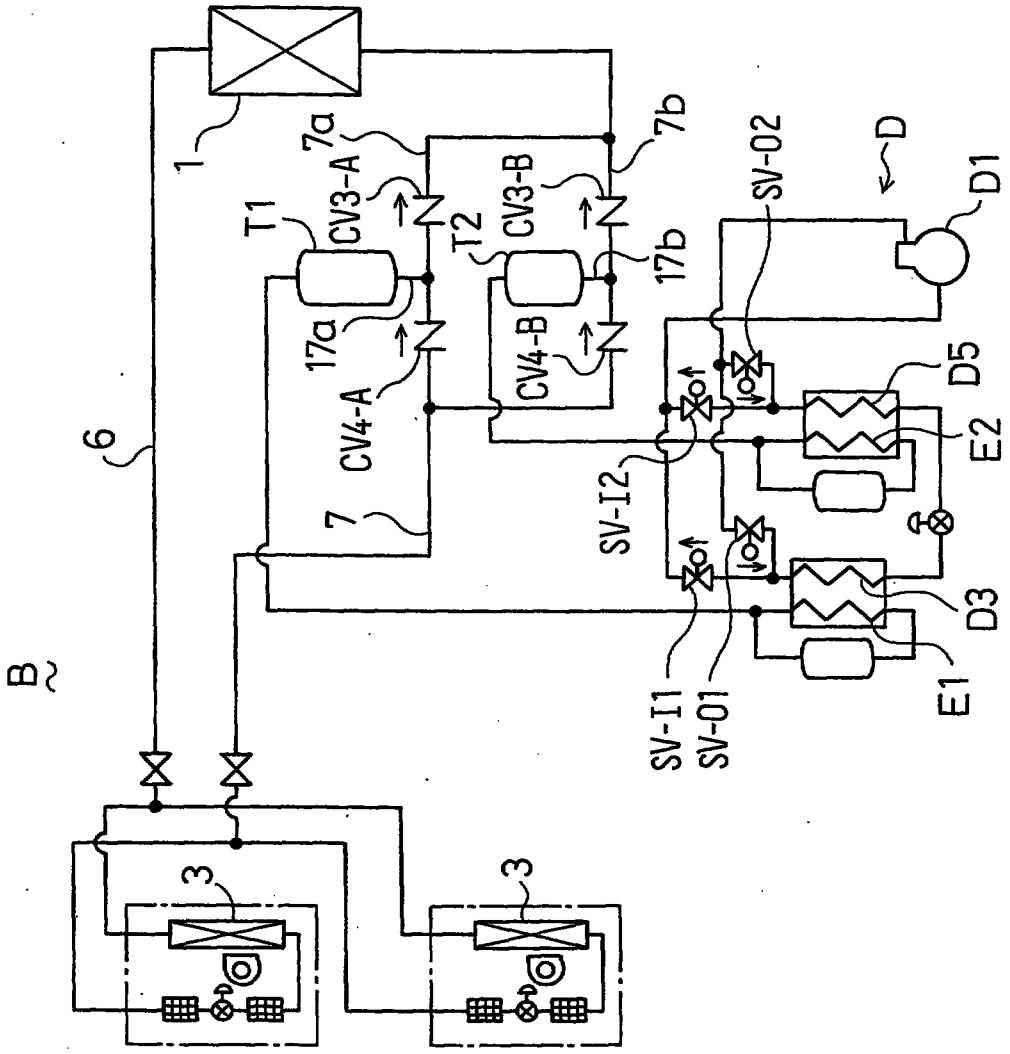

1
7a
7b
T1
CV3-A
T2
CV3-B
SV-02
D
D1
17b
17a
CV4-A
CV4-B
D5
6
E2
7
SV-I2
B
D3
E1
SV-I1
SV-01
3
3

Fig. 14

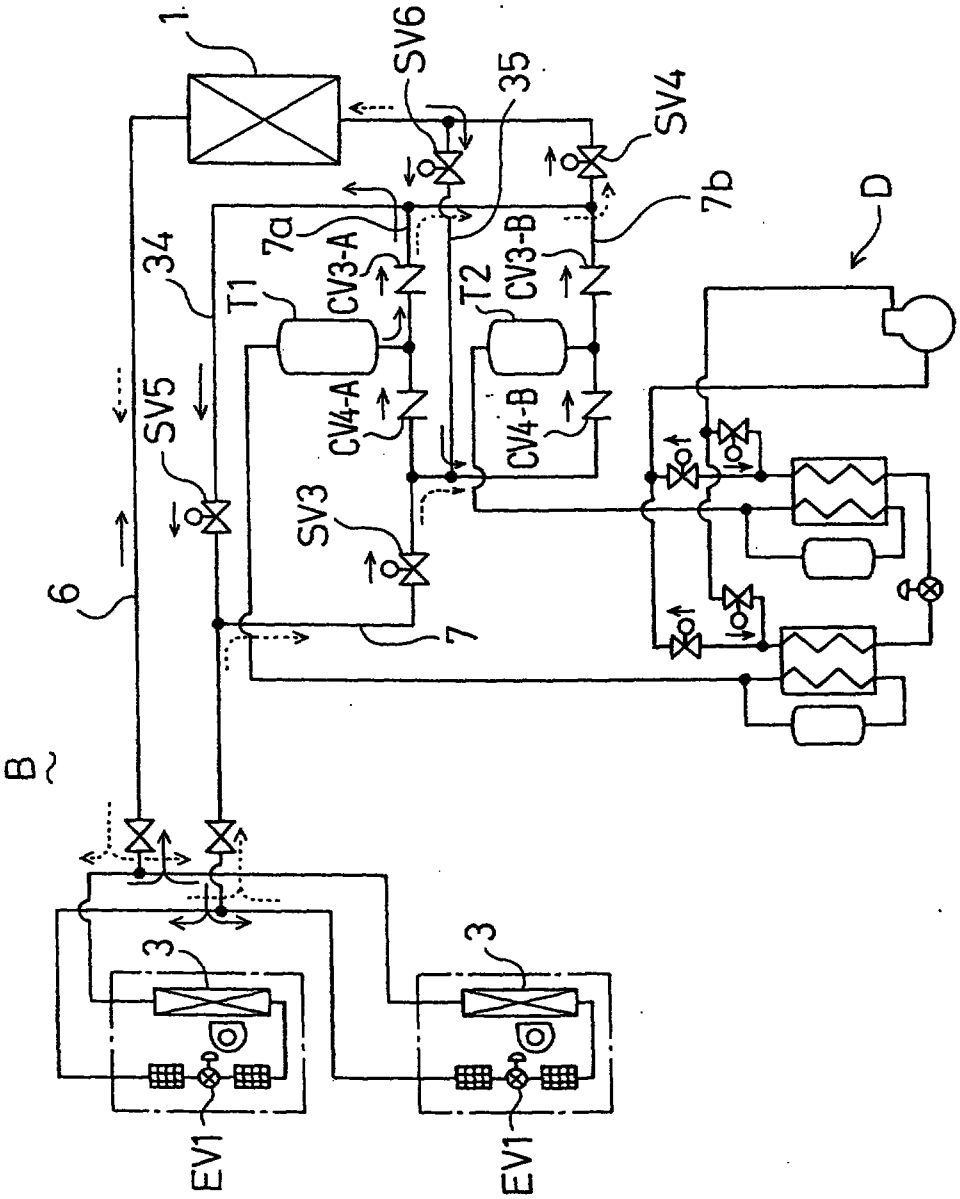

1
SV6
35
SV4
7b
D
34
7a
CV3-A
T1
T2
CV3-B
SV5
CV4-A
CV4-B
SV3
6
7
B
3
3
EV1
EV1

Fig. 15

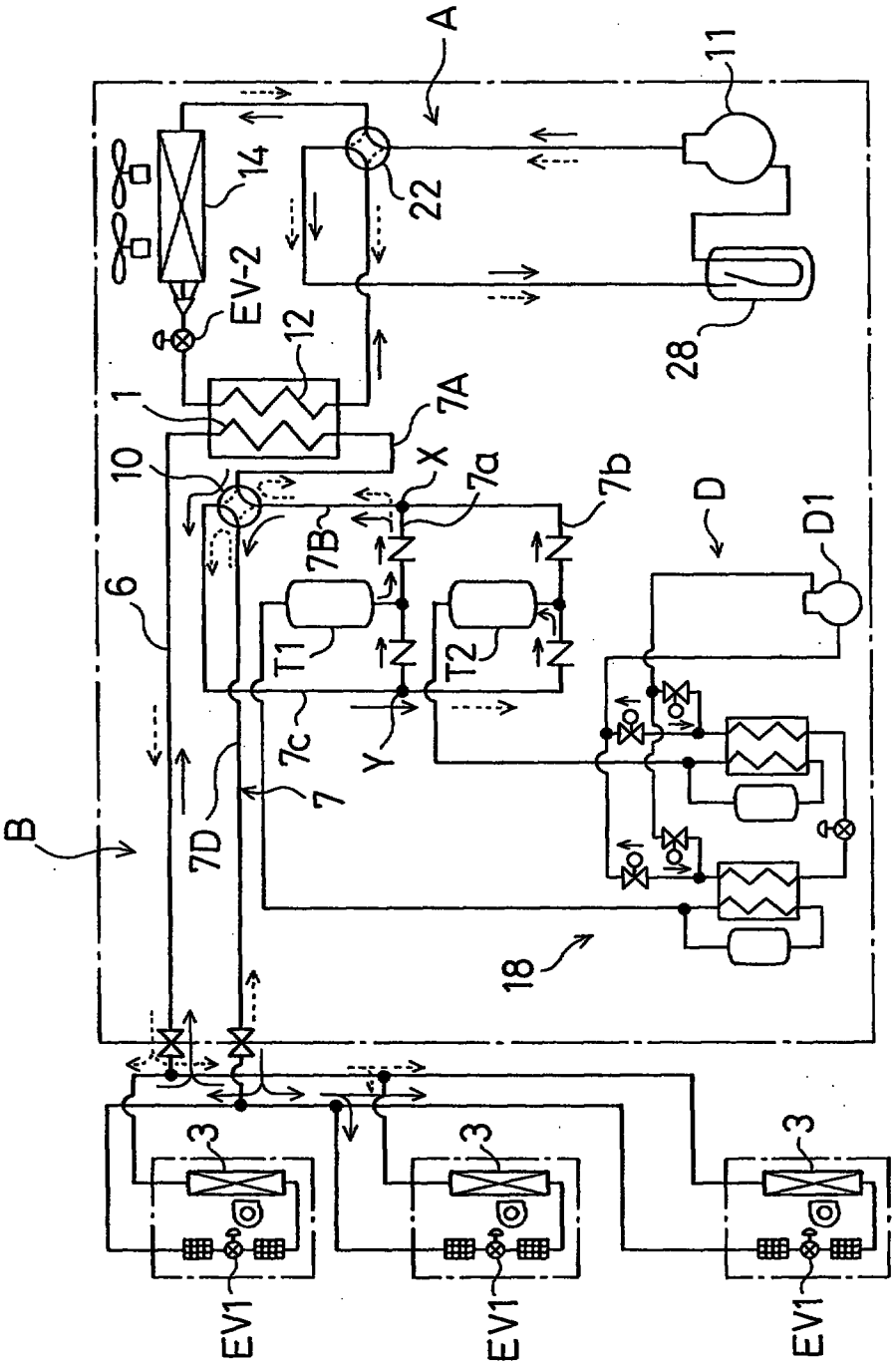

A
11
14
22
EV-2
12
28
1
7A
10
X
7a
7b
D
D1
7B
6
T1
T2
7c
Y
7
7D
B
18
3
3
3
EV1
EV1
EV1

Fig. 16

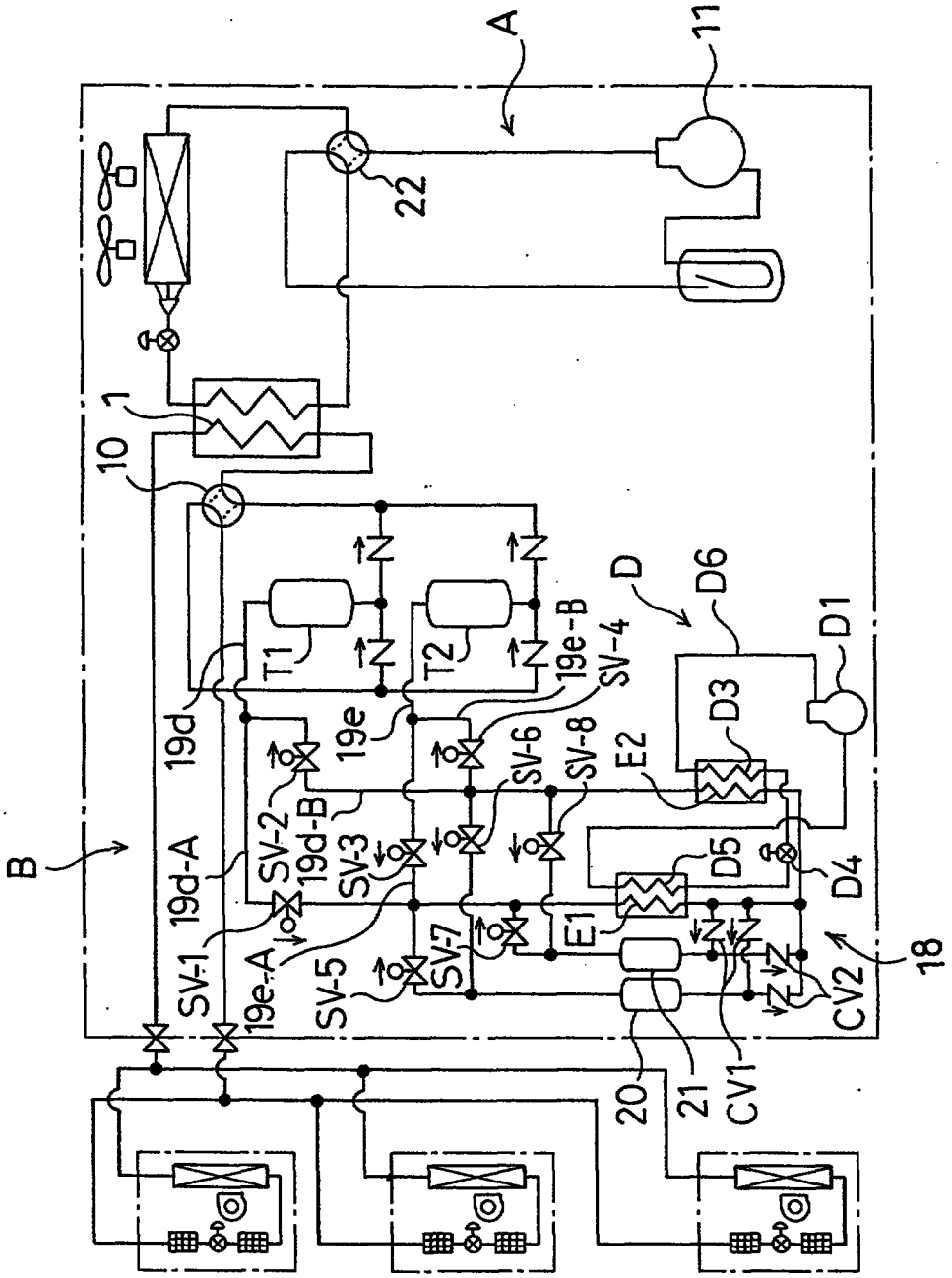

A
11
22
1
10
T1
T2
19e-B
SV-4
D
D6
D1
19d
19e
SV-6
SV-8
D3
E2
B
19d-A
SV-2
19d-B
SV-3
D5
D4
SV-1
19e-A
SV-5
SV-7
E1
CV2
18
20
21
CV1

Fig. 17

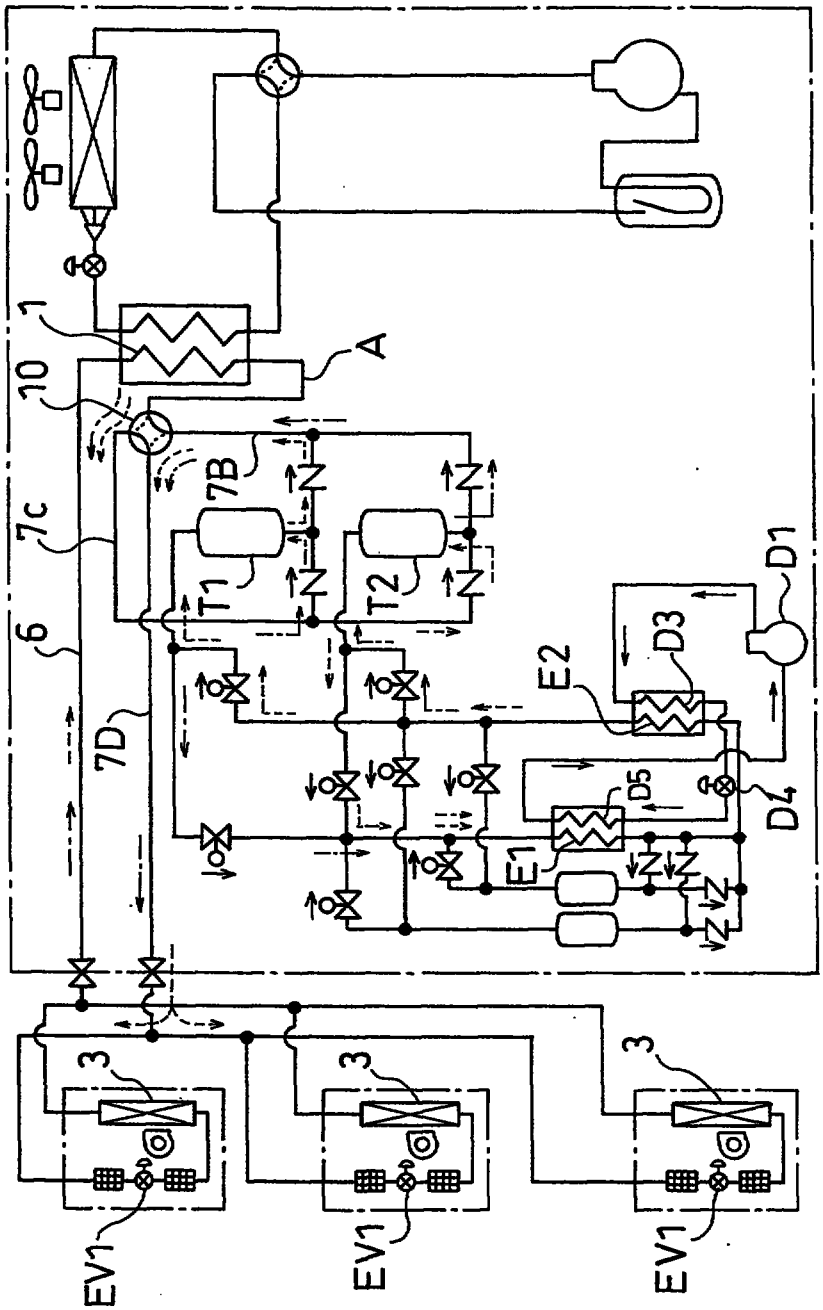


Fig. 18

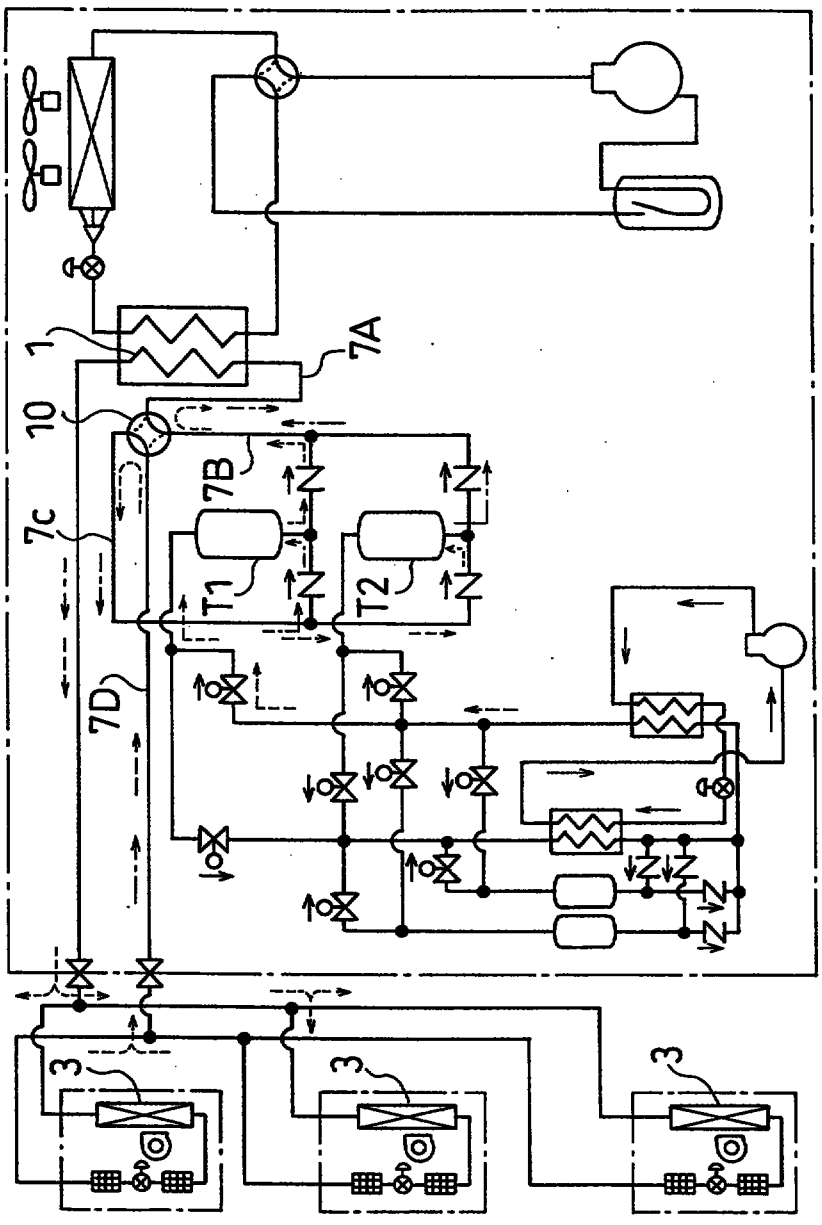


Fig. 19

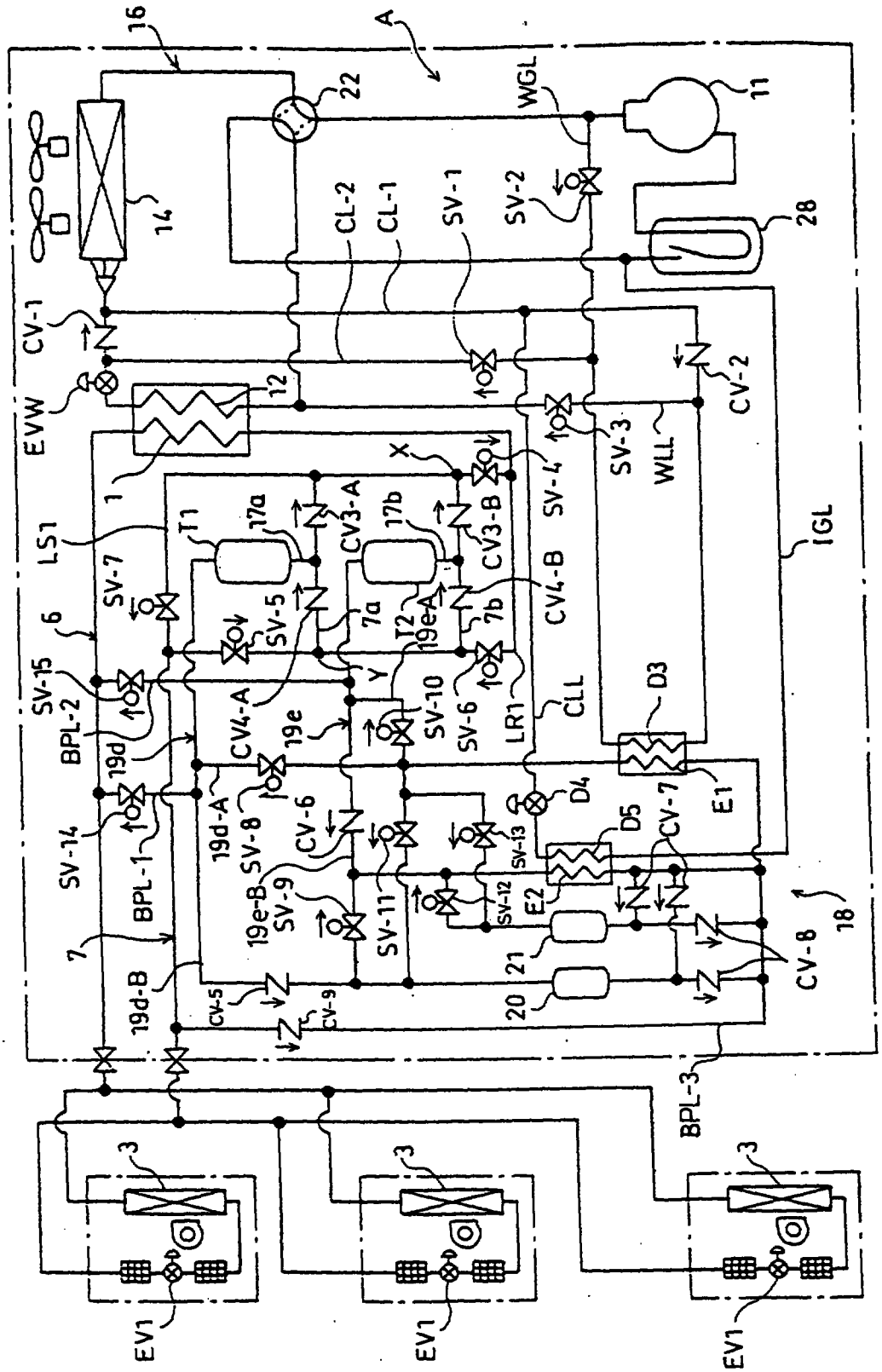

A
16
22
WGL
11
14
CL-2
CL-1
SV-1
SV-2
28
CV-1
EVW
12
CV-2
SV-3
WLL
X
SV-4
1
LS1
T1
17a
CV3-A
17b
CV3-B
CV4-B
IGL
SV-7
SV-5
7a
I2
19e-A
7b
6
SV-15
BPL-2
19d
CV4-A
19e
SV-10
SV-6
LR1
CLL
D3
Y
19d-A
SV-8
CV-6
D4
E1
SV-14
BPL-1
19e-B
SV-9
SV-11
SV-13
SV-12
D5
CV-7
E2
7
19d-B
CV-5
CV-9
21
20
CV-8
18
BPL-3
3
EV1

Fig. 20

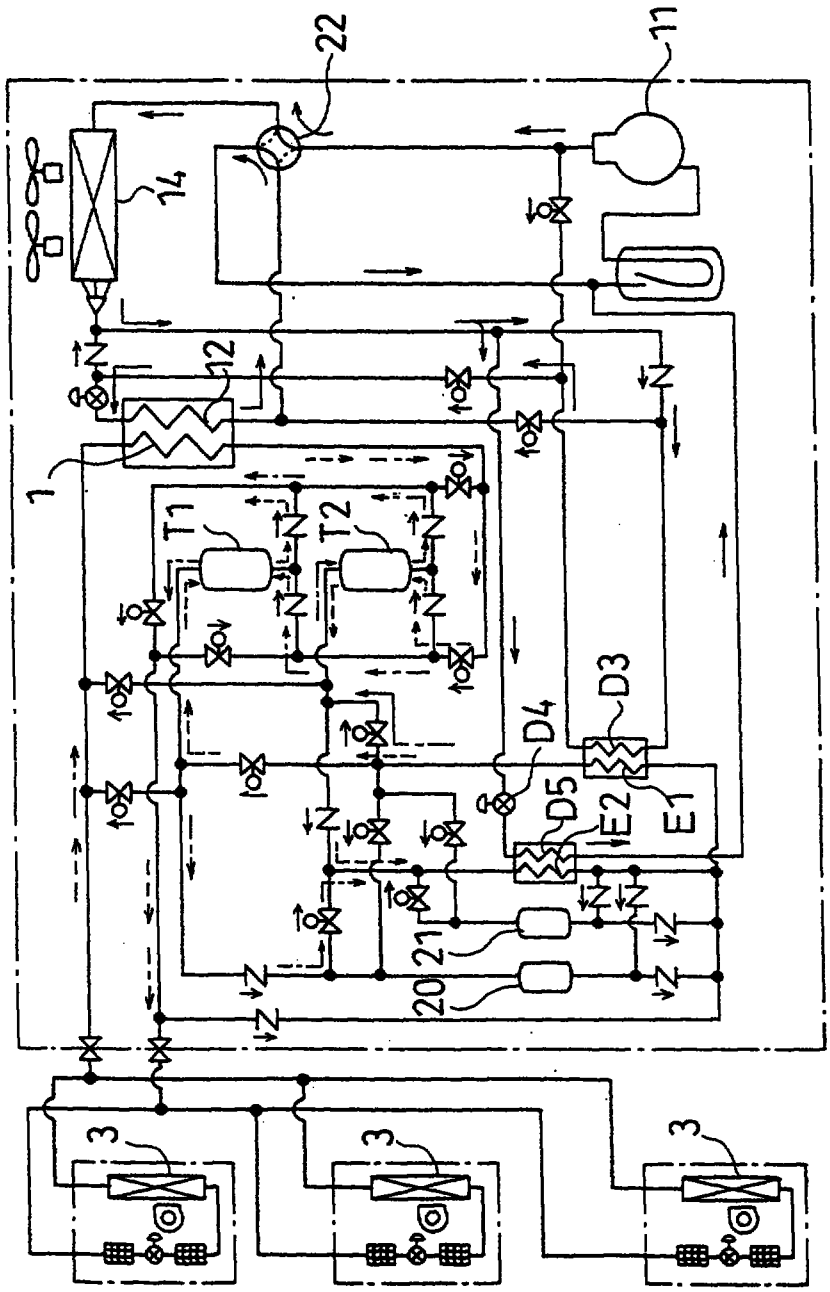


Fig. 21

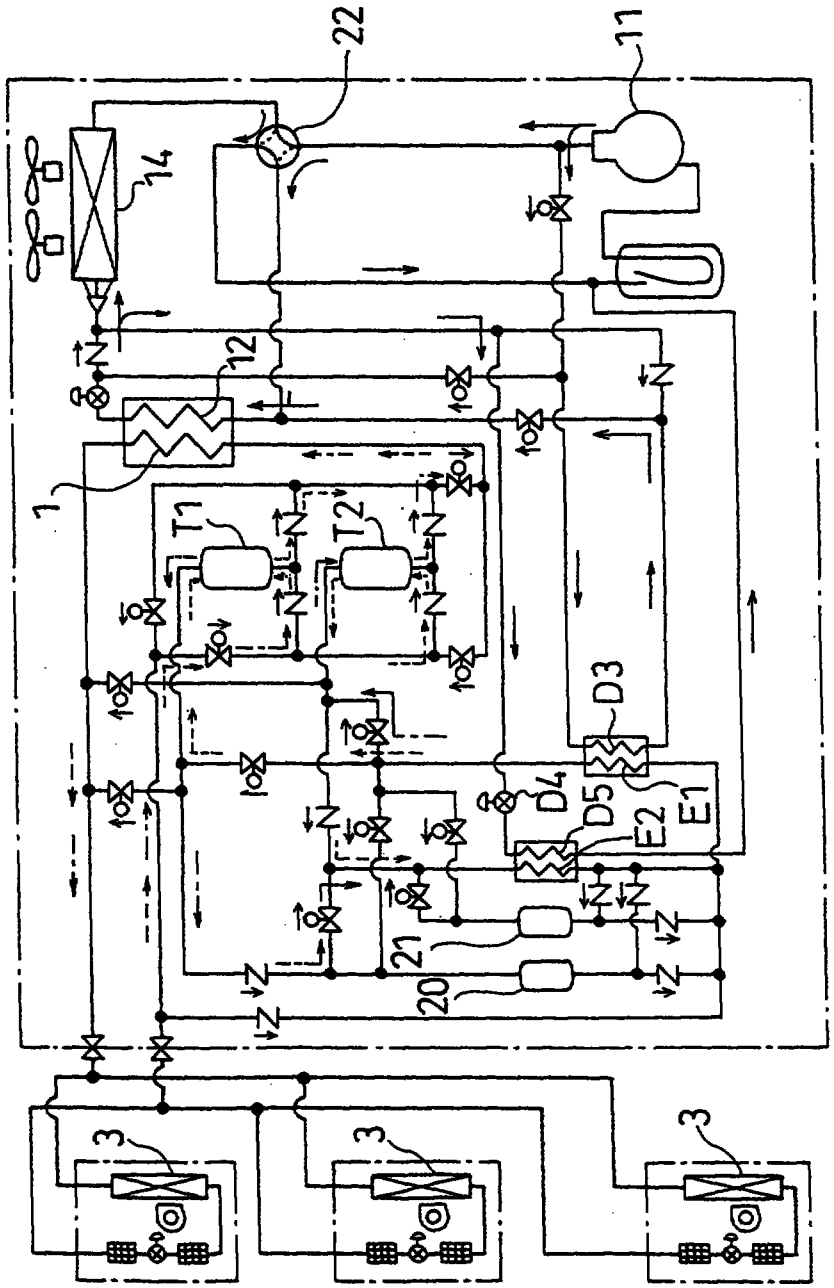


Fig. 22

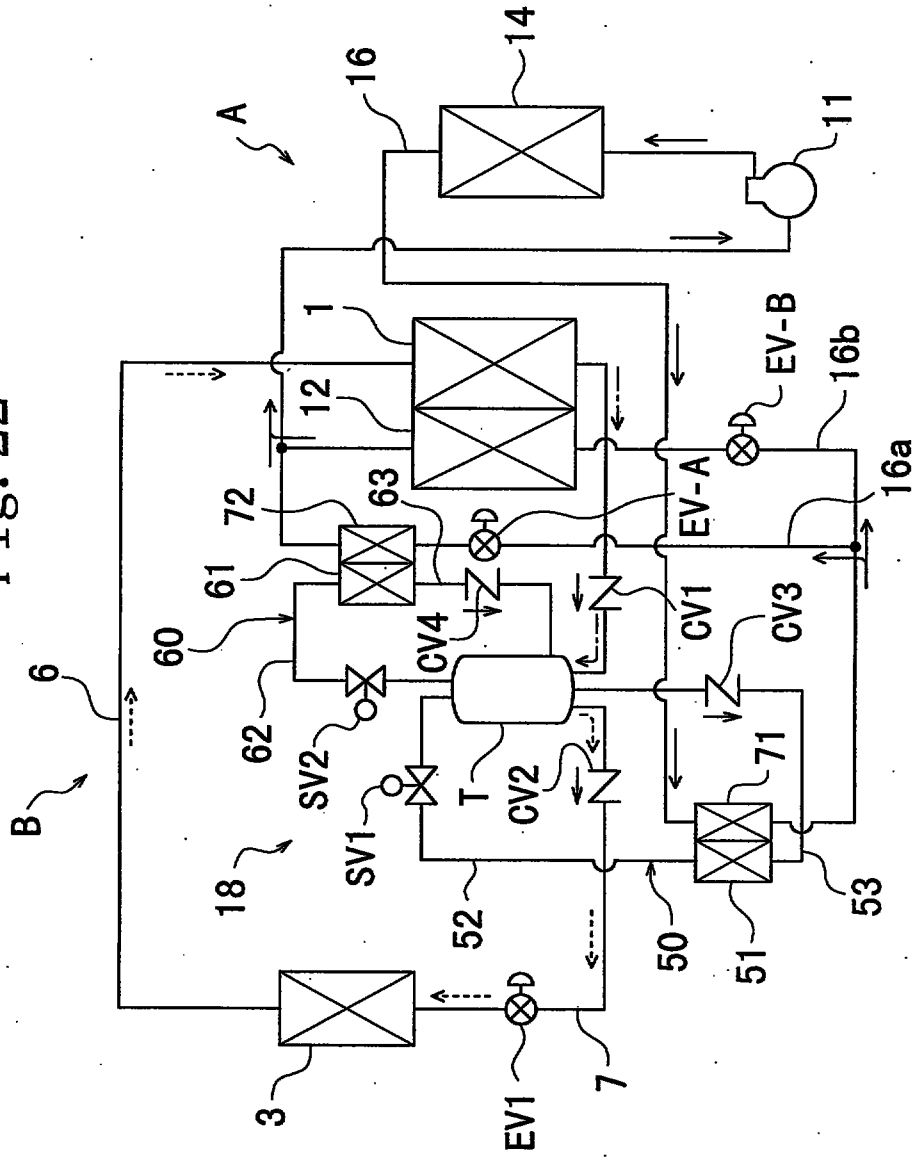

A
B
1
3
6
7
11
12
14
16
16a
16b
18
50
51
52
53
60
61
62
63
71
72
CV1
CV2
CV3
CV4
EV-A
EV-B
EV1
SV1
SV2
T

Fig. 23

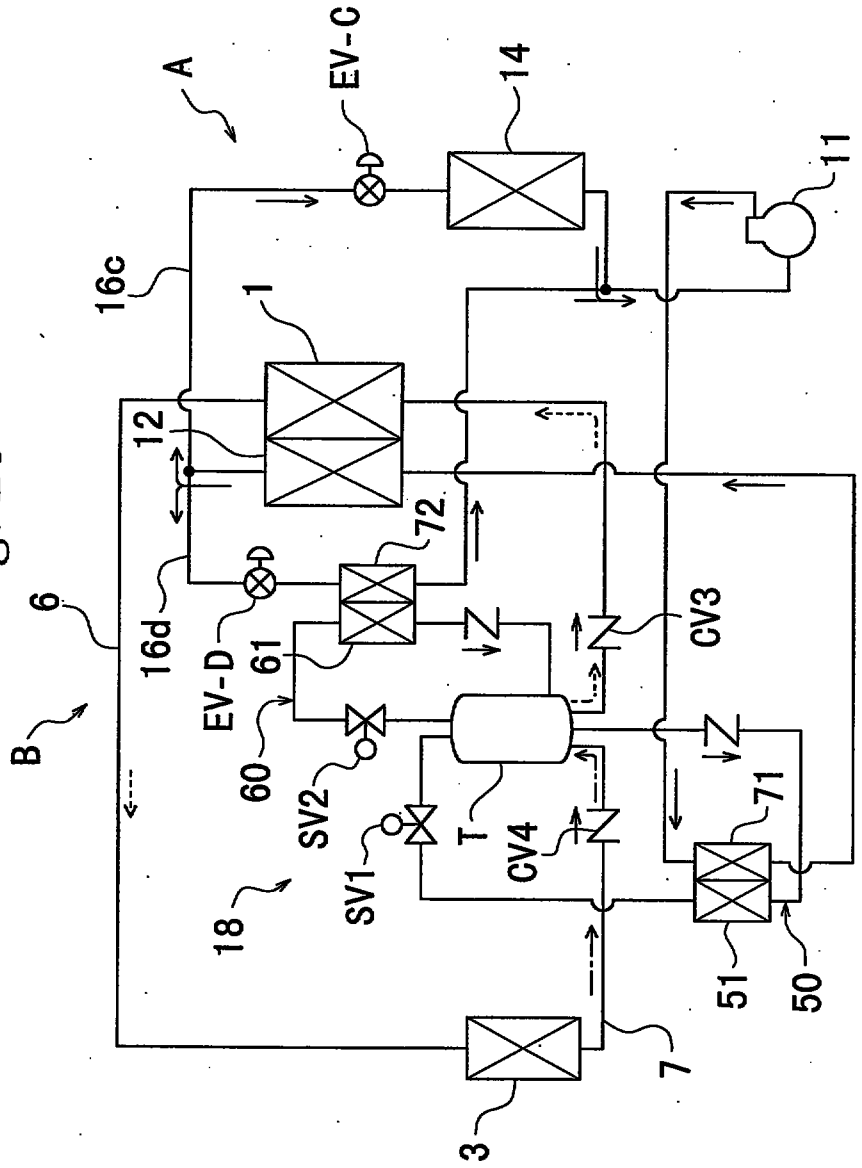

A
EV-C
14
11
16c
1
12
72
6
16d
EV-D
61
CV3
B
60
SV2
T
CV4
71
SV1
18
51
50
7
3

Fig. 24

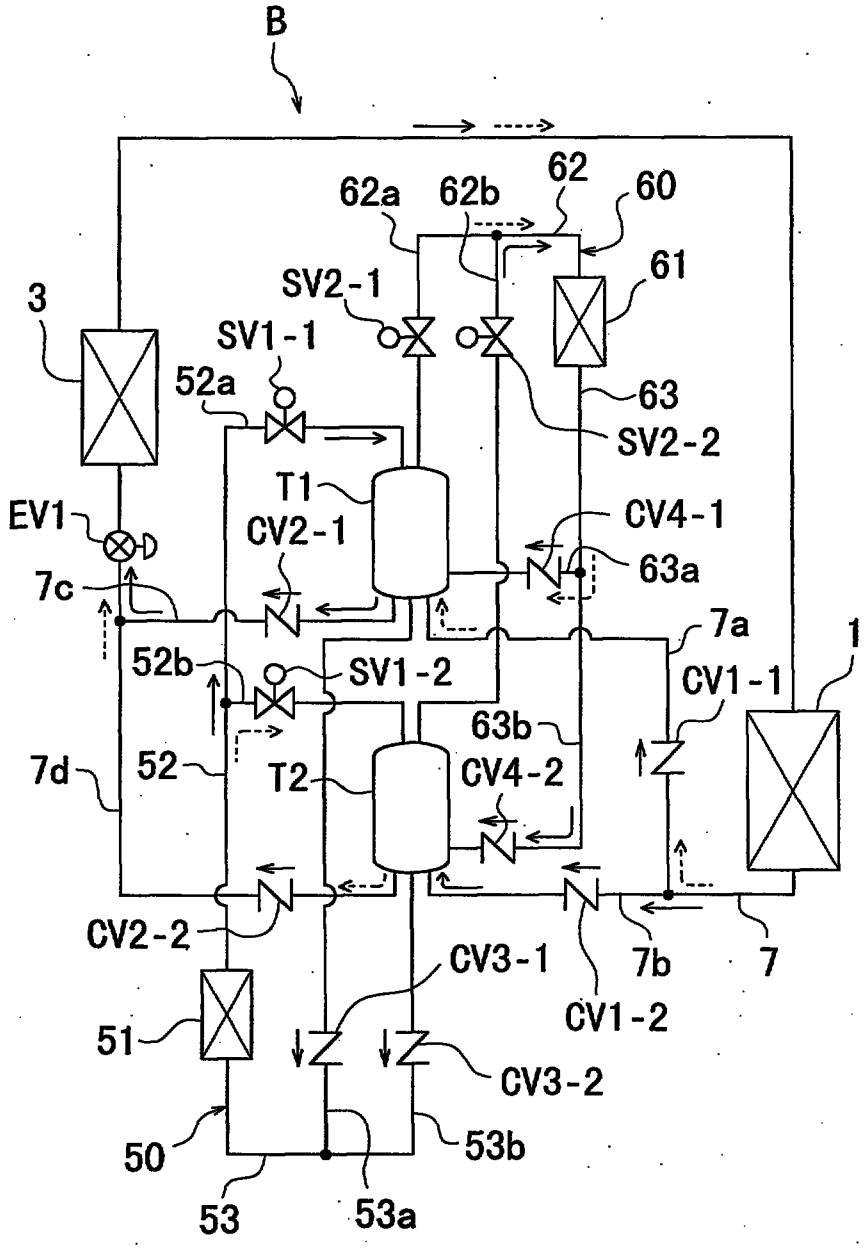

B
62a
62b
62
60
61
3
SV2-1
SV1-1
52a
63
SV2-2
T1
EV1
CV2-1
CV4-1
63a
7c
7a
1
52b
SV1-2
CV1-1
63b
7d
52
T2
CV4-2
CV2-2
CV3-1
7b
7
CV1-2
51
CV3-2
50
53b
53
53a

Fig. 25

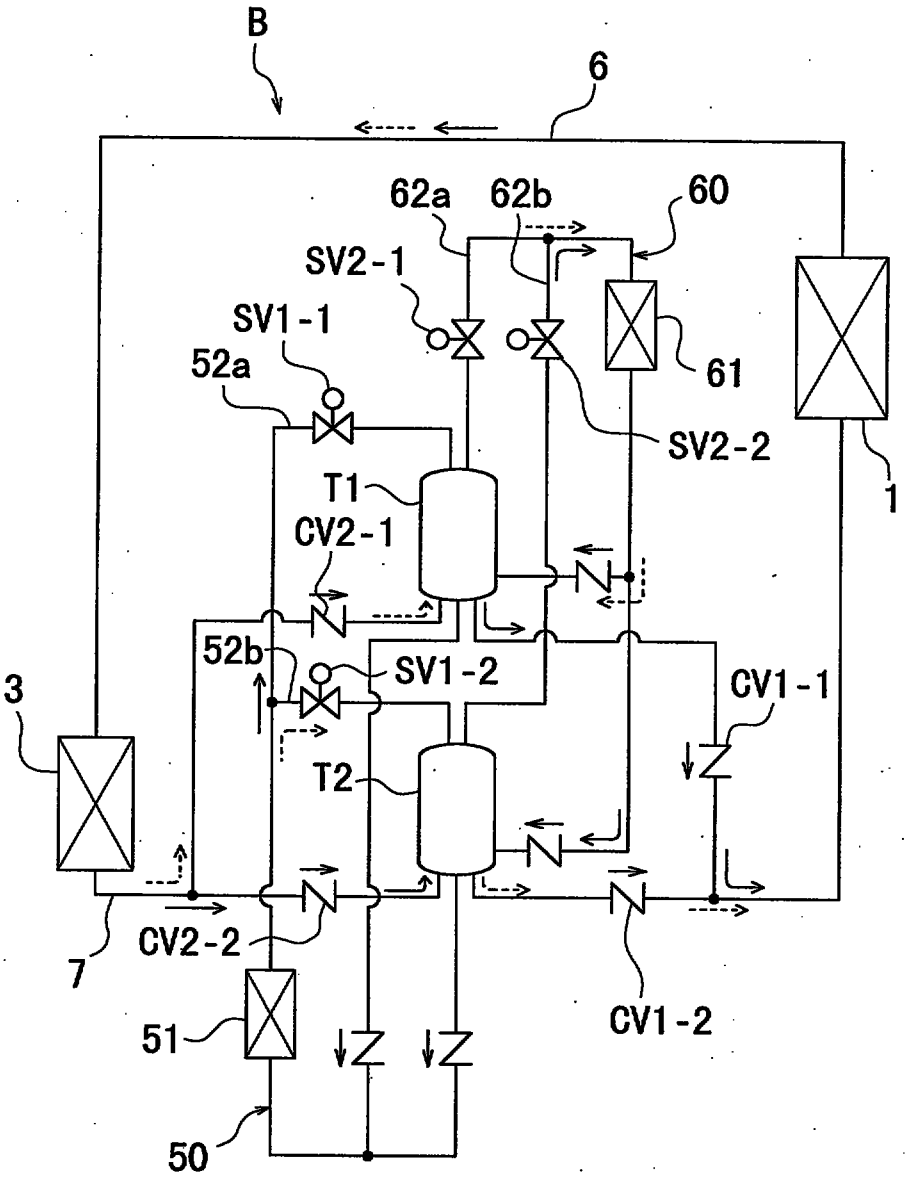

B
6
62a
62b
60
SV2-1
SV1-1
52a
61
SV2-2
1
T1
CV2-1
52b
SV1-2
3
CV1-1
T2
7
CV2-2
CV1-2
51
50

Fig. 26

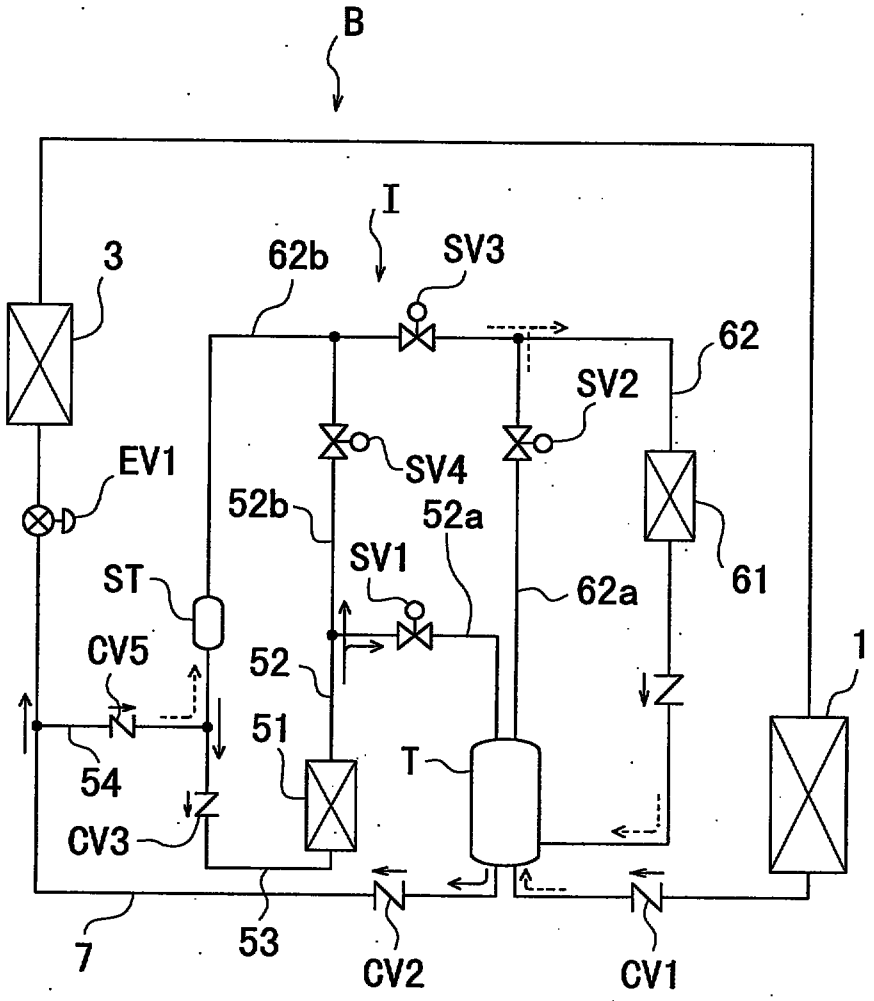

B
I
3
62b
SV3
62
SV2
SV4
EV1
52b
52a
SV1
61
ST
62a
CV5
52
1
51
54
T
CV3
7
53
CV2
CV1

Fig. 27

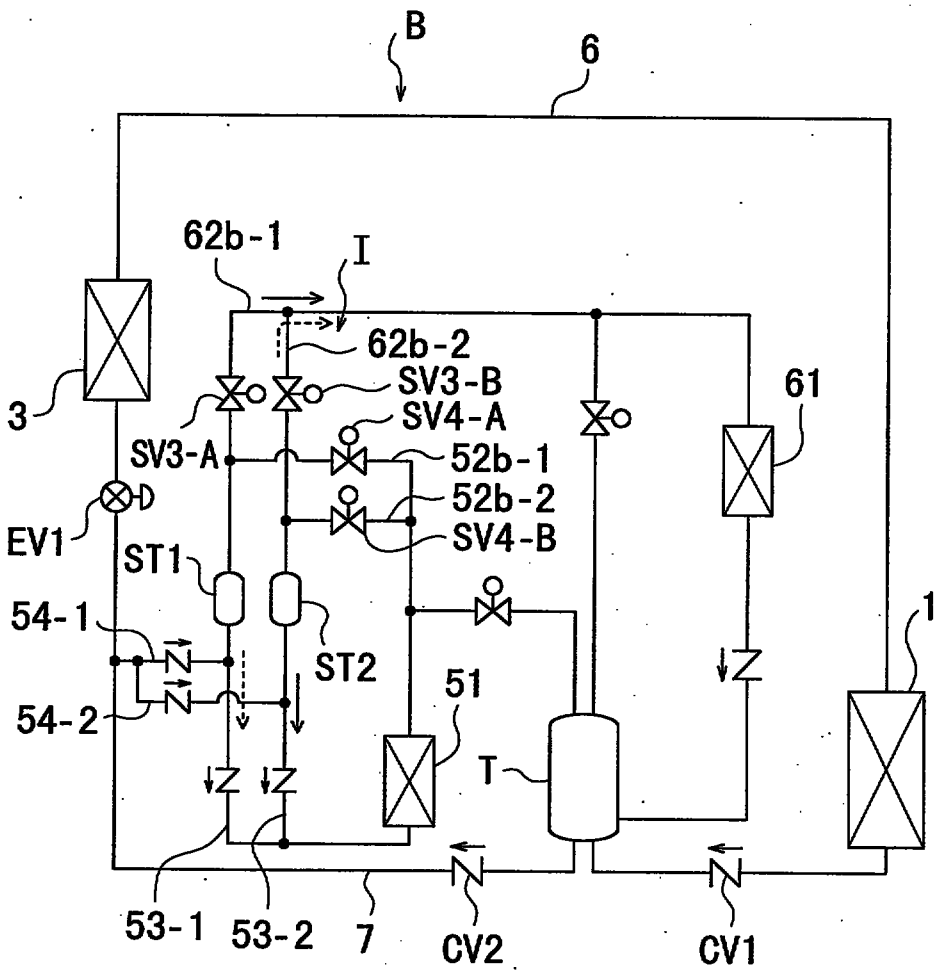

B
6
62b-1
I
62b-2
SV3-B
SV4-A
SV3-A
52b-1
52b-2
SV4-B
3
61
EV1
ST1
54-1
ST2
1
54-2
51
T
53-1
53-2
7
CV2
CV1

Fig. 28

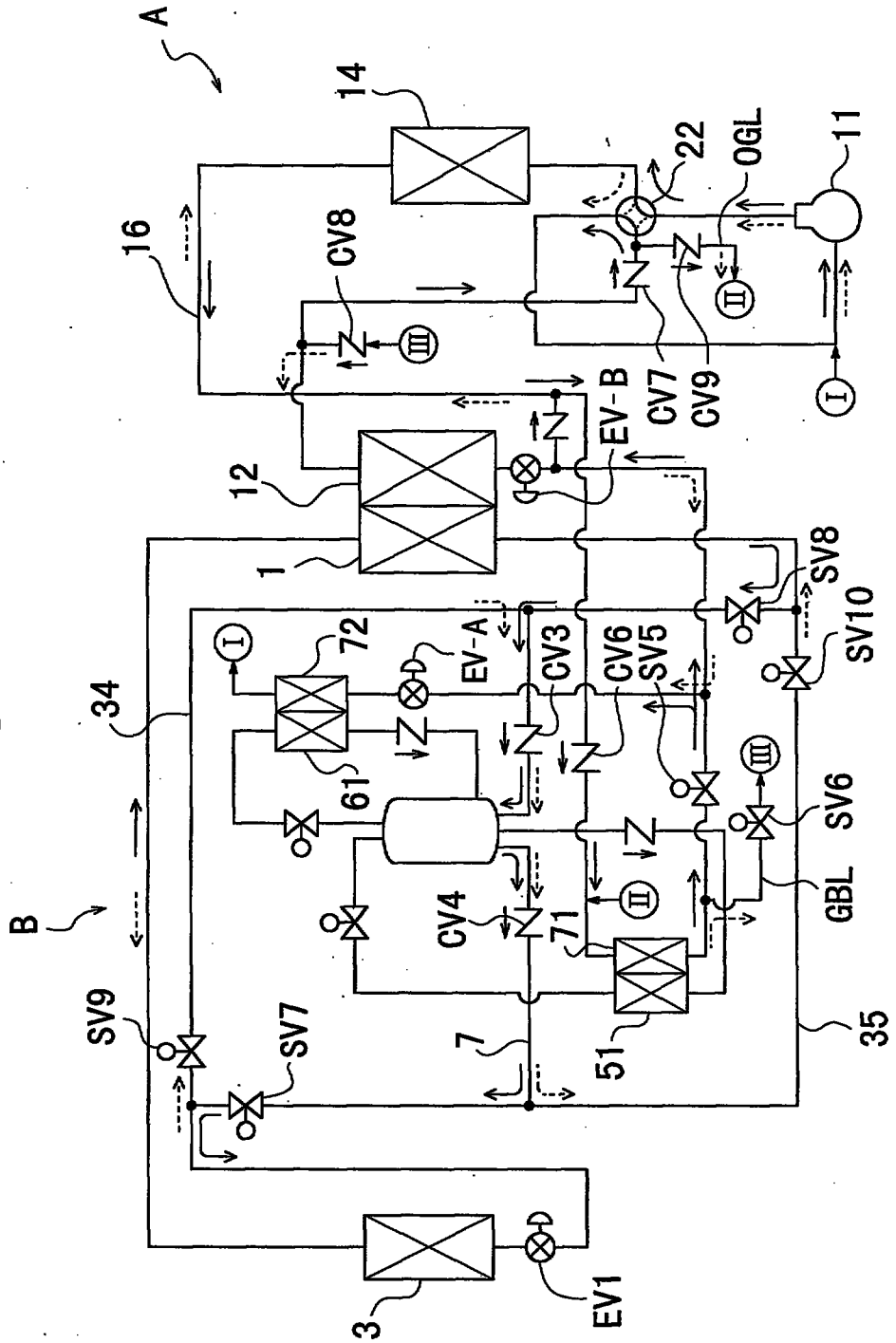

A
B
14
22
OGL
11
16
CV8
CV7
CV9
EV-B
12
1
SV8
SV10
72
EV-A
CV3
CV6
SV5
34
61
SV6
GBL
CV4
71
SV9
SV7
7
51
35
3
EV1

Fig. 29

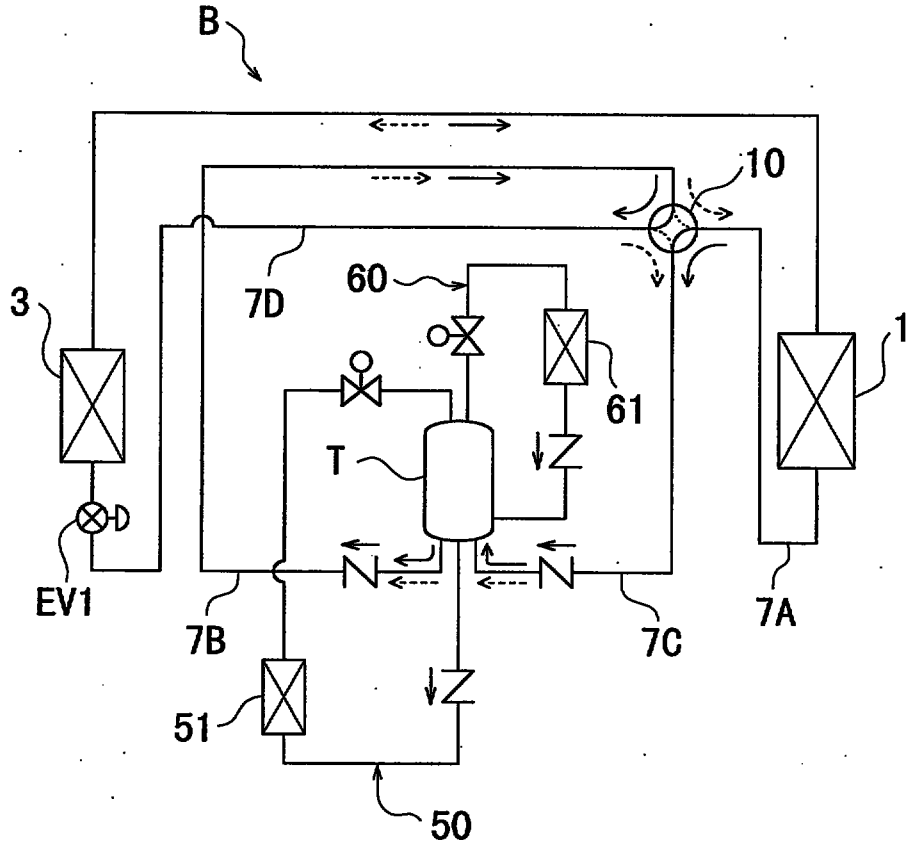


Fig. 30

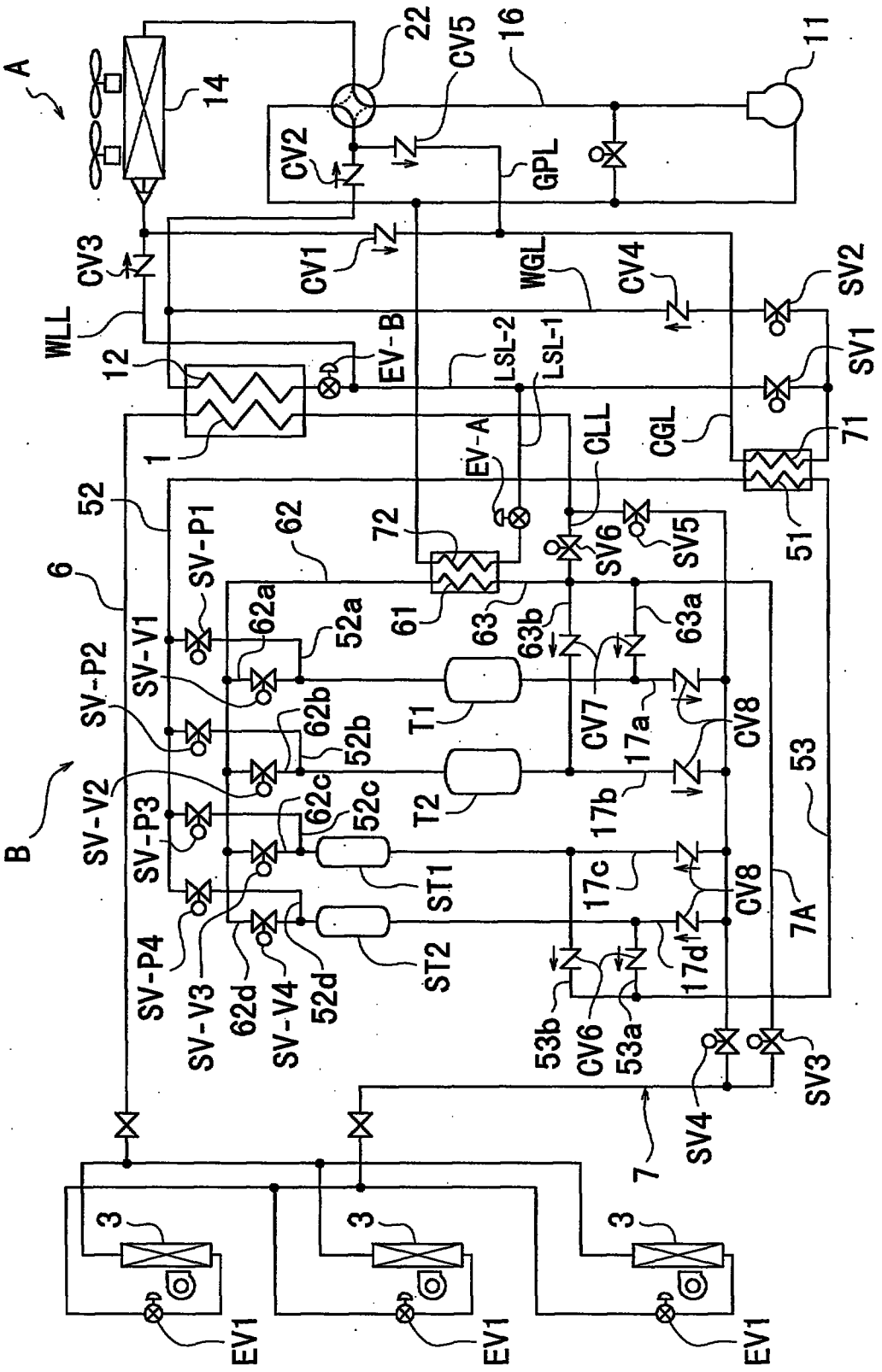

A
B
14
22
CV5
16
11
CV2
GPL
CV3
CV1
WGL
CV4
SV2
WLL
12
EV-B
LSL-2
LSL-1
SV1
1
EV-A
CLL
CGL
71
52
SV-P1
62
72
SV6
SV5
51
6
SV-V1
62a
52a
61
63
63b
63a
SV-P2
62b
52b
T1
CV7
17a
CV8
SV-V2
T2
53
SV-P3
62c
52c
17b
ST1
17c
CV8
7A
SV-P4
ST2
17d
SV-V3
62d
SV-V4
52d
53b
CV6
53a
SV3
SV4
7
3
EV1

Fig. 31

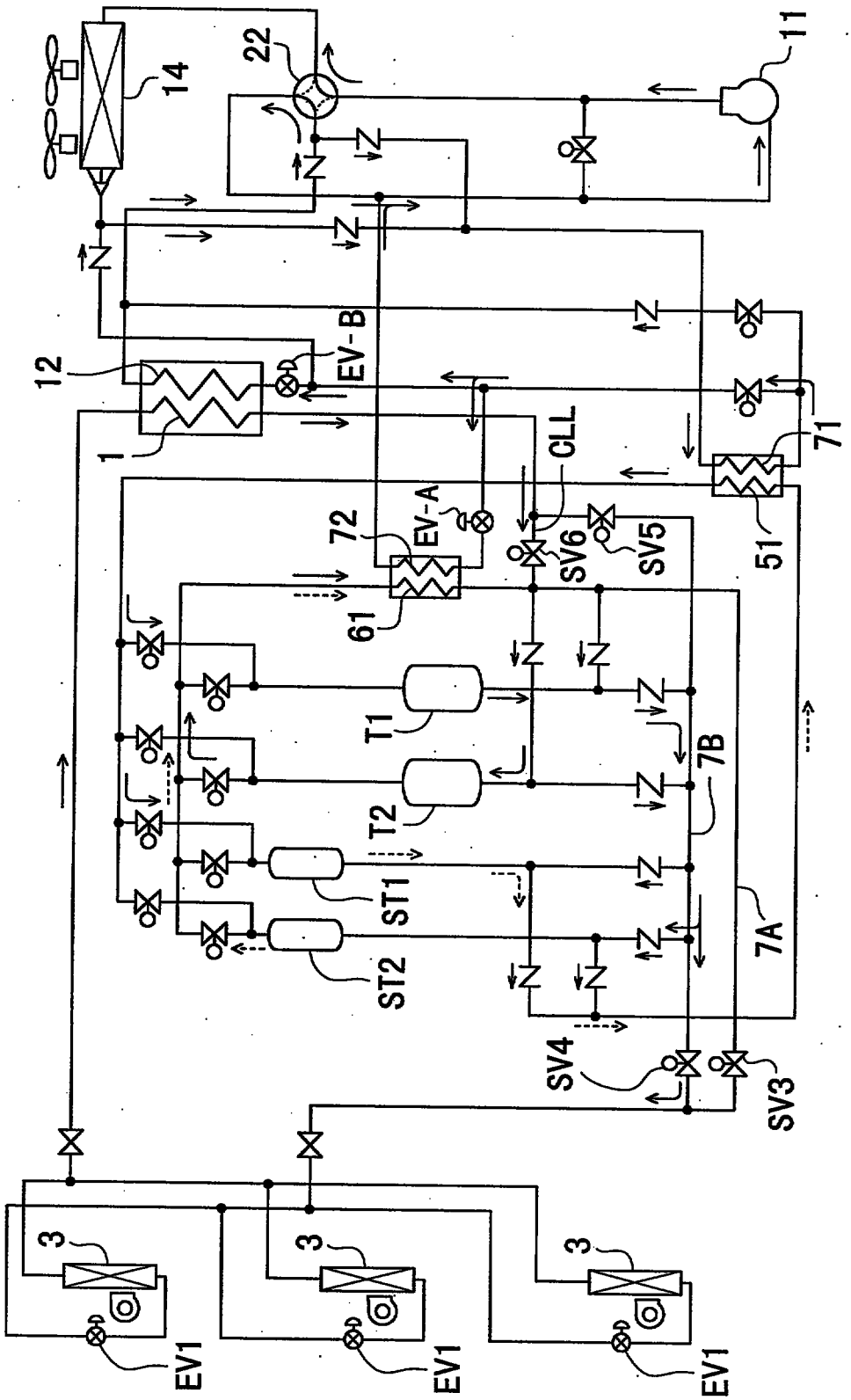

14
22
11
EV-B
12
1
71
CLL
72
EV-A
SV6
SV5
51
61
T1
7B
T2
ST1
ST2
7A
SV4
SV3
3
EV1

Fig. 32

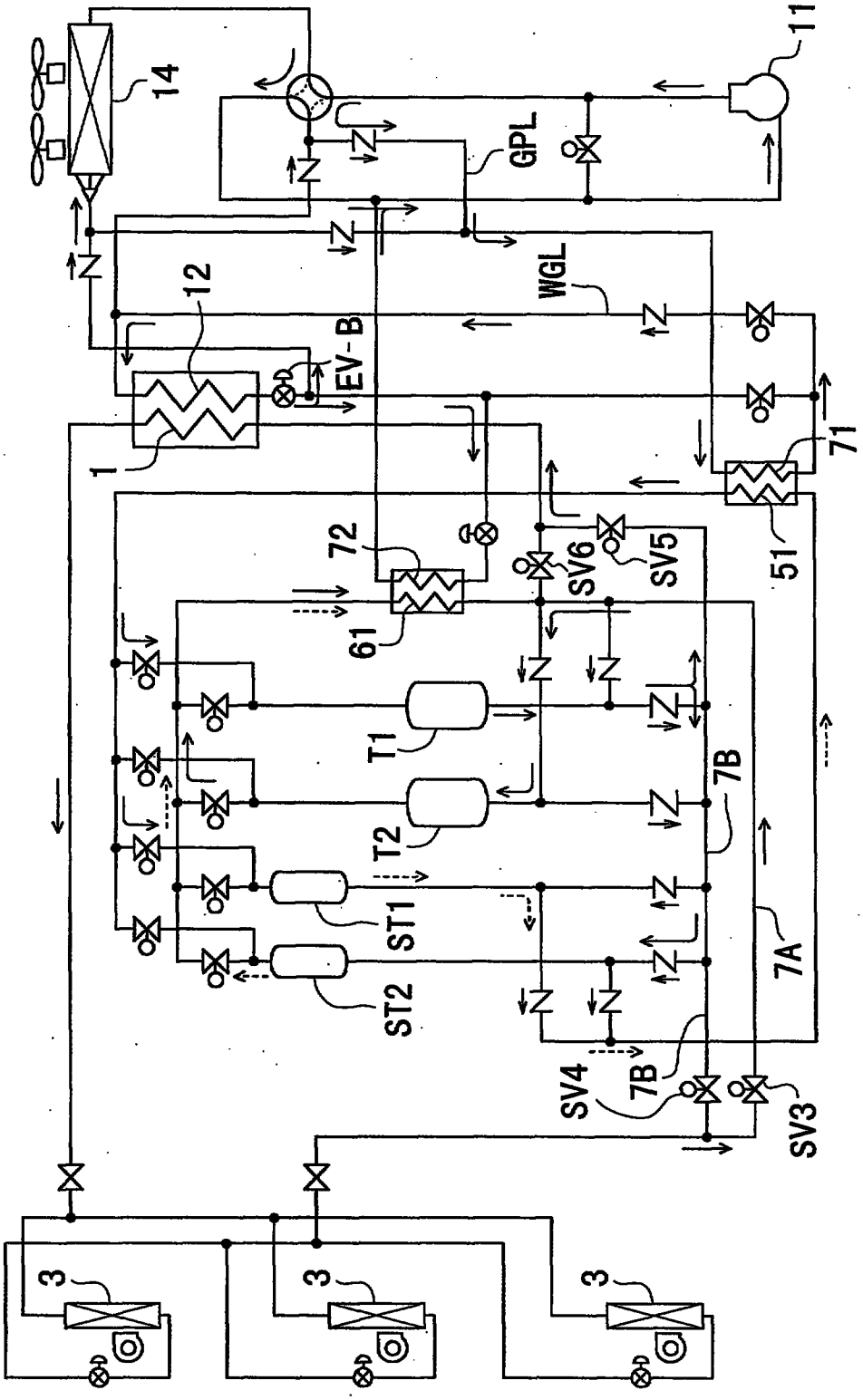

14
11
GPL
WGL
12
EV-B
1
71
72
SV6
SV5
51
61
T1
T2
7B
ST1
ST2
7A
SV4
7B
SV3
3
3
3

Fig. 33

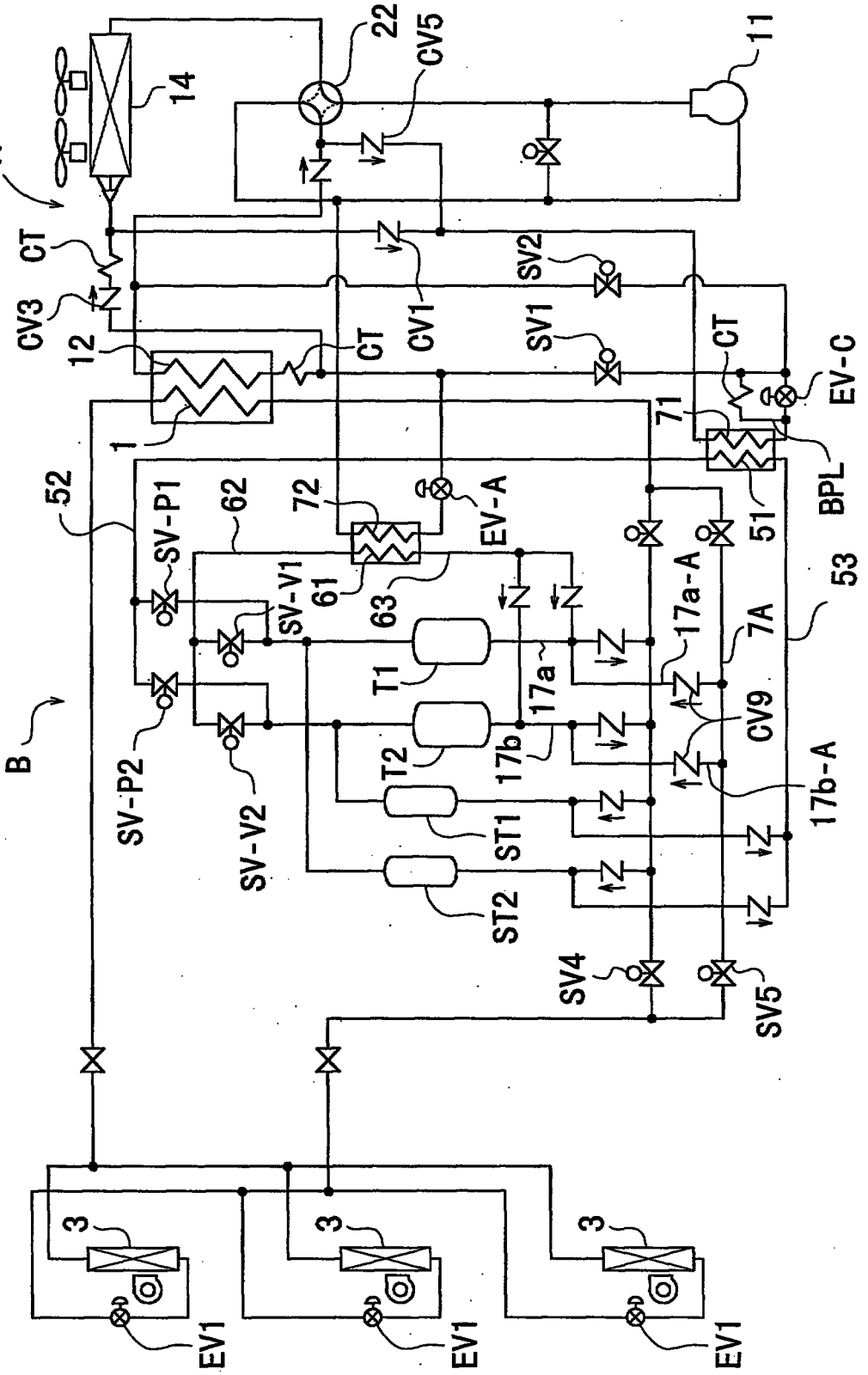

A
B
14
22
CV5
11
CT
CV3
12
1
CV1
SV2
SV1
CT
EV-C
71
BPL
51
52
SV-P1
62
72
EV-A
61
63
SV-V1
T1
T2
ST1
ST2
17a
17b
17a-A
7A
53
CV9
17b-A
SV-P2
SV-V2
SV4
SV5
3
EV1

Fig. 34

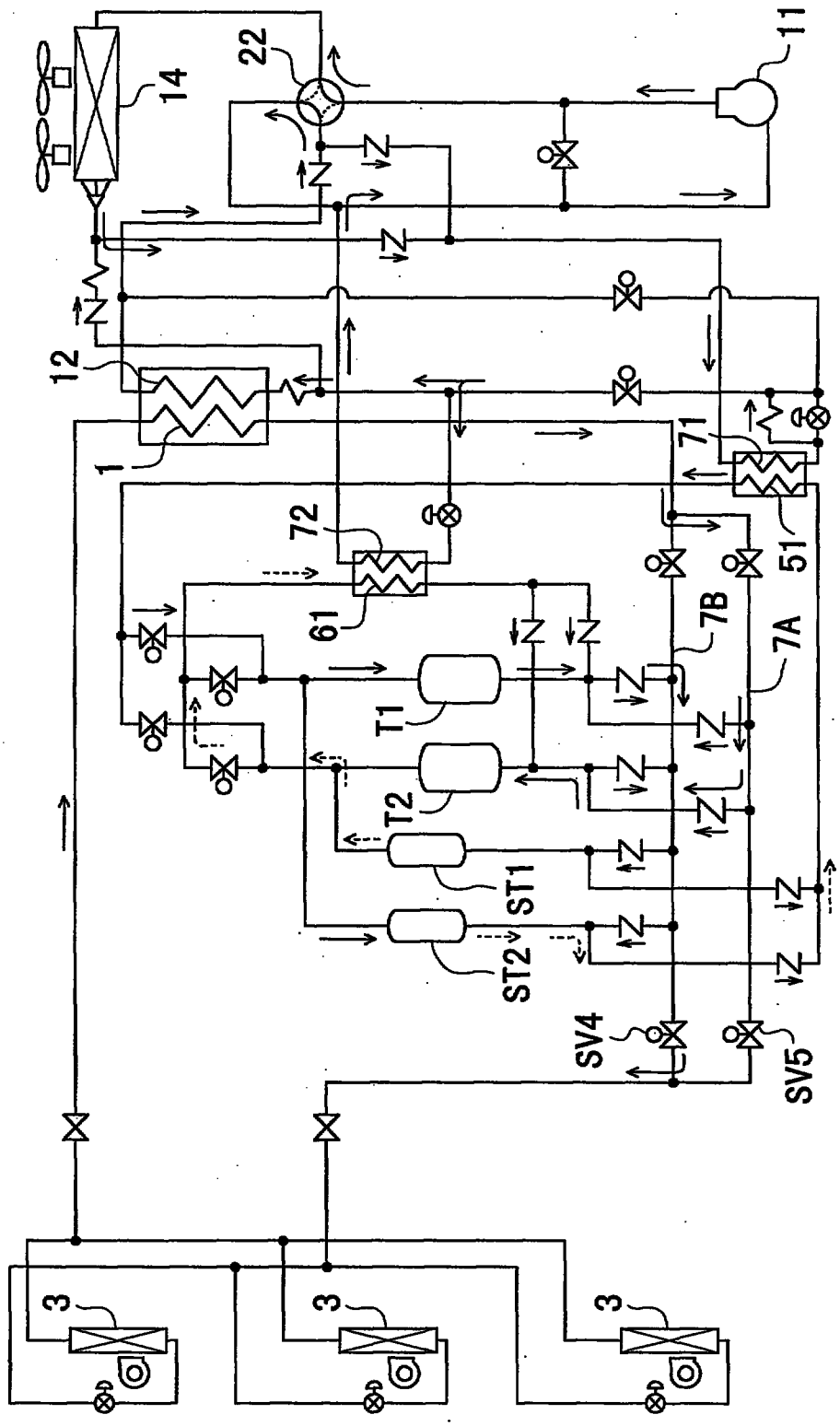


Fig. 35

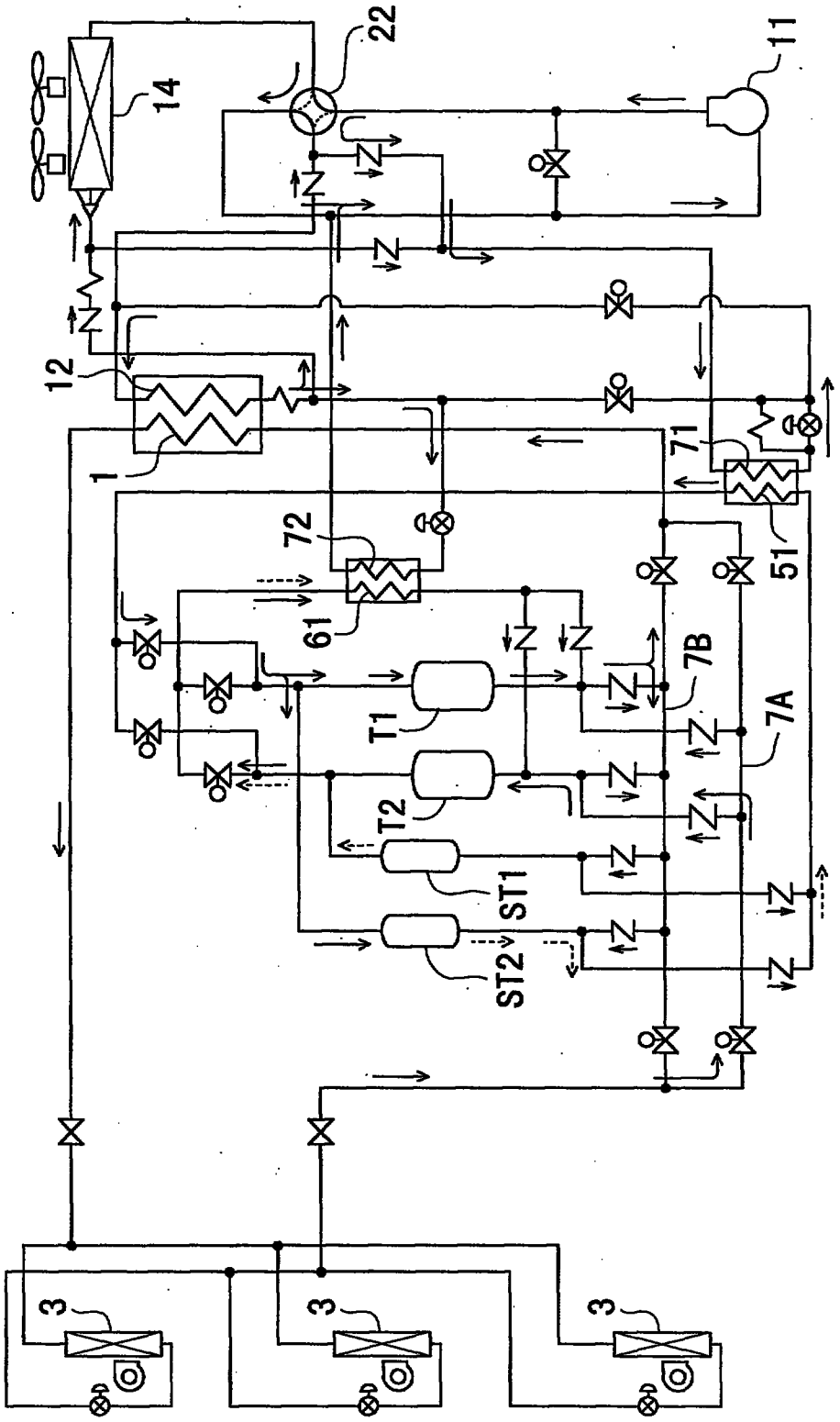


Fig. 36

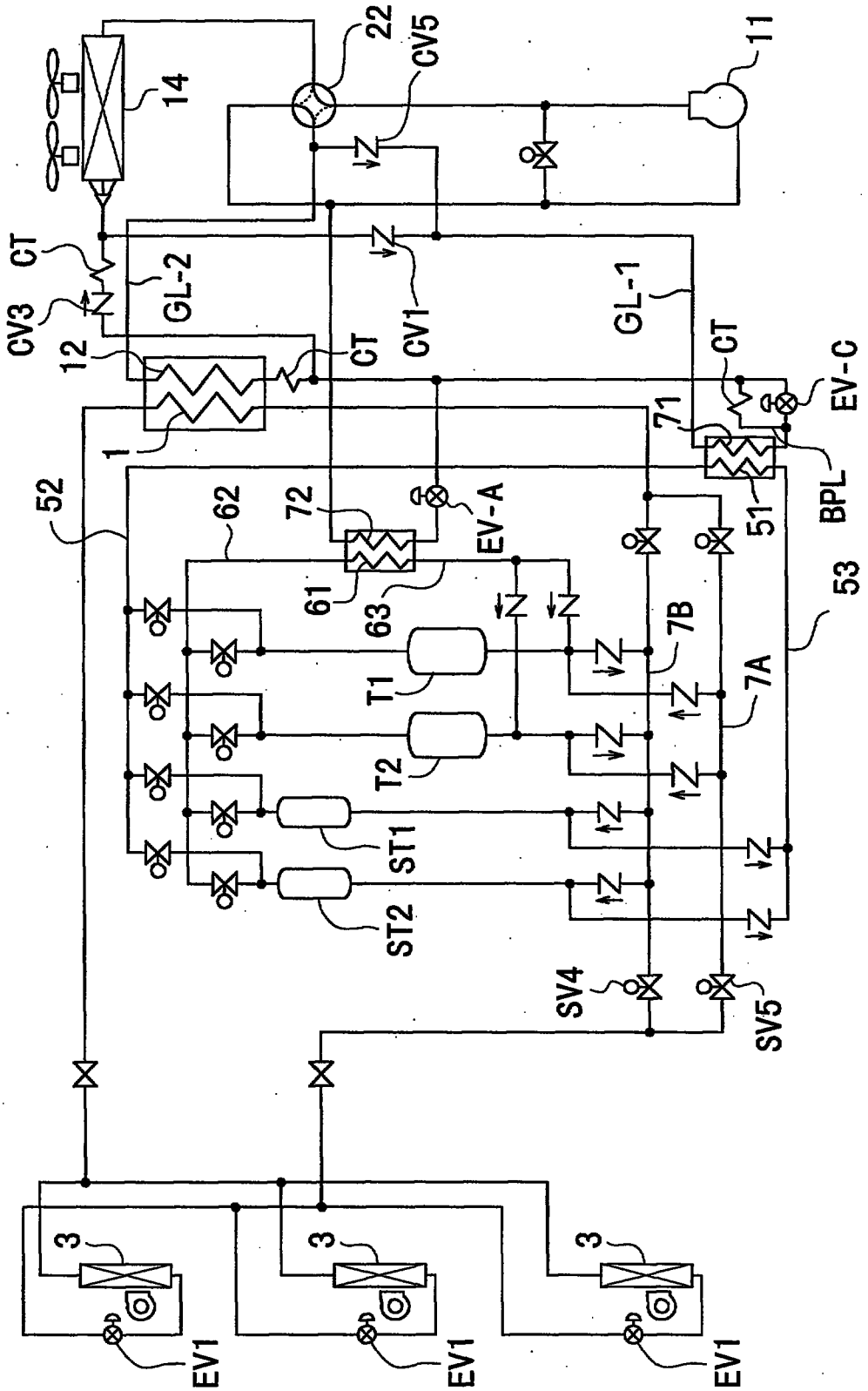

14
22
CV5
11
CT
CV3
GL-2
CV1
GL-1
12
CT
CT
EV-C
1
71
BPL
52
62
72
EV-A
51
61
63
53
7B
T1
7A
T2
ST1
ST2
SV4
SV5
3
3
3
EV1
EV1
EV1

Fig. 37

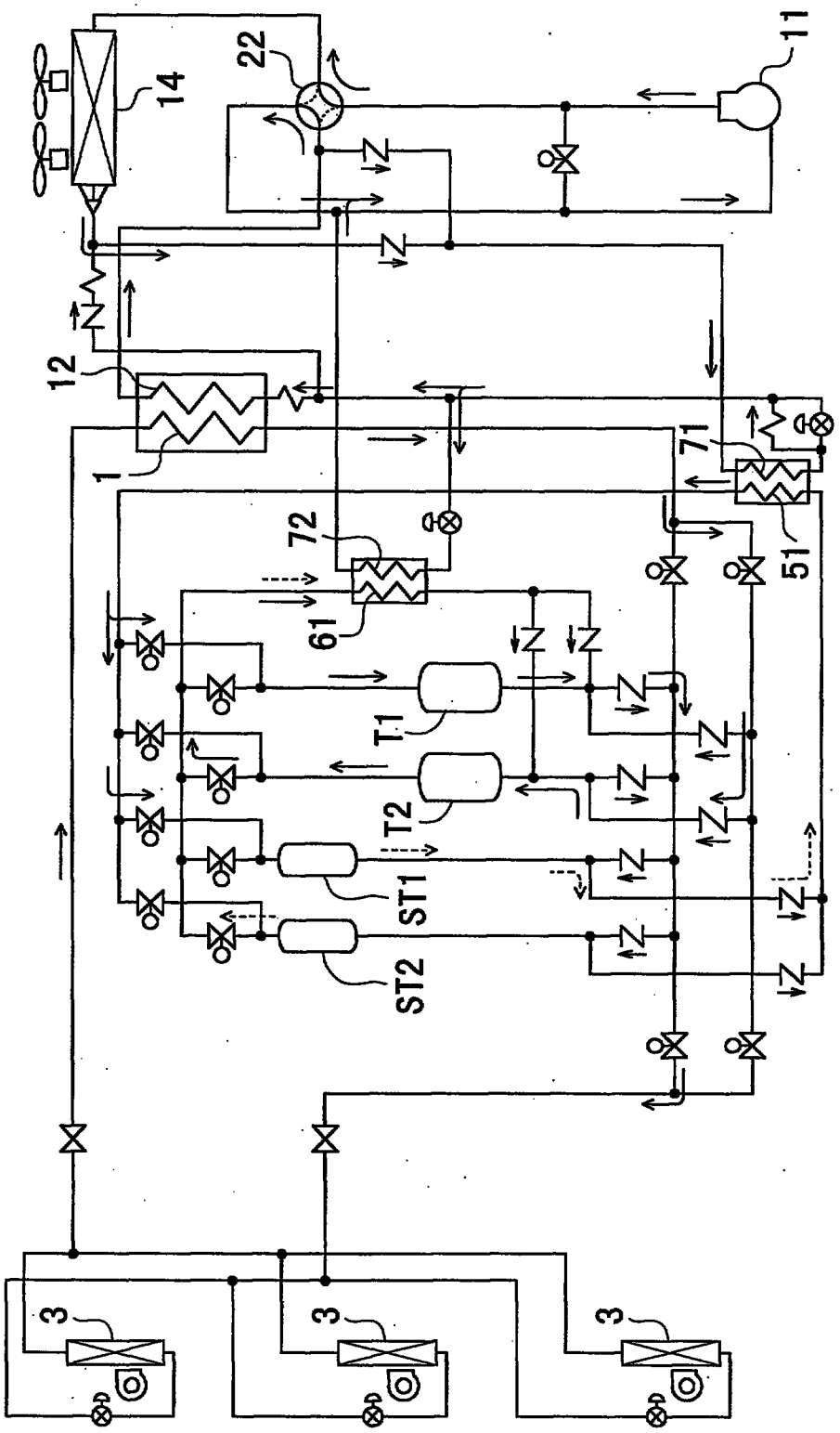


Fig. 38

