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Application Data Sheet 37 CFR 1.76	Attorney Docket Number	0021944-080US-1
	Application Number	
Title of Invention	COOLING SYSTEM FOR HIGH DENSITY HEAT LOAD	

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Applicant Information:

Applicant 1 Remove

Applicant Authority	◉ Inventor	○ Legal Representative under 35 U.S.C. 117	○ Party of Interest under 35 U.S.C. 118	
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Columbus	OH	US

Application Data Sheet 37 CFR 1.76	Attorney Docket Number	0021944-080US-1
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Title of Invention	COOLING SYSTEM FOR HIGH DENSITY HEAT LOAD	

Citizenship under 37 CFR 1.41(b) i	US				
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Application Data Sheet 37 CFR 1.76	Attorney Docket Number	0021944-080US-1
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Title of Invention	COOLING SYSTEM FOR HIGH DENSITY HEAT LOAD	

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All Inventors Must Be Listed - Additional Inventor Information blocks may be generated within this form by selecting the **Add** button.	Add		

Correspondence Information:

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☐ **An Address is being provided for the correspondence Information of this application.**			
Customer Number	26720		
Email Address	hoip@lockelord.com	Add Email	Remove Email

Application Information:

Title of the Invention	COOLING SYSTEM FOR HIGH DENSITY HEAT LOAD		
Attorney Docket Number	0021944-080US-1	**Small Entity Status Claimed**	☐
Application Type	Nonprovisional		
Subject Matter	Utility		
Suggested Class (if any)		**Sub Class (if any)**	
Suggested Technology Center (if any)			
Total Number of Drawing Sheets (if any)	3	**Suggested Figure for Publication (if any)**	2

Publication Information:

☐	Request Early Publication (Fee required at time of Request 37 CFR 1.219)
☐	**Request Not to Publish.** I hereby request that the attached application not be published under 35 U.S.C. 122(b) and certify that the invention disclosed in the attached application **has not and will not** be the subject of an application filed in another country, or under a multilateral international agreement, that requires publication at eighteen months after filing.

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Application Data Sheet 37 CFR 1.76	Attorney Docket Number	0021944-080US-1
	Application Number	
Title of Invention	COOLING SYSTEM FOR HIGH DENSITY HEAT LOAD	

Domestic Benefit/National Stage Information:

This section allows for the applicant to either claim benefit under 35 U.S.C. 119(e), 120, 121, or 365(c) or indicate National Stage entry from a PCT application. Providing this information in the application data sheet constitutes the specific reference required by 35 U.S.C. 119(e) or 120, and 37 CFR 1.78(a)(2) or CFR 1.78(a)(4), and need not otherwise be made part of the specification.

Prior Application Status: Patented — Remove

Application Number	Continuity Type	Prior Application Number	Filing Date (YYYY-MM-DD)	Patent Number	Issue Date (YYYY-MM-DD)
	Continuation of	10904889	2004-12-02	8261565	2012-09-11

Prior Application Status: Expired — Remove

Application Number	Continuity Type	Prior Application Number	Filing Date (YYYY-MM-DD)
10904889	non provisional of	60527527	2003-12-05

Additional Domestic Benefit/National Stage Data may be generated within this form by selecting the **Add** button. — Add

Foreign Priority Information:

This section allows for the applicant to claim benefit of foreign priority and to identify any prior foreign application for which priority is not claimed. Providing this information in the application data sheet constitutes the claim for priority as required by 35 U.S.C. 119(b) and 37 CFR 1.55(a).

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Application Number	Country i	Parent Filing Date (YYYY-MM-DD)	Priority Claimed
			○ Yes ⦿ No

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Assignee Information:

Providing this information in the application data sheet does not substitute for compliance with any requirement of part 3 of Title 37 of the CFR to have an assignment recorded in the Office.

Assignee 1 — Remove

If the Assignee is an Organization check here. ☒

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Country i	US	Postal Code	43083
Phone Number		Fax Number	
Email Address			

Additional Assignee Data may be generated within this form by selecting the **Add** button. — Add

Application Data Sheet 37 CFR 1.76	Attorney Docket Number	0021944-080US-1
	Application Number	

Title of Invention	COOLING SYSTEM FOR HIGH DENSITY HEAT LOAD

Signature:

A signature of the applicant or representative is required in accordance with 37 CFR 1.33 and 10.18. Please see 37 CFR 1.4(d) for the form of the signature.					
Signature	/D. Brit Nelson/			Date (YYYY-MM-DD)	2012-08-31
First Name	D. Brit	Last Name	NELSON	Registration Number	40370

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COOLING SYSTEM FOR HIGH DENSITY HEAT LOAD

CROSS-REFERENCE TO RELATED APPLICATION

This application is a continuation of US Application Serial No. 10/904,889, filed December 2, 2004, which claims the benefit of U.S. Provisional Patent Application Serial No. 60/527,527, filed December 5, 2003, both of which are incorporated by reference.

BACKGROUND

The present disclosure generally relates to cooling systems, and more particularly, to a cooling system for a high density heat load.

Electronic equipment in a critical space, such as a computer room or telecommunications room, requires precise, reliable control of room temperature, humidity, and airflow. Excessive heat or humidity can damage or impair the operation of computer systems and other components. For this reason, precision cooling systems are operated to provide cooling in these situations. However, problems may occur when cooling such high density heat loads using a direct expansion (DX) cooling system. Existing DX systems for high-density loads monitor air temperatures and other variables to control the cooling capacity of the system in response to load changes. Thus, existing DX systems require rather sophisticated controls, temperature sensors, and other control components. Therefore, a need exists for a cooling system that is responsive to varying density heat loads and that requires less control of valves and other system components. Moreover, conventional computer room air conditioning systems require excessive floor space for managing high-density heat loads. The present disclosure is directed to overcoming, or at least reducing the effects of, one or more of the problems set forth above.

SUMMARY

A cooling system is disclosed for transferring heat from a heat load to an environment. The cooling system has a working fluid, which is a volatile working fluid in exemplary embodiments. The cooling system includes first and second cooling cycles that are thermally connected to one another. The first cooling cycle includes a pump, a first heat exchanger, and a second heat exchanger.

The first heat exchanger is in fluid communication with the pump through piping and is in thermal communication with the heat load, which may be a computer room, electronics enclosure, or other space. The first heat exchanger can be an air-to-fluid heat exchanger, for example. In addition, a flow regulator can be positioned between the pump and the first heat exchanger.

The second heat exchanger includes first and second fluid paths in thermal communication with one another. The second heat exchanger can be a fluid-to-fluid heat exchanger, for example. The first fluid path for the working fluid of the cooling system connects the first heat exchanger to the pump. The second fluid path forms part of the second cooling cycle.

In one embodiment of the disclosed cooling system, the second cooling cycle includes a chilled water system in thermal communication with the environment. In another embodiment of the disclosed cooling system, the second cooling cycle includes a refrigeration system in thermal communication with the environment. The refrigeration system can include a compressor, a condenser, and an expansion device. The compressor is in fluid communication with one end of the second fluid path of the second heat exchanger. The condenser, which can be an air-to-fluid heat exchanger, is in fluid communication with the environment. The condenser has an inlet connected to the compressor and has an outlet connected to another end of the second fluid path through the second heat exchanger. The expansion device is positioned between the outlet of the condenser and the other end of the second fluid path.

The foregoing summary is not intended to summarize each potential embodiment or every aspect of the present disclosure.

BRIEF DESCRIPTION OF THE DRAWINGS

The foregoing summary, a preferred embodiment, and other aspects of the subject matter of the present disclosure will be best understood with reference to the following detailed description of specific embodiments when read in conjunction with the accompanying drawings, in which:

Figure 1 schematically illustrates one embodiment of a cooling system according to certain teachings of the present disclosure.

Figure 2 schematically illustrates another embodiment of a cooling system according to certain teachings of the present disclosure.

Figure 3 illustrates a cycle diagram of the disclosed cooling system.

Figure 4 illustrates a cycle diagram of a typical vapor compression refrigeration system.

While the disclosed cooling system is susceptible to various modifications and alternative forms, specific embodiments thereof have been shown by way of example in the drawings and are described herein in detail. The figures and written description are not intended to limit the scope of the inventive concepts in any manner. Rather, the figures and written description are provided to illustrate the inventive concepts to a person of ordinary skill in the art by reference to particular embodiments.

DETAILED DESCRIPTION

Referring to Figures 1 and 2, the disclosed cooling system 10 includes a first cooling cycle 12 in thermal communication with a second cycle 14. The disclosed cooling system 10 also includes a control system 100. Both the first and second cycles 12 and 14 include independent working fluids. The working fluid in the first cycle is any volatile fluid suitable for use as a conventional refrigerant, including but not limited to chlorofluorocarbons (CFCs), hydrofluorocarbons (HFCs), or hydrochloro-fluorocarbons (HCFCs). Use of a volatile working fluid eliminates the need for using water located above sensitive equipment, as is sometimes done in conventional systems for cooling computer room. The first cycle 12 includes a pump 20, one or more first heat exchangers (evaporators) 30, a second heat exchanger 40, and piping to interconnect the various components of the first cycle 12. The first cycle 12 is not a vapor compression refrigeration system. Instead, the first cycle 12 uses the pump 20 instead of a compressor to circulate a volatile working fluid for removing heat from a heat load. The pump 20 is preferably capable of pumping the volatile working fluid throughout the first cooling cycle 12 and is preferably controlled by the control system 100.

The first heat exchanger 30 is an air-to-fluid heat exchanger that removes heat from the heat load (not shown) to the first working fluid as the first working fluid passes through the first fluid path in first heat exchanger 30. For example, the air-to-fluid heat exchanger 30 can include a plurality of tubes for the working fluid arranged to allow warm air to pass therebetween. It will be appreciated that a number of air-to-fluid heat exchangers known in the art can be used with the disclosed cooling system 10. A flow regulator 32 can be connected between the piping 22 and the inlet of the evaporator 30 to regulate the flow of working fluid into the evaporator 30. The flow regulator 32 can be a solenoid valve or other type of device for regulating flow in the cooling system 10. The flow regulator 32 preferably maintains a constant output flow independent of the inlet pressure over the operating pressure range of the system. In the embodiment of Figures 1 and 2, the first cycle 12 includes a plurality of evaporators 30 and flow regulators 32 connected to the piping 22. However, the disclosed system can have one or more than one evaporator 30 and flow regulators 32 connected to the piping 22.

The second heat exchanger 40 is a fluid-to-fluid heat exchanger that transfers the heat from the first working fluid to the second cycle 14. It will be appreciated that a number of fluid-to-fluid heat exchangers known in the art can be used with the disclosed cooling system 10. For example, the fluid-to-fluid heat exchanger 40 can include a plurality of tubes for one fluid positioned in a chamber or shell containing the second fluid. A coaxial ("tube-in-tube") exchanger would also be suitable. In certain embodiments, it is preferred to use a plate heat exchanger. The first cycle 12 can also include a receiver 50 connected to the outlet piping 46 of the second heat exchanger 40 by a bypass line 52. The receiver 50 may store and accumulate the working fluid in the first cycle 12 to allow for changes in the temperature and heat load.

In one embodiment, the air-to-fluid heat exchanger 30 can be used to cool a room holding computer equipment. For example, a fan 34 can draw air from the room (heat load) through the heat exchanger 30 where the first working fluid absorbs heat from the air. In another embodiment, the air-to-fluid heat exchanger 30 can be used to directly remove heat from electronic equipment (heat load) that generates the heat by mounting the heat exchanger 30 on or close to the equipment. For example, electronic equipment is typically contained in an enclosure (not shown). The heat exchanger 30 can mount to the enclosure, and fans 34 can draw air from the enclosure through the heat exchanger 30. Alternatively, the first exchanger 30 may be in direct thermal contact with the heat source (e.g. a cold plate). It will be appreciated by those skilled in the art that the heat transfer rates, sizes, and other design variables of the components of the disclosed cooling system 10 depend on the size of the disclosed cooling system 10, the magnitude of the heat load to be managed, and on other details of the particular implementation.

In the embodiment of the disclosed cooling system 10 depicted in Figure 1, the second cycle 14 includes a chilled water cycle 60 connected to the fluid-to-fluid heat exchanger 40 of the first cycle 12. In particular, the second heat exchanger 40 has first and second portions or fluid paths 42 and 44 in thermal communication with one another. The first path 42 for the volatile working fluid is connected between the first heat exchanger 30 and the pump. The second fluid path 44 is connected to the chilled water cycle 60. The chilled water cycle 60 may be similar to those known in the art. The chilled water system 60 includes a second working fluid that absorbs heat from the first working fluid passing through the fluid-to-fluid heat exchanger 40. The second working fluid is then chilled by techniques known in the art for a conventional chilled water cycle. In general, the second working fluid can be either volatile or non-volatile. For example, in the embodiment of Figure 1, the second working fluid can be water, glycol or mixtures thereof. Therefore, the embodiment of the first cycle 12 in Figure 1 can be constructed as an independent unit that houses the pump 20, air-to-fluid heat exchanger 30, and fluid-to-fluid heat exchanger 40 and can be connected to an existing chilled water service that is available in the building housing the equipment to be cooled, for example.

In the embodiment of the disclosed cooling system 10 in Figure 2, the first cycle 12 is substantially the same as that described above. However, the second cycle 14 includes a vapor compression refrigeration system 70 connected to the second portion or flow path 44 of heat exchanger 40 of the first cycle 12. Instead of using chilled water to remove the heat from the first cycle 12 as in the embodiment of Figure 1, the refrigeration system 70 in Figure 2 is directly connected to or is the "other half" of the fluid-to-fluid heat exchanger 40. The vapor compression refrigeration system 70 can be substantially similar to those known in the art. An exemplary vapor compression refrigeration system 70 includes a compressor 74, a condenser 76, and an expansion device 78. Piping 72 connects these components to one another and to the second flow path 44 of the heat exchanger 40.

The vapor compression refrigeration system 70 removes heat from the first working fluid passing through the second heat exchanger 40 by absorbing heat from the exchanger 40 with a second working fluid and expelling that heat to the environment (not shown). The second working fluid can be either volatile or non-volatile. For example, in the embodiment of Figure 2, the second working fluid can be any conventional chemical refrigerant, including but not limited to chlorofluorocarbons (CFCs), hydrofluorocarbons (HFCs), or hydrochloro-fluorocarbons (HCFCs). The expansion device 78 can be a valve, orifice or other apparatus known to those skilled in the art to produce a pressure drop in the working fluid passing therethrough. The compressor 74 can be any type of compressor known in the art to be suitable for refrigerant service such as reciprocating compressors, scroll compressors, or the like. In the embodiment depicted in Figure 2, the cooling system 10 is self-contained. For example, the vapor compression refrigeration system 70 can be part of a single unit that also houses pump 20 and fluid-to-fluid heat exchanger 30.

During operation of the disclosed system, pump 20 moves the working fluid via piping 22 to the air-to-fluid heat exchanger 30. Pumping increases the pressure of the working fluid, while its enthalpy remains substantially the same. (See leg 80 of the cycle diagram in Figure 3). The pumped working fluid can then enter the air-to-fluid heat exchanger or evaporator 30 of the first cycle 12. A fan 34 can draw air from the heat load through the heat exchanger 30. As the warm air from the heat load (not shown) enters the air-to-fluid heat exchanger 30, the volatile working fluid absorbs the heat. As the fluid warms through the heat exchanger, some of the volatile working fluid will evaporate. (See leg 82 of the cycle diagram in Figure 3). In a fully loaded system 10, the fluid leaving the first heat exchanger 30 will be a saturated vapor. The vapor flows from the heat exchanger 30 through the piping 36 to the fluid-to-fluid heat exchanger 40. In the piping or return line 36, the working fluid is in the vapor state, and the pressure of the fluid drops while its enthalpy remains substantially constant. (See leg 84 of the cycle diagram in Figure 3). At the fluid-to-fluid heat exchanger 40, the vapor in the first fluid path 42 is condensed by transferring heat to the second, colder fluid of the second cycle 12 in the second fluid path 44. (See leg 86 of the cycle diagram in Figure 3). The condensed working fluid leaves the heat exchanger 40 via piping 44 and enters the pump 20, where the first cycle 12 can be repeated.

The second cooling cycle 14 operates in conjunction with first cycle 12 to remove heat from the first cycle 12 by absorbing the heat from the first working fluid into the second working fluid and rejecting the heat to the environment (not shown). As noted above, the second cycle 14 can include a chilled water system 60 as shown in Figure 1 or a vapor compression refrigeration system 70 as shown in Figure 2. During operation of chilled water system 60 in Figure 1, for example, a second working fluid can flow through the second fluid path 44 of heat exchanger 40 and can be cooled in a water tower (not shown). During operation of refrigeration system 70 in Figure 2, for example, the second working fluid passes through the second portion 44 of fluid-to-fluid heat exchanger 40 and absorbs heat from the volatile fluid in the first cycle 12. The working fluid evaporates in the process. (See leg 92 of the typical vapor-compression refrigeration cycle depicted in Figure 4). The vapor travels to the compressor 74 where the working fluid is compressed. (See leg 90 of the refrigeration cycle in Figure 4). The compressor 74 can be a reciprocating, scroll or other type of compressor known in the art. After compression, the working fluid travels through a discharge line to the condenser 76, where heat from the working fluid is dissipated to an external heat sink, *e.g.,* the outdoor environment. (See leg 96 of the refrigeration cycle in Figure 4). Upon leaving condenser 76, refrigerant flows through a liquid line to expansion device 75. As the refrigerant passes through the expansion device 75, the second working fluid experiences a pressure drop. (See leg 94 of the refrigeration cycle in Figure 4.) Upon leaving expansion device 75, the working fluid flows through the second fluid path of fluid-to-fluid heat exchanger 40, which acts as an evaporator for the refrigeration cycle 70.

Conventional cooling systems for computer rooms or the like take up valuable floor space. The present cooling system 10, however, can cool high-density heat loads without consuming valuable floor space. Furthermore, in comparison to conventional types of cooling solutions for high-density loads, such as computing rooms, cooling system 10 conserves energy, because pumping a volatile fluid requires less energy than pumping a non-volatile fluid such as water. In addition, pumping the volatile fluid reduces the size of the pump that is required as well as the overall size and cost of the piping that interconnects the system components.

The disclosed system 10 advantageously uses the phase change of a volatile fluid to increase the cooling capacity per square foot of a space or room. In addition, the disclosed system 10 also eliminates the need for water in cooling equipment mounted above computing equipment, which presents certain risks of damage to the computing equipment in the event of a leak. Moreover, since the system is designed to remove sensible heat only, the need for condensate removal is eliminated. As is known in the art, cooling air to a low temperature increases the relative humidity, meaning condensation is likely to occur. If the evaporator is mounted on an electronics enclosure, for example, condensation may occur within the enclosure, which poses significant risk to the electronic equipment. In the present system, the temperature in the environment surrounding the equipment is maintained above the dew point to ensure that condensation does not occur. Because the disclosed cooling system does not perform latent cooling, all of the cooling capacity of the system will be used to cool the computing equipment.

The disclosed cooling system 10 can handle varying heat loads without the complex control required on conventional direct expansion systems. The system is self-regulating in that the pump 20 provides a constant flow of volatile fluid to the system. The flow regulators 32 operate so as to limit the maximum flow to each heat exchanger 30. This action balances the flow to each heat exchanger 30 so that each one gets approximately the same fluid flow. If a heat exchanger is under "high" load, then the volatile fluid will tend to flash off at a higher rate than one under a lower load. Without the flow regulator 32, more of the flow would tend to go to the "lower" load heat exchanger because it is the colder spot and lower fluid pressure drop. This action would tend to "starve" the heat exchanger under high load and it would not cool the load properly.

The key system control parameter that is used to maintain all sensible cooling is the dewpoint in the space to be controlled. The disclosed cooling system 10 controls the either the chilled water or the vapor compression system so that the fluid going to the above mentioned heat exchangers 30 is always above the dewpoint in the space to be controlled. Staying above the dewpoint insures that no latent cooling can occur.

The foregoing description of preferred and other embodiments is not intended to limit or restrict the scope or applicability of the inventive concepts conceived by the Applicants. In exchange for disclosing the inventive concepts contained herein, the Applicants desire all patent rights afforded by the appended claims. Therefore, it is intended that the appended claims include all modifications and alterations to the full extent that they come within the scope of the following claims or the equivalents thereof.

WHAT IS CLAIMED IS:

1. A cooling system for transferring heat from a heat load, the cooling system comprising:

a two-phase working fluid;

a pump configured to increase the pressure of the working fluid without substantially increasing the enthalpy of the working fluid;

an air-to-fluid heat exchanger in fluid communication with the pump and in thermal communication with the heat load;

a fluid-to-fluid heat exchanger having a first fluid path in fluid communication with the air-to-fluid heat exchanger and the pump, and a second fluid path, the first and second fluid paths being in thermal communication with one another;

a second heat transfer system in fluid communication with the second fluid path and comprising:

a second portion of the fluid-to-fluid heat exchanger;

a working fluid; and

a compressor;

wherein air passing through the air-to-fluid heat exchanger causes at least a portion of the two-phase working fluid to change phase from a liquid to a gas within the air-to-fluid heat exchanger; and

a controller operatively coupled to at least the second fluid path and configured to prevent condensation on the air-to-fluid heat exchanger by controlling the amount of heat transferred to the second fluid path so that a temperature of the two-phase working fluid within the air-to-fluid heat exchanger is above a dew point temperature of the air passing through the air-to-fluid heat exchanger.

2. The cooling system of claim 1, further comprising a flow regulator positioned between the pump and the air-to-fluid heat exchanger.

3. The cooling system of claim 1, further comprising a working fluid receiver in fluid communication between the fluid-to-fluid heat exchanger and the pump.

4. The cooling system of claim 1, further comprising a working fluid flow regulating valve in fluid communication between the pump and the air-to-fluid heat exchanger.

5. The cooling system of claim 1, further comprising a working fluid flow regulating valve in fluid communication between the pump and the air-to-fluid heat exchanger and a working fluid receiver in fluid communication between the fluid-to-fluid heat exchanger and the pump.

6. A cooling system for transferring heat from a heat load to an environment, the cooling system comprising:

a first cooling cycle containing a two-phase working fluid; and

a second cooling cycle thermally connected to the first cooling cycle;

wherein the first cooling cycle comprises:

a pump configured to increase the pressure of the working fluid without substantially increasing the enthalpy of the working fluid;

an air-to-fluid heat exchanger in fluid communication with the pump and in thermal communication with the heat load;

a second heat exchanger having a first fluid path for the working fluid in fluid communication with the air-to-fluid heat exchanger and the pump, and a second fluid path comprising a portion of the second cooling cycle;

wherein the first and second fluid paths are in thermal communication with one another;

wherein the heat load causes at least a portion of the two-phase working fluid to change phase from a liquid to a gas within the air-to-fluid heat exchanger; and

wherein the second cooling cycle comprises a vapor compression refrigeration system in thermal communication with the environment and wherein the second cooling cycle is controlled to maintain a temperature of the two-phase working fluid entering the air-to-fluid heat exchanger above a dew point of air flowing through the air-to-fluid heat exchanger.

7. The cooling system of claim 6, further comprising a working fluid receiver in the first cooling cycle between the second heat exchanger and the pump.

8. The cooling system of claim 6, further comprising a working fluid flow regulating valve in fluid communication between the pump and the air-to-fluid heat exchanger.

9. The cooling system of claim 6, further comprising a working fluid flow regulating valve in fluid communication between the pump and the air-to-fluid heat exchanger and a working fluid receiver in fluid communication between the second heat exchanger and the pump.

10. A cooling system for transferring heat from a heat load to an environment, the cooling system comprising:

a working fluid pump configured to increase the pressure of a two-phase working fluid without substantially increasing the enthalpy of the working fluid;

an air-to-fluid heat exchanger connected to the pump and having a fluid path in thermal communication with the heat load;

a second heat exchanger having first and second fluid paths in thermal communication with one another, wherein the first fluid path provides fluid communication from the air-to-fluid heat exchanger to the pump, and wherein the second fluid path is adapted to thermally connect the air-to-fluid heat exchanger in the first fluid path to a vapor compression refrigeration system that is in thermal communication with the environment;

wherein air passing through the air-to-fluid heat exchanger transfers heat from the heat load and causes at least a portion of the working fluid to change phase from a liquid to a gas; and

a controller operatively coupled to the vapor compression refrigeration system and configured to maintain a temperature of the working fluid between the second heat exchanger and the air-to-fluid heat exchanger above a dew point temperature of the air passing through the air-to-fluid heat exchanger so that the cooling system removes only sensible heat from the air and thereby prevents condensation on the air-to-fluid heat exchanger.

11. The cooling system of claim 10, further comprising a working fluid receiver in fluid communication between the second heat exchanger and the pump.

12. The cooling system of claim 10, further comprising a working fluid flow regulating valve in fluid communication between the pump and the air-to-fluid heat exchanger.

13. The cooling system of claim 10, further comprising a working fluid flow regulating valve in fluid communication between the pump and the air-to-fluid heat exchanger and a working fluid receiver in fluid communication between the second heat exchanger and the pump.

14 A heat transfer system, comprising:

a first heat transfer subsystem adapted to circulate there through a first working fluid, wherein the first working fluid is selected from the group consisting of: chlorofluorocarbons, hydrofluorocarbons and hydrochlorofluorocarbons, comprising:

at least one air-to-fluid heat exchanger in thermal communication with a heat load;

a pump configured to increase the pressure of the first working fluid without substantially increasing the enthalpy of the first working fluid; and

at least a portion of a second heat exchanger;

a second heat transfer subsystem comprising:

at least a second portion of the second heat exchanger;

the second heat transfer subsystem adapted to circulate a second working fluid there through; and

wherein air passing through the air-to-fluid heat exchanger causes at least a portion of the first working fluid to undergo a phase change from a liquid to a gas in the first heat transfer subsystem; and

a system controller operatively coupled to the second subsystem and configured to prevent condensation on the air-to-fluid heat exchanger by maintaining the first working fluid leaving the second heat exchanger above a dew point temperature of the air passing through the air-to-fluid heat exchanger.

15. The system of claim 14, wherein the heat load is a room.

16. The system of claim 14, wherein the heat load is an electronics cabinet.

17. The system of claim 14 further comprising a flow regulator associated with at least one air-to-fluid heat exchanger and which is adapted to control an amount of first working fluid flowing through the associated air-to-fluid heat exchanger.

18. The system of claim 17, wherein the flow regulator is adapted to control the amount of first working fluid flowing through the air-to-fluid heat exchanger independently of fluid pressure.

19. The system of claim 17, wherein the flow regulator is adapted to maintain a substantially constant flow of first working fluid through the air-to-fluid heat exchanger.

20. The system of claim 14, further comprising a receiver in fluid communication with the first heat transfer subsystem for accumulating a portion of the first working fluid.

21. The system of claim 20, wherein the receiver is adapted to accumulate a portion of the first working fluid based upon temperature and/or heat load.

22. The system of claim 14, further comprising a flow regulator associated with a plurality of air-to-fluid heat exchangers and which are adapted to limit an amount of first working fluid flowing through each of the associated air-to-fluid heat exchangers.

23. The system of claim 14, wherein the second heat exchanger is selected from the group consisting of: a tube-in-tube heat exchanger, a shell and tube heat exchanger and a plate and frame heat exchanger.

24. The cooling system of claim 14, further comprising a working fluid receiver in the first heat transfer subsystem between the second heat exchanger and the pump.

25. The cooling system of claim 14, further comprising a working fluid flow regulating valve in fluid communication between the pump and the air-to-fluid heat exchanger and a working fluid receiver in fluid communication between the second heat exchanger and the pump.

26. A cooling system for removing heat from a high density heat load, comprising:

a first heat transfer system comprising:

a two-phase working fluid;

a plurality of air-to-fluid heat exchangers configured to transfer heat from the load to the working fluid so that at least a portion of the working fluid changes phase from a liquid to a gas within at least one of the air-to-fluid heat exchangers;

a working fluid flow regulator associated with at least one of the plurality of air-to-fluid heat exchangers and configured to limit the maximum working fluid flow to each associated air-to-fluid heat exchanger;

a working fluid receiver configured to hold working fluid based on working fluid temperature or cooling system load;

a pump configured to increase the pressure of the working fluid without substantially increasing the enthalpy of the working fluid; and

a first portion of a fluid-to-fluid heat exchanger;

wherein all of the first heat transfer system is arranged in fluid communication;

a second heat transfer system comprising:

a second portion of the fluid-to-fluid heat exchanger;

a working fluid;

a compressor; and

wherein all of the second heat transfer system is arranged in fluid communication;

wherein the first heat transfer system is thermally coupled to the second heat transfer system by the fluid-to-fluid heat exchanger; and

a cooling system controller monitoring the dew point temperature of air flowing through the air-to-fluid heat exchanger, and operatively connected to the second heat transfer system to prevent condensation on the air-to-fluid heat exchangers by maintaining the first heat transfer system working fluid entering the air-to-fluid heat exchangers at a temperature above the dew point temperature of the air flow.

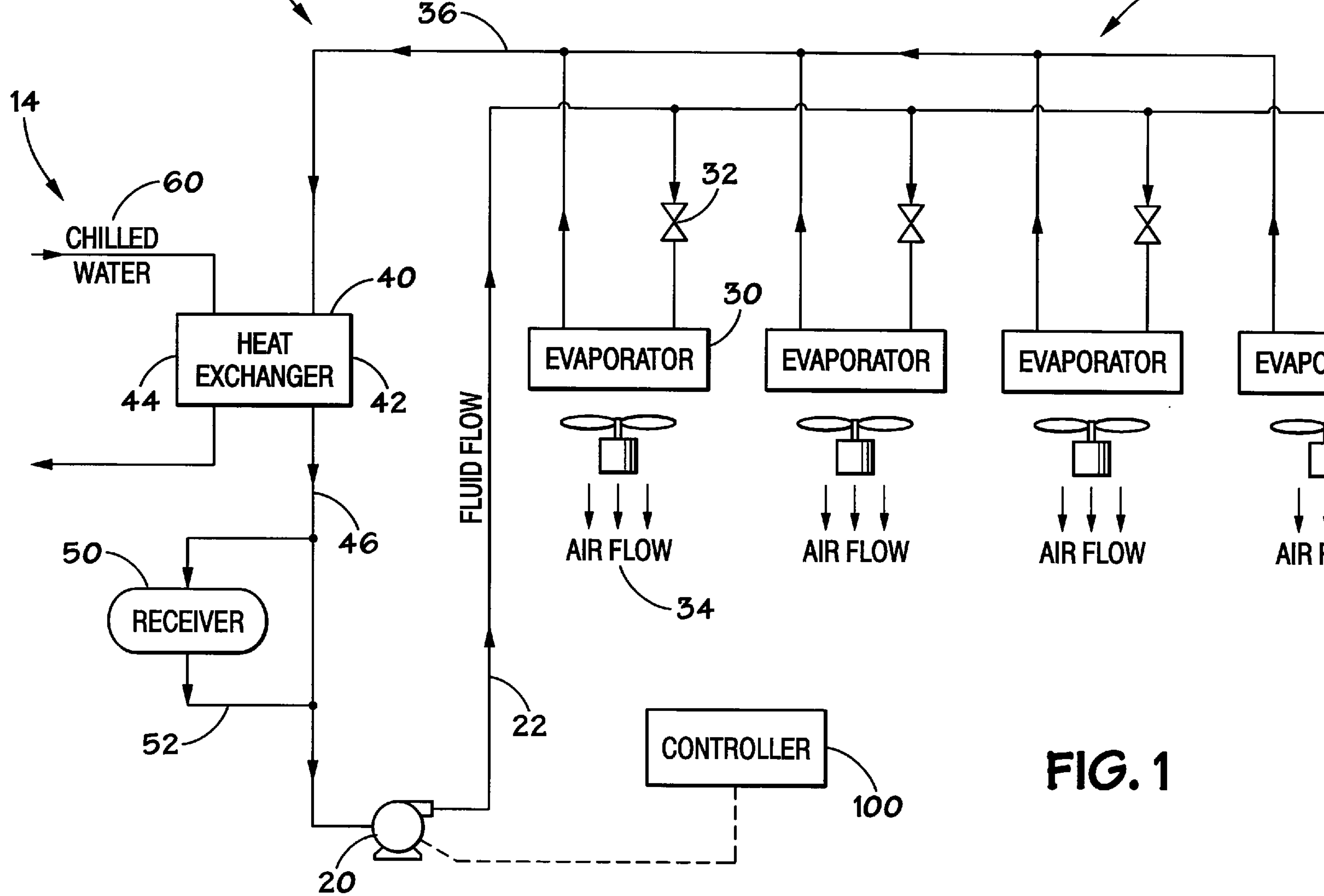
27. The system of claim 26, wherein the air-to-fluid heat exchangers are located within an enclosure and the high density heat load is created by electronics within the enclosure, the enclosure having a forced air flow path across the electronics and through the air-to-fluid heat exchangers.

28. The system of claim 27, wherein first heat transfer system is configured so that the working fluid is cooled in the fluid-to-fluid heat exchanger and then flows to the receiver and pump, and then flows through the regulating valves and then into each air-to-fluid heat exchanger where at least a portion of the working fluid boils in each air-to-fluid heat exchanger, and then the heated working fluid returns to the fluid-to-fluid heat exchanger where it is once again cooled.

29. The system of claim 26, further including a working fluid flow regulator associated with each air-to-fluid heat exchanger and each flow regulator configured to limit the maximum working fluid flow to each air-to-fluid heat exchanger.

ABSTRACT

A cooling system for transferring heat from a heat load to an environment has a volatile working fluid. The cooling system includes first and second cooling cycles that are thermally connected to the first cooling cycle. The first cooling cycle is not a vapor compression cycle and includes a pump, an air-to-fluid heat exchanger, and a fluid-to-fluid heat exchanger. The second cooling cycle can include a chilled water system for transferring heat from the fluid-to-fluid heat exchanger to the environment. Alternatively, the second cooling cycle can include a vapor compression system for transferring heat from the fluid-to-fluid heat exchanger to the environment.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FLUID FLOW
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EVAPORATOR
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CONTROLLER
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FIG. 1

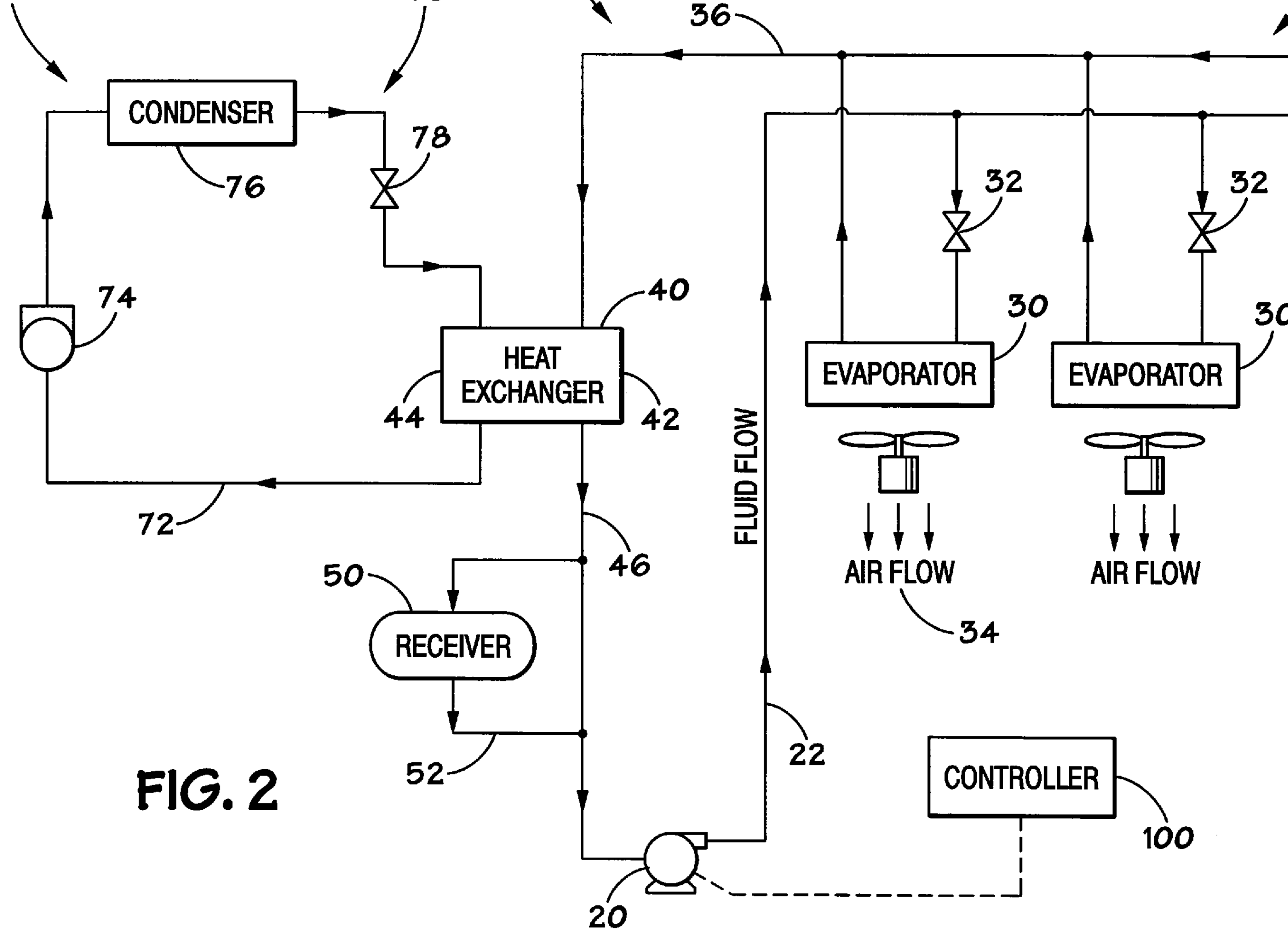
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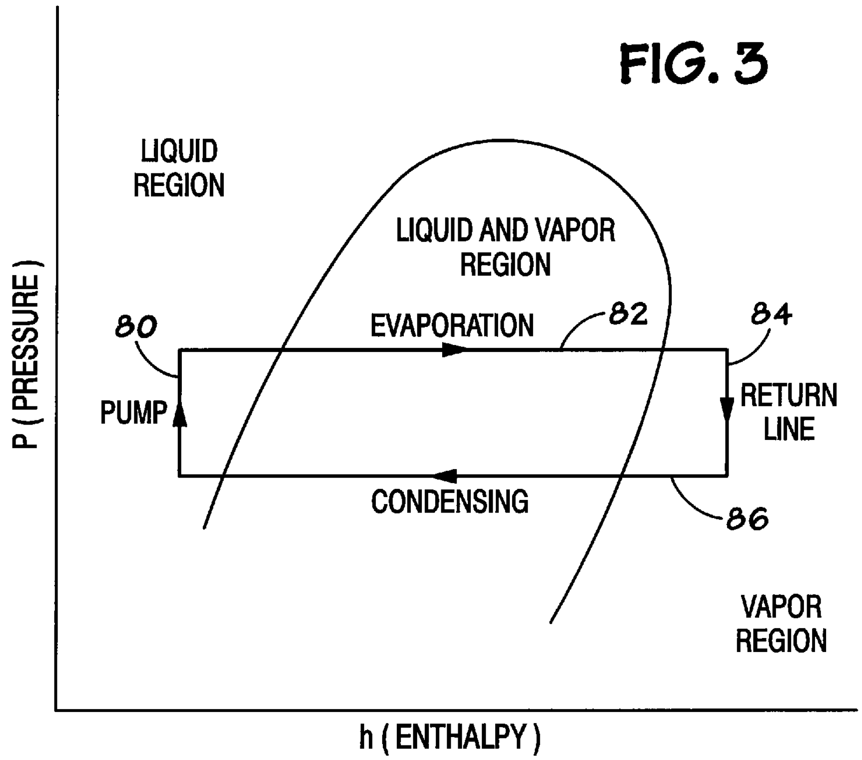
FIG. 2

FIG. 3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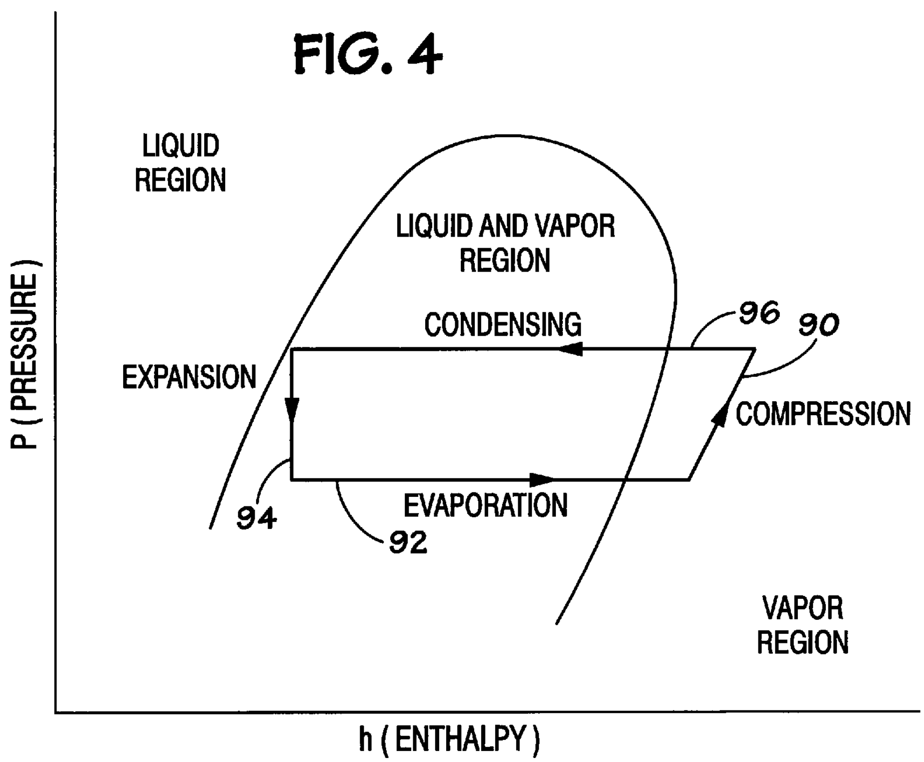
h (ENTHALPY)

FIG. 4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P (PRESSURE)
h (ENTHALPY)

PATENT
021972.080US

DECLARATION (37 CFR 1.63) FOR UTILITY OR DESIGN APPLICATION USING ANAPPLICATION DATA SHEET (37 CFR 1.76)

Title of Application: COOLING SYSTEM FOR HIGH DENSITY HEAT LOAD

This declaration is directed to

- ☒ The attached application.
- ☐ International Application Serial No. , filed on .

As a below named inventor, I hereby declare that:

I believe that I am the original and first inventor (if only one name is listed below) or the below named inventors are the original, first and joint inventors (if plural names are listed below) of the subject matter which is claimed and for which a patent is sought;

I have reviewed and understand the contents of the above-identified application, including the claims, as amended by any amendment specifically referred to above;

I acknowledge the duty to disclose to the United States Patent and Trademark Office all information known to me to be material to patentability as defined in 37 CFR 1.56, including for continuation-in-part applications, material information which became available between the filing date of the prior application and the national or PCT International filing date of the continuation-in-part application.

All statements made herein of my own knowledge are true, all statements made herein on information and belief are believed to be true, and further that these statements were made with the knowledge that willful false statements and the like are punishable by fine or imprisonment, or both, under 18 U.S.C. 1001, and may jeopardize the validity of the application or any patent issuing thereon.

Inventor's Full Name: Steven A. Borror	Citizen of: US
Signature: SA Borror	

Inventor's Full Name: Frank E. DiPaolo	Citizen of: US
Signature: Frank E. DiPaolo	

Inventor's Full Name: Thomas E. Harvey	Citizen of: US
Signature: Thomas E. Harvey	

Inventor's Full Name: Steven M. Madara	Citizen of: US
Signature: Steven M. Madara	

Inventor's Full Name: Reasey J. Mam Signature: ____________	Citizen of: US

Inventor's Full Name: Stephen C. Sillato Signature: ____________	Citizen of: US

Electronic Patent Application Fee Transmittal

Application Number:	
Filing Date:	
Title of Invention:	COOLING SYSTEM FOR HIGH DENSITY HEAT LOAD
First Named Inventor/Applicant Name:	Steven A. BORROR
Filer:	David B. Nelson/Neva Dare
Attorney Docket Number:	0021944-080US-1

Filed as Large Entity

Utility under 35 USC 111(a) Filing Fees

Description	Fee Code	Quantity	Amount	Sub-Total in USD($)
Basic Filing:				
Utility application filing	1011	1	380	380
Utility Search Fee	1111	1	620	620
Utility Examination Fee	1311	1	250	250
Pages:				
Claims:				
Claims in excess of 20	1202	9	60	540
Independent claims in excess of 3	1201	2	250	500
Miscellaneous-Filing:				

Description	Fee Code	Quantity	Amount	Sub-Total in USD($)
Petition:				
Patent-Appeals-and-Interference:				
Post-Allowance-and-Post-Issuance:				
Extension-of-Time:				
Miscellaneous:				
	Total in USD ($)			**2290**

Electronic Acknowledgement Receipt

EFS ID:	13641597
Application Number:	13601481
International Application Number:	
Confirmation Number:	9349
Title of Invention:	COOLING SYSTEM FOR HIGH DENSITY HEAT LOAD
First Named Inventor/Applicant Name:	Steven A. BORROR
Customer Number:	26720
Filer:	David B. Nelson/Neva Dare
Filer Authorized By:	David B. Nelson
Attorney Docket Number:	0021944-080US-1
Receipt Date:	31-AUG-2012
Filing Date:	
Time Stamp:	16:56:33
Application Type:	Utility under 35 USC 111(a)

Payment information:

Submitted with Payment	yes
Payment Type	Deposit Account
Payment was successfully received in RAM	$2290
RAM confirmation Number	5179
Deposit Account	121322
Authorized User	

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File Listing:

Document Number	Document Description	File Name	File Size(Bytes)/ Message Digest	Multi Part /.zip	Pages (if appl.)
1	Application Data Sheet	0021944-080US-1_ADS.pdf	1421703 48ad2926b944e92506a58e5e52f8bf70205bd078	no	6
Warnings:					
Information:					
2		080US-1_PA.pdf	199561 d313622296c0f0d40bd82e8eff2af85cef37e3db	yes	17

Multipart Description/PDF files in .zip description

Document Description	Start	End
Specification	1	9
Claims	10	16
Abstract	17	17

Warnings:					
Information:					
3	Drawings-only black and white line drawings	080US-1_Formal_Dwgs.pdf	138139 084d7b90f04c7e83cb215d77bd6456d0b594826a	no	3
Warnings:					
Information:					
4	Oath or Declaration filed	0021944-080US-1_Executed_Declaration.pdf	55639 08b35273783997cf5f6c01e118a886a6f1dd6f1d	no	2
Warnings:					
Information:					
5	Fee Worksheet (SB06)	fee-info.pdf	38000 1f9ab306c550a95302f8c14464c1a2c30f15fc02	no	2
Warnings:					
Information:					
			Total Files Size (in bytes):	1853042	

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New Applications Under 35 U.S.C. 111
If a new application is being filed and the application includes the necessary components for a filing date (see 37 CFR 1.53(b)-(d) and MPEP 506), a Filing Receipt (37 CFR 1.54) will be issued in due course and the date shown on this Acknowledgement Receipt will establish the filing date of the application.

National Stage of an International Application under 35 U.S.C. 371
If a timely submission to enter the national stage of an international application is compliant with the conditions of 35 U.S.C. 371 and other applicable requirements a Form PCT/DO/EO/903 indicating acceptance of the application as a national stage submission under 35 U.S.C. 371 will be issued in addition to the Filing Receipt, in due course.

New International Application Filed with the USPTO as a Receiving Office
If a new international application is being filed and the international application includes the necessary components for an international filing date (see PCT Article 11 and MPEP 1810), a Notification of the International Application Number and of the International Filing Date (Form PCT/RO/105) will be issued in due course, subject to prescriptions concerning national security, and the date shown on this Acknowledgement Receipt will establish the international filing date of the application.

IN THE
UNITED STATES PATENT AND TRADEMARK OFFICE

Appl. No.	:	13/601,481	Confirmation No.	:	9349
Applicant	:	Steven A. Borror, et al.	Customer No.	:	26720
Assignee	:	Liebert Corporation	Atty Docket No.	:	0021944-080US-1
Filed	:	August 31, 2012			

Commissioner for Patents
P.O. Box 1450
Alexandria, VA 22313-1450

Applicants submitted continuation patent application number 13/601,481 on August 31, 2012. A typographical error was made in the Application Data Sheet for the entry of inventor Steven C. Sillato in the first block of the given name. The error has been corrected and the Application Data Sheet is hereby resubmitted.

No fee is believed to be required in conjunction with resubmission of the Application Data Sheet. However, should any fees be deemed necessary in order to affect this correction, the Commissioner is hereby authorized to deduct any required fees from the Locke Lord Bissell & Liddell, LLP Deposit Account No. 12-1322, referencing matter number **0021944-080US-1**.

Respectfully submitted,

Locke Lord, LLP

By /D. Brit Nelson/
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August 31, 2012

Application Data Sheet 37 CFR 1.76	Attorney Docket Number	0021944-080US-1
	Application Number	

Title of Invention	COOLING SYSTEM FOR HIGH DENSITY HEAT LOAD

The application data sheet is part of the provisional or nonprovisional application for which it is being submitted. The following form contains the bibliographic data arranged in a format specified by the United States Patent and Trademark Office as outlined in 37 CFR 1.76.
This document may be completed electronically and submitted to the Office in electronic format using the Electronic Filing System (EFS) or the document may be printed and included in a paper filed application.

Secrecy Order 37 CFR 5.2

☐ Portions or all of the application associated with this Application Data Sheet may fall under a Secrecy Order pursuant to 37 CFR 5.2 (Paper filers only. Applications that fall under Secrecy Order may not be filed electronically.)

Applicant Information:

Applicant 1 Remove

Applicant Authority	◉ Inventor	○ Legal Representative under 35 U.S.C. 117	○ Party of Interest under 35 U.S.C. 118	
Prefix	**Given Name**	**Middle Name**	**Family Name**	**Suffix**
	Steven	A.	BORROR	

Residence Information (Select One) ◉ US Residency ○ Non US Residency ○ Active US Military Service

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Citizenship under 37 CFR 1.41(b) i	US				

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Postal Code	43085	**Country** i	US

Applicant 2 Remove

Applicant Authority	◉ Inventor	○ Legal Representative under 35 U.S.C. 117	○ Party of Interest under 35 U.S.C. 118	
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Citizenship under 37 CFR 1.41(b) i	US				

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Postal Code	43083	**Country** i	US

Applicant 3 Remove

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City	Columbus	**State/Province**	OH	**Country of Residence** i	US

Application Data Sheet 37 CFR 1.76	Attorney Docket Number	0021944-080US-1
	Application Number	

Title of Invention	COOLING SYSTEM FOR HIGH DENSITY HEAT LOAD

Citizenship under 37 CFR 1.41(b) i	US		
Mailing Address of Applicant:			
Address 1	c/o Liebert Corporation		
Address 2	1050 Dearborn Drive		
City	Columbus	**State/Province**	OH
Postal Code	43083	**Country i**	US

Applicant 4 Remove

Applicant Authority	◉ Inventor	○ Legal Representative under 35 U.S.C. 117	○ Party of Interest under 35 U.S.C. 118	
Prefix	**Given Name**	**Middle Name**	**Family Name**	**Suffix**
	Steven	M.	MADARA	

Residence Information (Select One)	◉ US Residency	○ Non US Residency	○ Active US Military Service
City Dublin	**State/Province** OH	**Country of Residence i**	US
Citizenship under 37 CFR 1.41(b) i	US		
Mailing Address of Applicant:			
Address 1	c/o Liebert Corporation		
Address 2	1050 Dearborn Drive		
City Columbus		**State/Province**	OH
Postal Code	43083	**Country i**	US

Applicant 5 Remove

Applicant Authority	◉ Inventor	○ Legal Representative under 35 U.S.C. 117	○ Party of Interest under 35 U.S.C. 118	
Prefix	**Given Name**	**Middle Name**	**Family Name**	**Suffix**
	Reasey	J.	MAM	

Residence Information (Select One)	◉ US Residency	○ Non US Residency	○ Active US Military Service
City Westerville	**State/Province** OH	**Country of Residence i**	US
Citizenship under 37 CFR 1.41(b) i	US		
Mailing Address of Applicant:			
Address 1	c/o Liebert Corporation		
Address 2	1050 Dearborn Drive		
City Columbus		**State/Province**	OH
Postal Code	43083	**Country i**	US

Applicant 6 Remove

Applicant Authority	◉ Inventor	○ Legal Representative under 35 U.S.C. 117	○ Party of Interest under 35 U.S.C. 118	
Prefix	**Given Name**	**Middle Name**	**Family Name**	**Suffix**
	Stephen	C.	SILLATO	

Residence Information (Select One)	◉ US Residency	○ Non US Residency	○ Active US Military Service
City Westerville	**State/Province** OH	**Country of Residence i**	US
Citizenship under 37 CFR 1.41(b) i	US		

Application Data Sheet 37 CFR 1.76	Attorney Docket Number	0021944-080US-1
	Application Number	
Title of Invention	COOLING SYSTEM FOR HIGH DENSITY HEAT LOAD	

Mailing Address of Applicant:			
Address 1	c/o Liebert Corporation		
Address 2	1050 Dearbom Drive		
City	Columbus	State/Province	OH
Postal Code	43083	Country i	US
All Inventors Must Be Listed - Additional Inventor Information blocks may be generated within this form by selecting the **Add** button.			Add

Correspondence Information:

Enter either Customer Number or complete the Correspondence Information section below. For further information see 37 CFR 1.33(a).		
☐ **An Address is being provided for the correspondence Information of this application.**		
Customer Number	26720	
Email Address	hoip@lockelord.com	Add Email Remove Email

Application Information:

Title of the Invention	COOLING SYSTEM FOR HIGH DENSITY HEAT LOAD		
Attorney Docket Number	0021944-080US-1	Small Entity Status Claimed	☐
Application Type	Nonprovisional		
Subject Matter	Utility		
Suggested Class (if any)		Sub Class (if any)	
Suggested Technology Center (if any)			
Total Number of Drawing Sheets (if any)	3	Suggested Figure for Publication (if any)	2

Publication Information:

☐	Request Early Publication (Fee required at time of Request 37 CFR 1.219)
☐	**Request Not to Publish.** I hereby request that the attached application not be published under 35 U.S. C. 122(b) and certify that the invention disclosed in the attached application **has not and will not** be the subject of an application filed in another country, or under a multilateral international agreement, that requires publication at eighteen months after filing.

Representative Information:

Representative information should be provided for all practitioners having a power of attorney in the application. Providing this information in the Application Data Sheet does not constitute a power of attorney in the application (see 37 CFR 1.32).
Enter either Customer Number or complete the Representative Name section below. If both sections are completed the Customer Number will be used for the Representative Information during processing.

Please Select One:	◉ Customer Number	○ US Patent Practitioner	○ Limited Recognition (37 CFR 11.9)
Customer Number	26720		

Application Data Sheet 37 CFR 1.76	Attorney Docket Number	0021944-080US-1
	Application Number	
Title of Invention	COOLING SYSTEM FOR HIGH DENSITY HEAT LOAD	

Domestic Benefit/National Stage Information:

This section allows for the applicant to either claim benefit under 35 U.S.C. 119(e), 120, 121, or 365(c) or indicate National Stage entry from a PCT application. Providing this information in the application data sheet constitutes the specific reference required by 35 U.S.C. 119(e) or 120, and 37 CFR 1.78(a)(2) or CFR 1.78(a)(4), and need not otherwise be made part of the specification.

Prior Application Status	Patented				Remove
Application Number	Continuity Type	Prior Application Number	Filing Date (YYYY-MM-DD)	Patent Number	Issue Date (YYYY-MM-DD)
	Continuation of	10904889	2004-12-02	8261565	2012-09-11

Prior Application Status	Expired		Remove
Application Number	Continuity Type	Prior Application Number	Filing Date (YYYY-MM-DD)
10904889	non provisional of	60527527	2003-12-05

Additional Domestic Benefit/National Stage Data may be generated within this form by selecting the **Add** button. Add

Foreign Priority Information:

This section allows for the applicant to claim benefit of foreign priority and to identify any prior foreign application for which priority is not claimed. Providing this information in the application data sheet constitutes the claim for priority as required by 35 U.S.C. 119(b) and 37 CFR 1.55(a).

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Application Number	Country i	Parent Filing Date (YYYY-MM-DD)	Priority Claimed
			○ Yes ◉ No

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Assignee 1 Remove

If the Assignee is an Organization check here. ☒

Organization Name	LIEBERT CORPORATION		
Mailing Address Information:			
Address 1	1050 Dearborn Drive		
Address 2			
City	Columbus	**State/Province**	OH
Country i	US	Postal Code	43083
Phone Number		Fax Number	
Email Address			

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Application Data Sheet 37 CFR 1.76	Attorney Docket Number	0021944-080US-1
	Application Number	

Title of Invention	COOLING SYSTEM FOR HIGH DENSITY HEAT LOAD

Signature:

A signature of the applicant or representative is required in accordance with 37 CFR 1.33 and 10.18. Please see 37 CFR 1.4(d) for the form of the signature.					
Signature	/D. Brit Nelson/			Date (YYYY-MM-DD)	2012-08-31
First Name	D. Brit	Last Name	NELSON	Registration Number	40370

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2. A record from this system of records may be disclosed, as a routine use, in the course of presenting evidence to a court, magistrate, or administrative tribunal, including disclosures to opposing counsel in the course of settlement negotiations.

3. A record in this system of records may be disclosed, as a routine use, to a Member of Congress submitting a request involving an individual, to whom the record pertains, when the individual has requested assistance from the Member with respect to the subject matter of the record.

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5. A record related to an International Application filed under the Patent Cooperation Treaty in this system of records may be disclosed, as a routine use, to the International Bureau of the World Intellectual Property Organization, pursuant to the Patent Cooperation Treaty.

6. A record in this system of records may be disclosed, as a routine use, to another federal agency for purposes of National Security review (35 U.S.C. 181) and for review pursuant to the Atomic Energy Act (42 U.S.C. 218(c)).

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EFS ID:	13642848
Application Number:	13601481
International Application Number:	
Confirmation Number:	9349
Title of Invention:	COOLING SYSTEM FOR HIGH DENSITY HEAT LOAD
First Named Inventor/Applicant Name:	Steven A. BORROR
Customer Number:	26720
Filer:	David B. Nelson/Neva Dare
Filer Authorized By:	David B. Nelson
Attorney Docket Number:	0021944-080US-1
Receipt Date:	31-AUG-2012
Filing Date:	
Time Stamp:	18:13:18
Application Type:	Utility under 35 USC 111(a)

Payment information:

Submitted with Payment	no

File Listing:

Document Number	Document Description	File Name	File Size(Bytes)/ Message Digest	Multi Part /.zip	Pages (if appl.)
1	Miscellaneous Incoming Letter	0021944-080US-1_Transmittal_Ltr_Corrected_ADS.pdf	100369 2b9d012c9c9efdff156c7469bfdd3bb85bbc2641	no	1

Warnings:

Information:

2	Application Data Sheet	0021944-080US-1_ADS_corrected.pdf	1421621 81621bf51eb842ba5022a36598cd6d6c4c97cb2a	no	6
Warnings:					
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		Total Files Size (in bytes):	1521990		

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If a new application is being filed and the application includes the necessary components for a filing date (see 37 CFR 1.53(b)-(d) and MPEP 506), a Filing Receipt (37 CFR 1.54) will be issued in due course and the date shown on this Acknowledgement Receipt will establish the filing date of the application.

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If a timely submission to enter the national stage of an international application is compliant with the conditions of 35 U.S.C. 371 and other applicable requirements a Form PCT/DO/EO/903 indicating acceptance of the application as a national stage submission under 35 U.S.C. 371 will be issued in addition to the Filing Receipt, in due course.

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INFORMATION DISCLOSURE STATEMENT BY APPLICANT (**Not for submission under 37 CFR 1.99)**		
	Application Number	13601481
	Filing Date	2012-08-31
	First Named Inventor	BORROR, Steven
	Art Unit	3744
	Examiner Name	Unknown
	Attorney Docket Number	0021944-080US-1

U.S.PATENTS						Remove
Examiner Initial*	Cite No	Patent Number	Kind Code[1]	Issue Date	Name of Patentee or Applicant of cited Document	Pages,Columns,Lines where Relevant Passages or Relevant Figures Appear
	1	5150277		1992-09-22	Bainbridge	
	2	6564858		2003-05-20	Stahl	
	3	6644384		2003-05-01	Stahl	
	4	5035628		1991-07-30	Casciotti et al.	
	5	5161087		1992-11-03	Frankeny et al.	
	6	5273438		1993-12-28	Bradley et al.	
	7	5376008		1994-12-27	Rodriguez	
	8	5395251		1995-03-07	Rodriguez et al.	

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	Examiner Name	Unknown
	Attorney Docket Number	0021944-080US-1

	9	5402313		1995-03-28	Casperson et al.	
	10	5410448		1995-04-25	Barker et al.	
	11	6046908		2000-04-04	Feng	
	12	6115242		2000-09-05	Lambrecht	
	13	6158502		2000-12-12	Thomas	
	14	6167948		2001-01-02	Thomas	
	15	6310773		2001-10-30	Yusuf et al.	
	16	6416330		2002-07-09	Yatskov	
	17	6435266		2002-08-20	Wu	
	18	6515862		2003-02-04	Wong et al.	
	19	6519955		2003-02-18	Marsala	

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Art Unit	3744
Examiner Name	Unknown
Attorney Docket Number	0021944-080US-1

	20	6679081		2004-01-20	Marsala	
	21	6772604		2004-08-10	Bash et al.	
	22	6992889		2006-01-31	Kashiwagi et al.	
	23	5329425		1994-07-12	Leyssens et al.	
	24	6550530		2003-04-22	Bilski	
	25	6999316		2006-02-14	Hamman	
	26	6258293		2001-07-10	Iizuka et al.	
	27	4756164		1988-07-12	James et al.	
	28	6557624		2003-05-06	Stahl et al.	
	29	4315300		1982-02-09	Parmerlee et al.	
	30	6185098		2001-02-06	Benavides	

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	First Named Inventor: BORROR, Steven
	Art Unit: 3744
	Examiner Name: Unknown
	Attorney Docket Number: 0021944-080US-1

	31	5603375		1997-02-18	Salt	
	32	3317798		1967-05-02	Chu et al.	
	33	5847927		1998-12-08	Minning et al.	
	34	6208510		2001-03-27	Trudeau et al.	
	35	6628520		2003-09-30	Patel et al.	
	36	5414591		1995-05-09	Kimura	
	37	6305180		2001-10-23	Miller et al.	
	38	4344296		1982-08-17	Staples et al.	
	39	5400615		1995-03-28	Pearson	
	40	5987902		1999-11-23	Scaringe et al.	
	41	6305463		2001-10-23	Salmonson	

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	42	5737923		1998-04-14	Gilley et al.	
	43	5713413		1998-02-03	Osakabe et al.	
	44	5406807		1995-01-18	Ashiwake et al.	
	45	5333677		1994-08-02	Molivadas	
	46	4314601		1982-02-09	Guiffre et al.	
	47	3817321		1974-06-18	Von Cube et al.	
	48	3774677		1973-11-27	Antoneti et al.	
	49	6679081		2004-01-20	Marsala	
	50	6508301		2003-01-21	Marsala	
	51	5984647		1999-11-16	Miyamoto et al.	
	52	2244312		1941-06-03	Newton	

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	Art Unit	3744
	Examiner Name	Unknown
	Attorney Docket Number	0021944-080US-1

	53	5522452		1996-06-04	Mizuno et al.	
	54	6742345		2004-06-01	Carr	
	55	5054542		1991-10-08	Young et al.	
	56	6827135		2004-12-07	Kramer et al.	
	57	4238933		1980-12-16	Coombs	
	58	4019679		1977-04-26	Vogt et al.	
	59	6148626		2000-11-21	Iwamoto	
	60	3005321		1961-10-24	Devery	
	61	5709100		1998-01-20	Baer et al.	
	62	7236359		2004-04-01	Strobel	
	63	7469555		2008-12-30	Taras et al.	

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	First Named Inventor	BORROR, Steven	
	Art Unit		3744
	Examiner Name	Unknown	
	Attorney Docket Number		0021944-080US-1

	64	4514746		1985-04-30	Lundqvist	
	65	6105387		2000-08-22	Hong	
	66	4308042		1981-12-29	Ecker	
	67	6976529		2005-12-20	Kester	
	68	6298677		2001-10-09	Bujak, Jr.	
	69	6205803		2001-03-27	Scaringe	
	70	5261251		1993-11-16	Galiyano	
	71	5335508		1994-08-09	Tippmann	
	72	6460355		2002-10-08	Trieskey	

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INFORMATION DISCLOSURE STATEMENT BY APPLICANT (Not for submission under 37 CFR 1.99)		
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	Filing Date	2012-08-31
	First Named Inventor	BORROR, Steven
	Art Unit	3744
	Examiner Name	Unknown
	Attorney Docket Number	0021944-080US-1

	1	20030182949		2003-10-02	Carr	
	2	20020184908		2002-12-12	Brotz et al.	
	3	20030126872		2003-07-10	Harano et al.	
	4	20030182949		2003-10-02	Carr	
	5	20020066280		2002-06-06	Ohkawara	
	6	20040100770		2004-05-27	Chu et al.	
	7	20060101837		2006-05-18	Manole	
	8	20030061824		2003-04-03	Marsala	
	9	20020124585		2002-09-12	Bash et al.	
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	1	2113012	GB	A	1983-07-27	FLAEKT AB		☐
	2	19804901	DE	A	1999-08-19	LOH KG RITTAL WERK		☐
	3	1357778	EP		2003-10-29	MITSUBISHI DENKA KABUSHIKI KAISHA		☐
	4	29908370	DE		1999-09-30	BADER ENGINEERING GmbH		☐
	5	0281762	EP		1988-09-14	TAKENAKA KOMUTEN CO. LTD.		☐
	6	1143778	EP		2006-12-13	Marsala		☐

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	1	BAER, D.B. "Emerging Cooling Requirements & Systems in Telecommunications Spaces." Telecommunications Energy Conference 2001, October 14-18, 2001, p 95-100.	☐

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	2	VOGEL, Marlin et al. "Low Profile Heat Sink Cooling Technologies for Next Generation CPU Thermal Designs." Electronic Cooling Online, 2-17-2005, 11 pages.	☐
	3	WEBB, Warren. "Take the heat: Cool that hot embedded design." EDN, 5-13-2004, 5 pages.	☐
	4	JAMSTEC/Earth Simulator Center, "Processor Node (PN) Cabinet." one page. http://www.es.jamstec.go.jp/esc/eng/Hardware/pnc.html [accessed 03/05/2004].	☐
	5	HANNEMANN, Robert et al. "Pumped Liquid Multiphase Cooling." ASME, 2004, IMECE 2004, Paper IMECE2004-60699, Anaheim, California, 5 pages.	☐
	6	"Thermal Form & Function - Rack Cooling System (RCS)." Thermal Form & Function LLC, 2005, Manchester, MA, one page. http://www.thermalformandfunction.com/racksystem.html [accessed 05/11/2006].	☐
	7	PITASI, M. "Thermal Management System Using Pumped Liquid R-134a with Two Phase Heat Transfer." Thermal Form & Function LLC, March 2002, Manchester, MA, pages 1-9. http://www.coolingzone.com/Guest/News/NL_MAR_2002/TFF/Tff.html	☐
	8	MARSALA, Joe. "Pumped Liquid/Two Phase Cooling for High Performance Systems." Thermal Form & Function LLC, 05/13/2003, Scottsdale, Arizona, 19 pages.	☐
	9	"Managing Extreme Heat Cooling Strategies for High-Density Computer Systems." Liebert Corporation, 12-07-2003, Columbus, Ohio, 16 pages.	☐
	10	KLINGEBERG, G. "Kombination Spart Strom bei der Schaltschrank-klimatisierung." Elekotrotechnik, Vogel Verley K. G., Wurzburg DE, November 5, 1998.	☐
	11	MIOT, F., copy of International Search Report for International Patent Application No. PCT/US2005/040745, dated 30 March 2006, European Patent Office.	☐
	12	MIOT, F., copy of Written Opinion for International Patent Application No. PCT/US2005/040745, dated 30 March 2006, European Patent Office.	☐

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	First Named Inventor	BORROR, Steven
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	Examiner Name	Unknown
	Attorney Docket Number	0021944-080US-1

	13	International Search Report for Corresponding International Patent Application No. PCT/US2006/028088.	☐
	14	Written Opinion of the International Searching Authority for Corresponding International Patent Application No. PCT/US2006/028088.	☐
	15	Emerson Network Power - Liebert Foundation; "Protection for Today's Needs...and Tomorrow's Opportunities; Enclosure Systems." 2003, p 1-16.	☐
	16	Emerson Network Power - Liebert Foundation; "Wall Mount Equipment Enclosures Designed for Tight Spaces." 2003. p 1-2.	☐
	17	SZILAGYI, B., copy of Extended European Search Report for European Application No. EP 10 18 3253.3, Munich, dated 26 January 2011.	☐
	18	American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., "Heating, Ventilating, and Air-Conditioning Systems and Equipment", SI Edition, 1996 ASHRAE Handbook, page 12.11-12.12, Atlanta, Georgia.	☐
	19	S.F. PEARSON, "Development of Improved Secondary Refrigerants", The Proceedings of The Institute of Refrigeration, 4 March 1993, The Institute of Refrigeration, Volume 89, 1992-1993, London.	☐
	20	SZILAGYI, B., copy of European Communication for European Patent Application No. 04 812 840.9, dated 19 February 2009.	☐
	21	Robert A. SWANSON, copy of letter to Albert B. Deaver, Jr., 2010-06-29 (unpublished).	☐
	22	SZILAGYI, B., copy of Extended Search Report for European Application No. EP 04 812 840.9, Munich, dated 6 July 2007.	☐
	23	SZILAGYI, B., copy of European Communication for European Patent Application No. 04 812 840.9, dated 28 March 2008.	☐

INFORMATION DISCLOSURE STATEMENT BY APPLICANT (Not for submission under 37 CFR 1.99)	Application Number	13601481
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	24	SZILAGYI, B., copy of International Search Report for International Patent Application No. PCT/US2004/040407, dated 30 June 2005, European Patent Office.	☐
	25	SZILAGYI, B., copy of Written Opinion for International Patent Application No. PCT/US2004/040407, dated 30 June 2005, European Patent Office.	☐

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EXAMINER SIGNATURE

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*EXAMINER: Initial if reference considered, whether or not citation is in conformance with MPEP 609. Draw line through a citation if not in conformance and not considered. Include copy of this form with next communication to applicant.

[1] See Kind Codes of USPTO Patent Documents at www.USPTO.GOV or MPEP 901.04. [2] Enter office that issued the document, by the two-letter code (WIPO Standard ST.3). [3] For Japanese patent documents, the indication of the year of the reign of the Emperor must precede the serial number of the patent document. [4] Kind of document by the appropriate symbols as indicated on the document under WIPO Standard ST.16 if possible. [5] Applicant is to place a check mark here if English language translation is attached.

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CERTIFICATION STATEMENT

Please see 37 CFR 1.97 and 1.98 to make the appropriate selection(s):

☐ That each item of information contained in the information disclosure statement was first cited in any communication from a foreign patent office in a counterpart foreign application not more than three months prior to the filing of the information disclosure statement. See 37 CFR 1.97(e)(1).

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☐ See attached certification statement.

☐ The fee set forth in 37 CFR 1.17 (p) has been submitted herewith.

☒ A certification statement is not submitted herewith.

SIGNATURE

A signature of the applicant or representative is required in accordance with CFR 1.33, 10.18. Please see CFR 1.4(d) for the form of the signature.

Signature	/D. Brit Nelson/	Date (YYYY-MM-DD)	2012-09-07
Name/Print	D. Brit Nelson	Registration Number	40370

This collection of information is required by 37 CFR 1.97 and 1.98. The information is required to obtain or retain a benefit by the public which is to file (and by the USPTO to process) an application. Confidentiality is governed by 35 U.S.C. 122 and 37 CFR 1.14. This collection is estimated to take 1 hour to complete, including gathering, preparing and submitting the completed application form to the USPTO. Time will vary depending upon the individual case. Any comments on the amount of time you require to complete this form and/or suggestions for reducing this burden, should be sent to the Chief Information Officer, U.S. Patent and Trademark Office, U.S. Department of Commerce, P.O. Box 1450, Alexandria, VA 22313-1450. DO NOT SEND FEES OR COMPLETED FORMS TO THIS ADDRESS. **SEND TO: Commissioner for Patents, P.O. Box 1450, Alexandria, VA 22313-1450.**

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7. A record from this system of records may be disclosed, as a routine use, to the Administrator, General Services, or his/her designee, during an inspection of records conducted by GSA as part of that agency's responsibility to recommend improvements in records management practices and programs, under authority of 44 U.S.C. 2904 and 2906. Such disclosure shall be made in accordance with the GSA regulations governing inspection of records for this purpose, and any other relevant (i.e., GSA or Commerce) directive. Such disclosure shall not be used to make determinations about individuals.

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PTO/SB/08a (01-10)
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INFORMATION DISCLOSURE STATEMENT BY APPLICANT (**Not for submission under 37 CFR 1.99**)		
	Application Number	13601481
	Filing Date	2012-08-31
	First Named Inventor	BORROR, Steven
	Art Unit	3744
	Examiner Name	Unknown
	Attorney Docket Number	0021944-080US-1

U.S.PATENTS						Remove
Examiner Initial*	Cite No	Patent Number	Kind Code[1]	Issue Date	Name of Patentee or Applicant of cited Document	Pages,Columns,Lines where Relevant Passages or Relevant Figures Appear
	1	7051802		2006-05-30	Baer	
	2	3253646		1966-05-31	Koltuniak et al.	
	3	3334684		1967-08-08	Roush et al.	
	4	3749981		1973-07-31	Koltuniak et al.	
	5	3754596		1973-08-28	Ward, Jr.	
	6	4240499		1980-12-23	Kals	
	7	4495780		1985-01-29	Kaneko et al.	
	8	4514746		1985-04-30	Lundqvist	

INFORMATION DISCLOSURE STATEMENT BY APPLICANT (**Not for submission under 37 CFR 1.99)**	Application Number: 13601481
	Filing Date: 2012-08-31
	First Named Inventor: BORROR, Steven
	Art Unit: 3744
	Examiner Name: Unknown
	Attorney Docket Number: 0021944-080US-1

	9	5123478		1992-06-23	Hosaka	
	10	5220809		1993-06-22	Voss	
	11	5467607		1995-11-21	Harvey	
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	18	6026891		2000-02-22	Fujiyoshi et al.	
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INFORMATION DISCLOSURE STATEMENT BY APPLICANT (Not for submission under 37 CFR 1.99)		
	Application Number	13601481
	Filing Date	2012-08-31
	First Named Inventor	BORROR, Steven
	Art Unit	3744
	Examiner Name	Unknown
	Attorney Docket Number	0021944-080US-1

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	21	6299526		2001-10-09	Cowan et al.	
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	26	5570740		1996-11-05	Flores et al.	

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	2	20050120737		2005-06-09	Borror et al.	

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	Filing Date	2012-08-31
	First Named Inventor	BORROR, Steven
	Art Unit	3744
	Examiner Name	Unknown
	Attorney Docket Number	0021944-080US-1

	3	20050168945		2005-08-04	Coglitore	
	4	20020039280		2002-04-04	O'Connor et al.	
	5	20010042616		2001-11-22	Baer et al.	
	6	20030051859		2003-03-20	Chesser et al.	
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	10	20020117291		2002-08-29	Cheon	

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	Application Number	13601481
	Filing Date	2012-08-31
	First Named Inventor	BORROR, Steven
	Art Unit	3744
	Examiner Name	Unknown
	Attorney Docket Number	0021944-080US-1

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☐ That each item of information contained in the information disclosure statement was first cited in any communication from a foreign patent office in a counterpart foreign application not more than three months prior to the filing of the information disclosure statement. See 37 CFR 1.97(e)(1).

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☐ That no item of information contained in the information disclosure statement was cited in a communication from a foreign patent office in a counterpart foreign application, and, to the knowledge of the person signing the certification after making reasonable inquiry, no item of information contained in the information disclosure statement was known to any individual designated in 37 CFR 1.56(c) more than three months prior to the filing of the information disclosure statement. See 37 CFR 1.97(e)(2).

☐ See attached certification statement.

☐ The fee set forth in 37 CFR 1.17 (p) has been submitted herewith.

☒ A certification statement is not submitted herewith.

SIGNATURE

A signature of the applicant or representative is required in accordance with CFR 1.33, 10.18. Please see CFR 1.4(d) for the form of the signature.

Signature	/D. Brit Nelson/	Date (YYYY-MM-DD)	2012-09-07
Name/Print	D. Brit Nelson	Registration Number	40370

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Electronic Acknowledgement Receipt

EFS ID:	13683828
Application Number:	13601481
International Application Number:	
Confirmation Number:	9349
Title of Invention:	COOLING SYSTEM FOR HIGH DENSITY HEAT LOAD
First Named Inventor/Applicant Name:	Steven A. BORROR
Customer Number:	26720
Filer:	David B. Nelson/Danny Vara
Filer Authorized By:	David B. Nelson
Attorney Docket Number:	0021944-080US-1
Receipt Date:	07-SEP-2012
Filing Date:	
Time Stamp:	13:54:04
Application Type:	Utility under 35 USC 111(a)

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File Listing:

Document Number	Document Description	File Name	File Size(Bytes)/ Message Digest	Multi Part /.zip	Pages (if appl.)
1	Foreign Reference	ForRef_1_2113012.pdf	1100788 addfe9d207173089071550 7604a5e23bfe1a8c4d	no	14

Warnings:

Information:

2	Foreign Reference	ForRef_2_19804901.pdf	36740 6814851f1d31a316f33b01629824271fefc74d48	no	1
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3	Foreign Reference	ForRef_3_1357778.pdf	2400426 42909cf7bef3e6fa4fe1afe108021c6377bd678b	no	53
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4	Foreign Reference	ForRef_4_299.pdf	554949 237b88eb2c4c778236f099e1b6c3f9875841bbf8	no	16
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7	Non Patent Literature	NPL_1_Baer.pdf	449096 9cca609d9976f3cf55f4f06a12ca8e8ff18abb7f	no	6
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8	Non Patent Literature	NPL_2_Vogel.pdf	304174 68e3027182d1fe18182d783537e7818755ba0cd5	no	11
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		Total Files Size (in bytes):	16640169		

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(12) UK Patent Application (19) GB (11) 2 113 012 A

(21) Application No **8234285**
(22) Date of filing **1 Dec 1982**
(30) Priority data
(31) **8107178**
8206618
(32) **2 Dec 1981**
19 Nov 1982
(33) **Sweden (SE)**
(43) Application published
27 Jul 1983
(51) **INT CL^3**
H05K 7/20
(52) Domestic classification
H1R BK
U1S 2214 H1R
(56) Documents cited
GB 1563091
GB 1521464
GB 1336076
GB 1006053
(58) Field of search
H1R

(71) Applicant
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(72) Inventor
Gosta Lundqvist
(74) Agent and/or Address for Service
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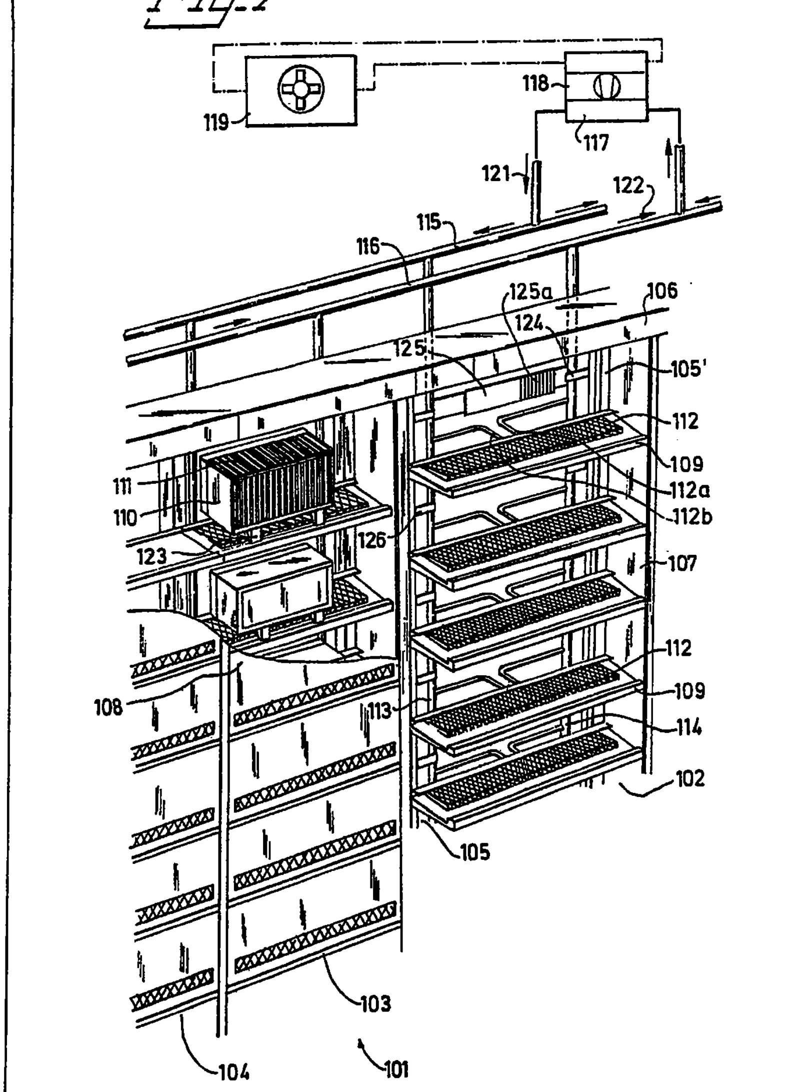
Chancery House, 53—64
Chancery Lane, London
WC2A 1SD

(54) **Apparatus for cooling telecommunications equipment in a rack**

(57) A rack contains a number of shelves (109) arranged on top of each other, which carry, among other things, board magazines (110) with printed circuit boards (111). An apparatus for cooling the printed circuit boards is based on the principle that one or more shelves in the rack are integrated each with its own heat exchanger, which is equipped with cooling flanges and at least one tubular coil. The heat exchanger is connected to a cooler (117) for a coolant. The heat exchanger (112) covers the greater part of the horizontal plane of the shelf. Extensive heat-conducting contact surfaces are provided between the board magazines and the body of the shelf and between the latter and the heat exchangers. Heat removal is also provided for by natural convection at the hot and cold surfaces and by radiation.

Fig. 7



GB 2 113 012 A

1:5

Fig. 1

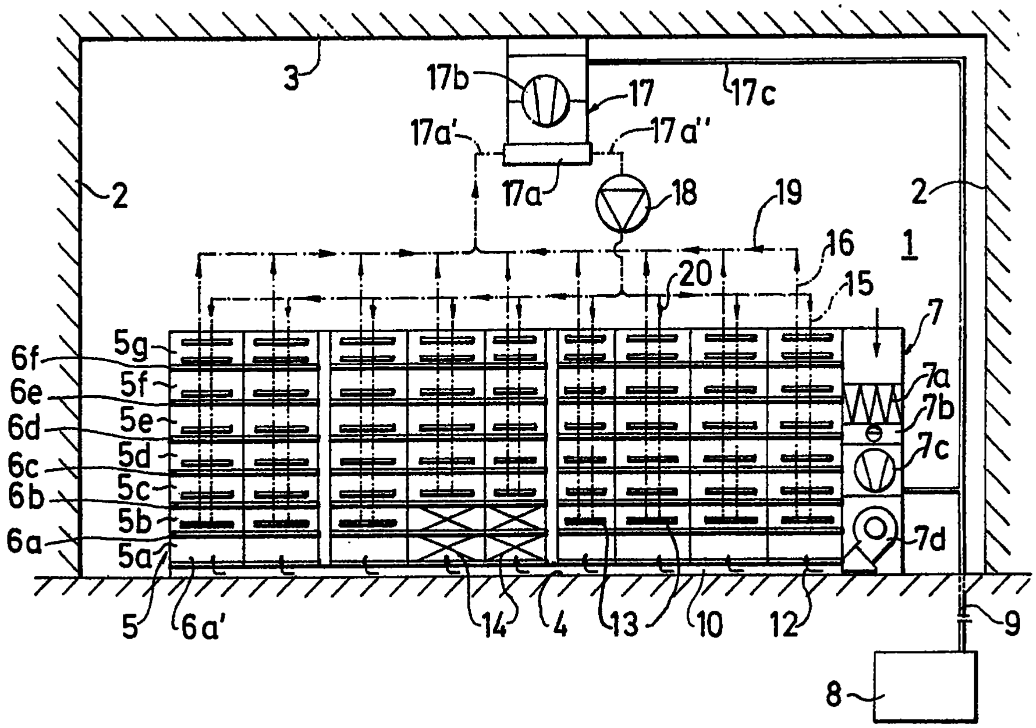


Fig. 4

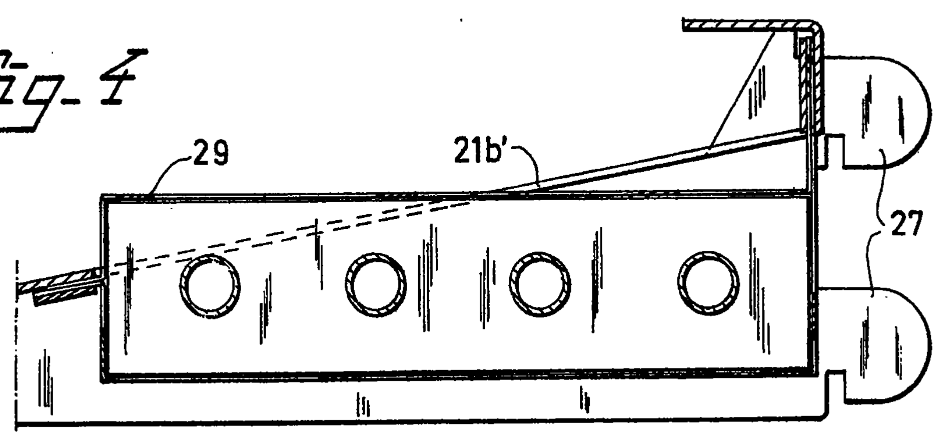


Fig. 5

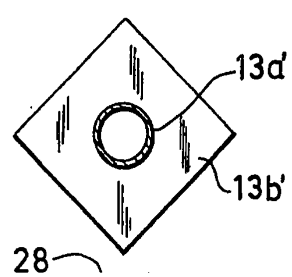


Fig. 6

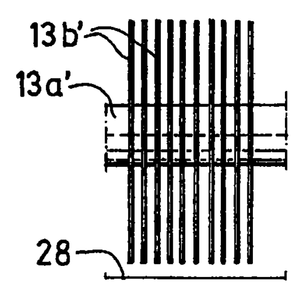


Fig. 2

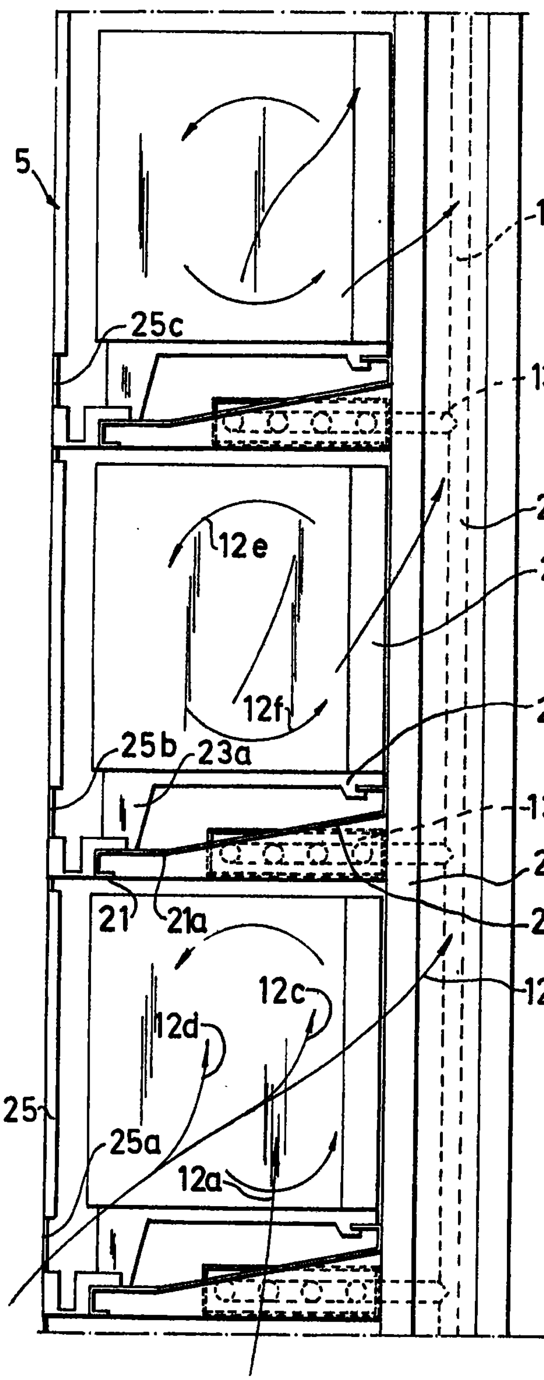

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Fig. 3

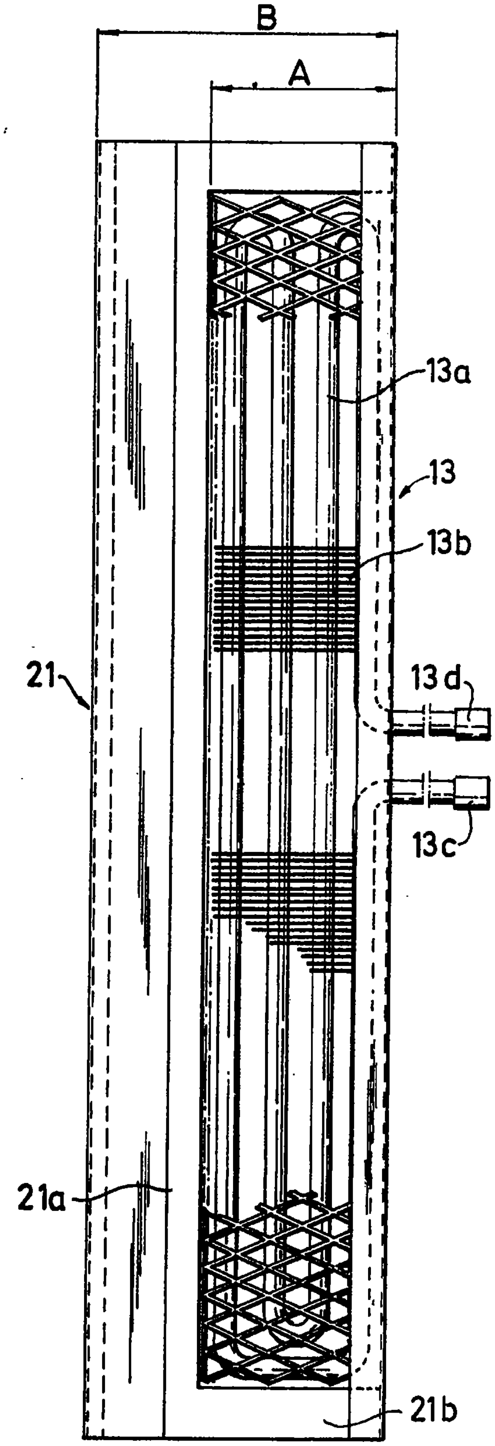
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Fig. 7

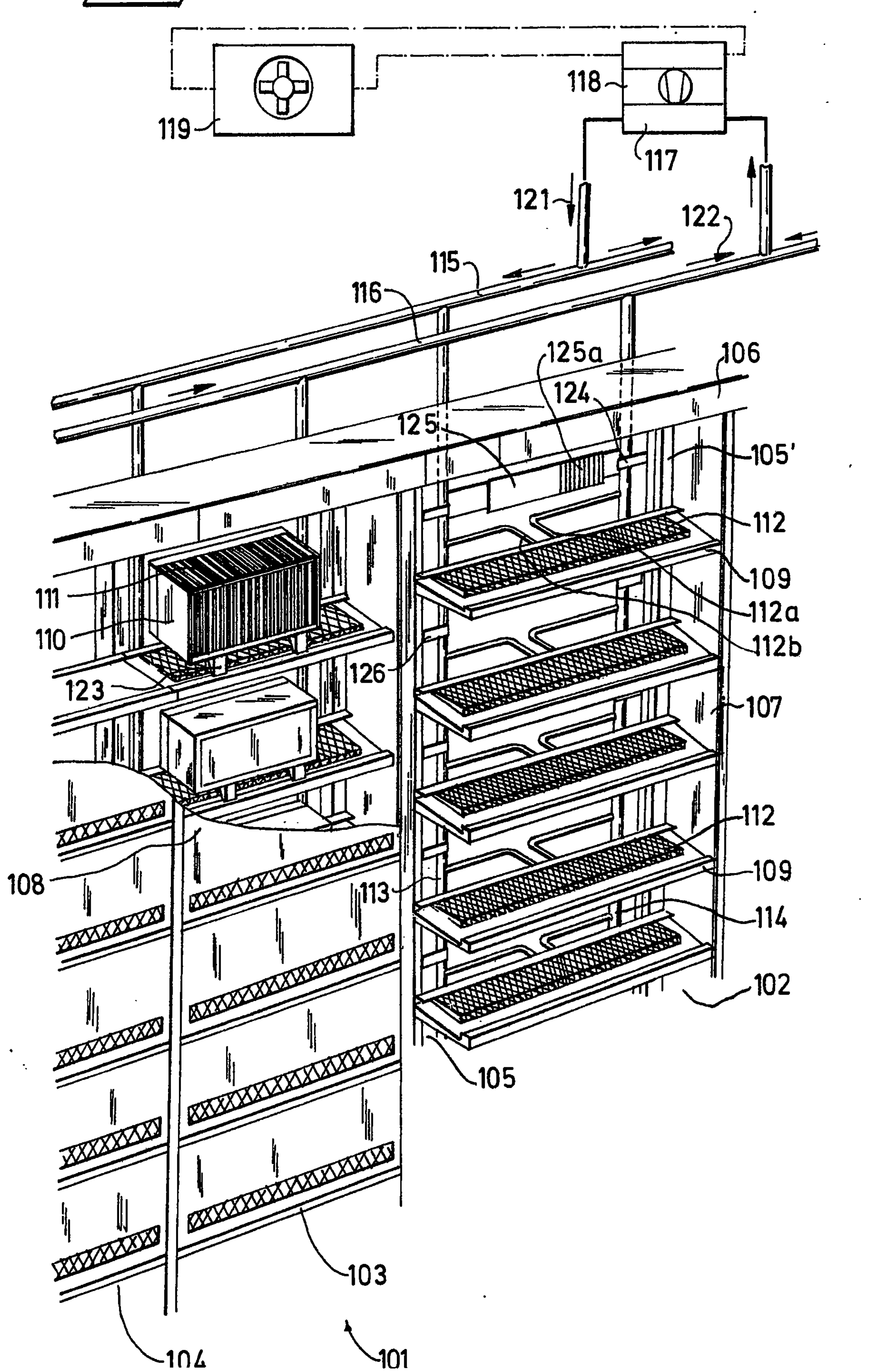

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Fig. 8

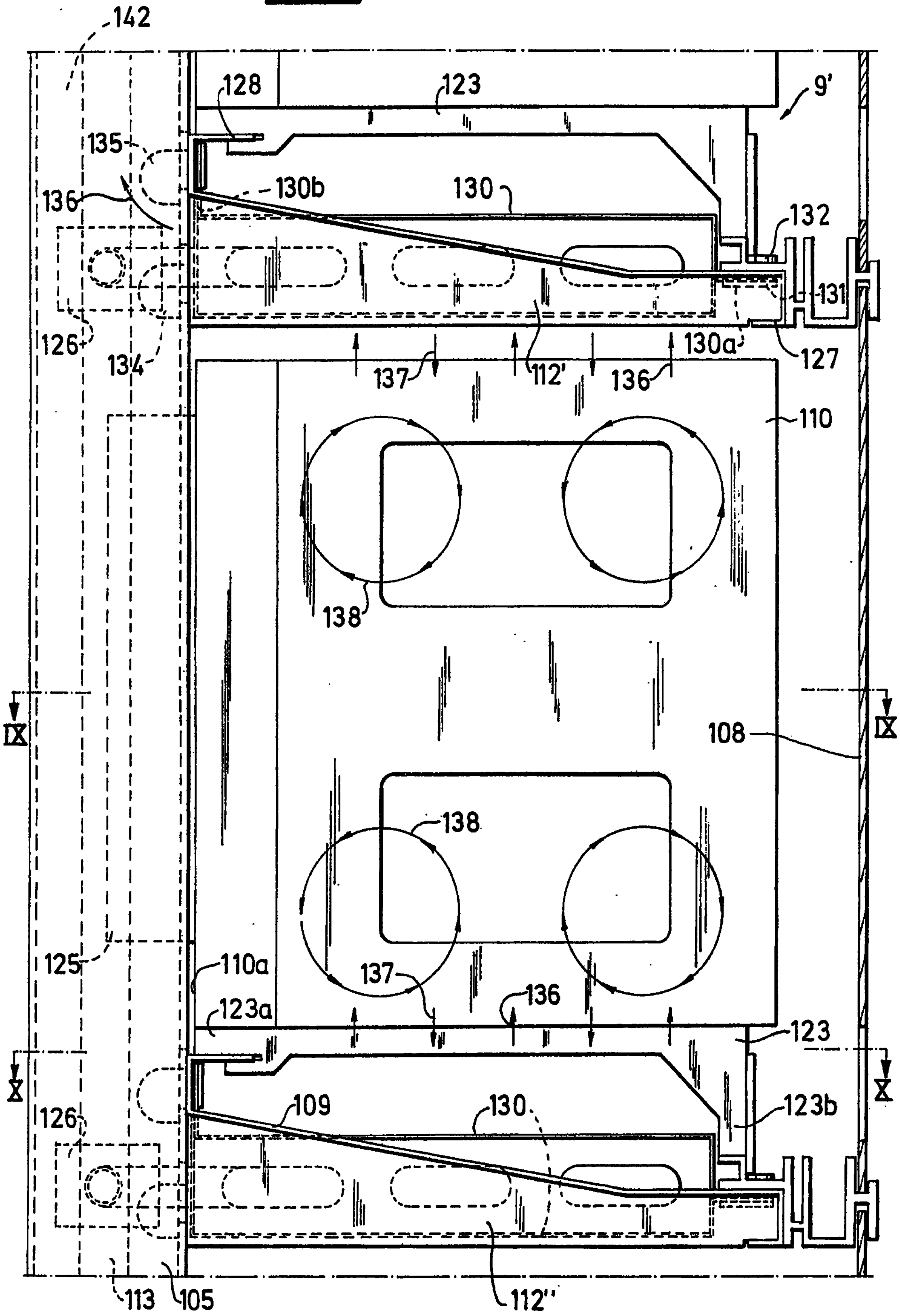
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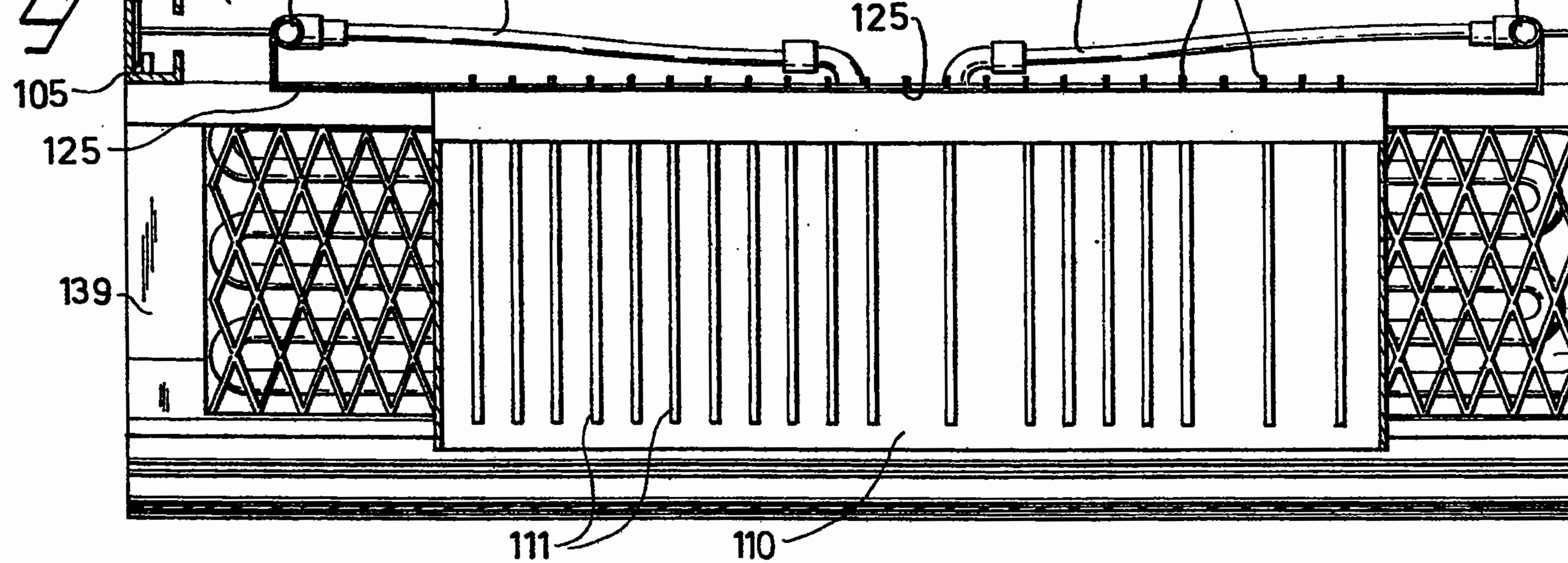
Fig-9



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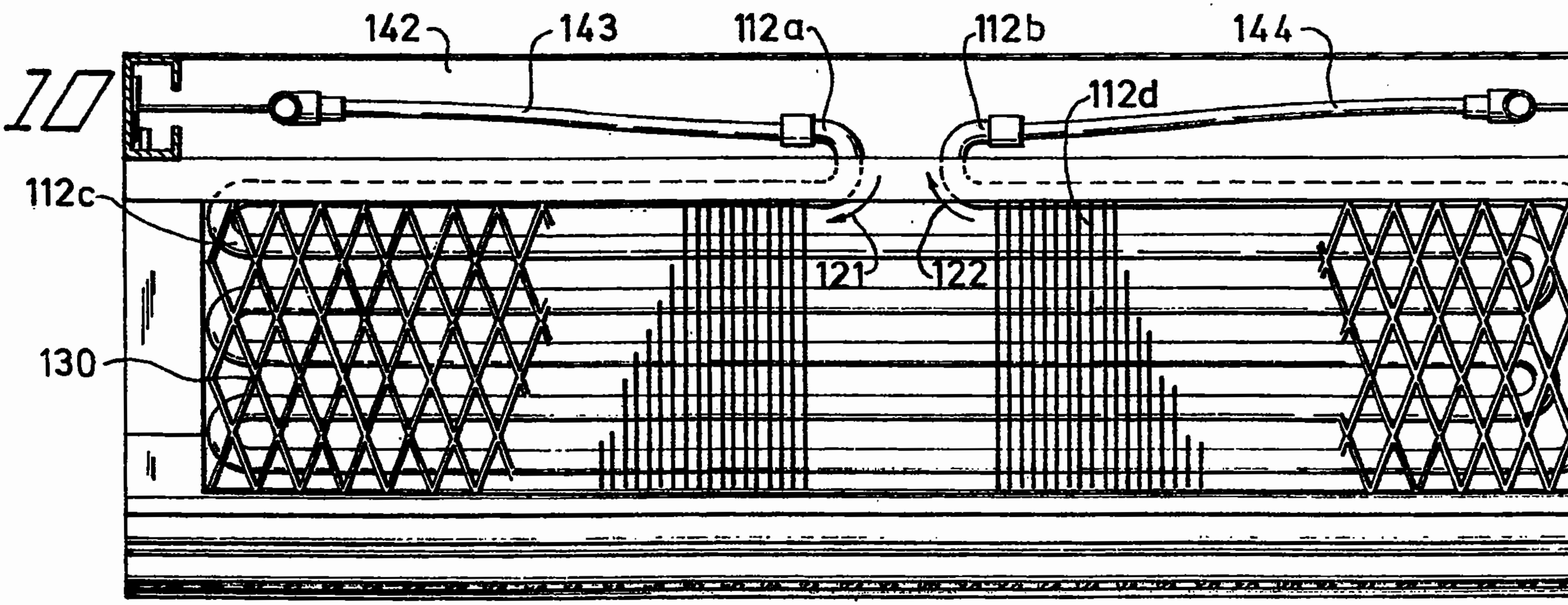
105
125
139
111
110

Fig-10



142
143
112a
112b
144
112d
112c
121
122
130

SPECIFICATION

Apparatus for cooling telecommunications equipment in a rack

Technical Field

The present invention relates to an apparatus for cooling telecommunications equipment mounted in a rack with a number of shelves located on top of each other, which equipment consists of component-carrying printed circuit boards housed in board magazines, which are in turn applied to the aforesaid shelves. The term "telecommunications equipment" is also used here to denote general electronic equipment mounted in racks that is used outside of the field of telecommunications as well.

Description of the Prior Art

A means of achieving cooling of telecommunications equipment mounted in one or more racks, for example at a telephone exchange, is already known. Such a rack may contain eight shelves, and two such racks generally form a so-called "double rack". The board magazines that are applied to the aforesaid shelves are thereby arranged to store, on end, a number of boards with printed circuits and/or components, such as relays, condensers, resistors, inductances etc. A magazine may contain 50 such boards, which means that a double rack may contain some 1 000 boards.

Heretofore, it has been proposed to cool such telecommunications equipment solely with the aid of so-called "open cooling systems", which comprise a compressed air source, for example in the form of a fan, whose initiated air stream is conducted from an air duct at the lower parts of the rack or racks and upwards in each rack so that it can pass the boards placed on end in the board magazines and be discharged at the top of the rack or racks out into the premises. The air is recirculated to a cooling unit, where the air can be cooled and dried before it is once again recirculated to the racks, and so on.

Description of the Present Invention

Technical Problem

A fervent desire exists to be able to increase the density of components on each printed circuit board so that a larger number of components can be accommodated within the same volume. This results in heat output from the boards in the racks that is difficult to handle with present-day cooling equipment for telecommunication racks. The heat outputs in question can be up to 10 Watts per board or more.

In light of the above, new ways must be chosen to obtain heat removal from the board magazines so that maximum component temperatures can be kept to a reasonable level, a must in order to ensure good function and long life of the components in question.

Solution

The main purpose of the present invention is to create an apparatus that solves the problem of efficient removal of the heat emitted by the printed circuit board components. What can thereby be said to be essentially characteristic for the new apparatus is that one or more shelves are integrated each with its own heat exchanger incorporating cooling flanges and at least one tubular coil, which is connected via the heat exchanger to a cooler for a coolant in the tubular coil; that the heat exchanger, viewed in the horizontal plane of the shelf, extends along the greater part of the shelf; that extensive heat-conducting metallic contact surfaces are arranged between the board magazines and the shelf and between the shelf and the heat exchanger; and that the aforesaid heat exchanger and cooler are arranged so that they are able to remove the aforementioned emitted heat even at a high packing density of the aforesaid telecommunications equipment in the rack and a high heat output from the said equipment, inter alia with the aid of heat conduction via the aforesaid extensive contact surfaces.

In further elaborations of the invention concept, specifications are recommended for the more detailed construction of the different parts of the apparatus. Thus, in one proposed embodiment the majority of the shelves in the rack are to be equipped each with its own heat exchanger, in addition to which the heat exchangers' tubular coils are interconnected and preferably connected to one and the same cooler. The cooler with appurtenant compressor may thereby take the form of a tubular evaporator or fluid cooler that is incorporated in a cooling circuit together with, inter alia, a condenser (water recooler). The latter, at least, should be located outside of the premises in which the rack or racks are set up.

The heat exchanger or heat exchangers and the appurtenant cooling equipment are thereby arranged and operated in such a manner that the surface temperatures of the components on the boards assume values that are acceptable in the context, for example max. about 75°C. The heat exchangers and the cooling equipment should preferably be arranged in such a manner that most of the components on each board assume relatively low temperatures, e.g. below 65°C. In principle, it is possible to arrange and operate the equipment in such a manner that lower component temperatures are achieved. In order to effect the above, each heat exchanger should be provided with a relatively large effective cooling surface area, in one embodiment preferably in excess of about 1.5 m^2. As an example, it can be mentioned that the cooling surface area can be chosen within the range 1.5—4 m^2. Each heat exchanger shall be provided with a relatively large cooling capacity, e.g. in excess of 150 Watts, and in this context it can be mentioned that the invention makes it possible to achieve very high cooling capacities from each heat exchanger, for example 500 Watts or more.

The integration between each heat exchanger and its associated shaft is further designed so that there is no undue mechanical weakening of the load-bearing capacity of the shelf, but rather the shelf can fulful its board-bearing functions.

According to the invention concept, the new apparatus shall be able to effect the cooling function essentially autonomously, i.e. without the aid of additional cooling equipment, for example in the form of row coolers, air conditioners and the like. It is, however, possible, if desired, to use the apparatus in combination with other cooling equipment, for example a row cooler.

Through the invention, the rack or racks that make use of the new apparatus will, in one embodiment, function as a cooling buffer in the premises where the rack or racks are set up, while simultaneously cooling the telecommunications equipment in the rack or racks.

In acordance with the concept of the invention, each shelf shall contain a horizontally extending cavity in which the heat exchanger for that shelf is mounted by means of a holding device. This holding device may take the form of a screen or grille running along the outer sides of the heat exchanger and be fastened along at least its long sides to the shelf. The shelf thereby consists of two essentially parallel load-bearing parts or beams, held together at their ends by side pieces and brackets, which can be of known type. The aforesaid cavity for the heat exchanger extends between the aforesaid parallel load-bearing parts. Cantilever supports also extend between the aforesaid load-bearing parts, and the middle sections of the cantilever supports extend over the aforesaid cavity. The cantilever supports in turn support the board magazines. The boards are placed on end in the board magazines and are enclosed entirely within the aforesaid board magazines. The board magazines are provided with through slots to permit air circulation past the sides of the boards and components. The board magazines incorporate a bottom part that supports the receptacles into which the boards can be inserted via matching plugs. The bottom part is also provided with through slots that permit air circulation.

Metallic contact surfaces thereby exist between the shelf and the aforesaid holding device for the heat exchanger and between the shelf and the board magazines. The extended contact surfaces are situated on, *inter alia*, the aforesaid cantilever supports. In one embodiment, the extended contact surfaces also include long channels by means of which the holding device for the heat exchanger is clamped to the shelf. The board magazines, the cantilever supports, the holding device for the heat exchanger and/or the shelf itself can, in addition, be in direct contact, e.g. via metallic plates, with the cooling system's lines or coils, whereby direct contact is also obtained between the cold surfaces on the aforesaid lines and the unit in question.

The above-described arrangement provides an effective indirect cooling function for the board components in the board magazines to complement the cooling function that is obtained through natural convection and radiation from the heat exchanger.

Through the proposed design principle, relatively large quantities of heat can be dissipated in this context by means of the heat exchanger and the appurtenant cooling equipment. Even if as large heat outputs as 10—15 Watts should be emitted from one or more boards in the board magazines, the design and operation of the heat exchangers and the appurtenant cooling equipment can be arranged to provide the necessary heat removal. The new arrangement can thereby effect removal of the heat developed in the racks without environmental nuisance, either by itself or in combination with supplementary cooling equipment, such as row coolers and air conditioners.

Despite a high packing density, the highest temperatures of the boards need not exceed predetermined values, e.g. about 75°C. Obviously, the life of the various components is prolonged by reduced temperature, and it can be mentioned in this context that if desired, it is possible to operate the system so that the temperatures of most of the components will be arround 65°C or lower.

Despite the above-mentioned advantages obtained with the invention, the main advantage is that the shelf construction and the rack system as a whole need not be subject to any modifications, aside from provision of the cavity for the heat exchanger. The heat exchangers in the rack will therefore not occupy any extra space, but rather only such space as was previously not utilized in the racks of known design. The integration can be executed in such a manner that the load-bearing capacity of the shelf is not unduly affected.

Description of the Drawings

A currently proposed embodiment of an apparatus that exhibits the significant characteristics of the invention will be described below with reference to appended drawings, where

Figure 1 shows, in schematic form premises, for example at a telephone exchange, with racks containing telecommunications equipment that is cooled by the new apparatus, the latter comprising a combination of shelf coolers and row coolers,

Figure 2 shows, in an end view, parts of a rack according to fig. 1,

Figure 3 shows, in a horizontal view, a heat exchanger mounted in a shelf in the rack according to fig. 2,

Figure 4 shows one embodiment of the heat exchanger and its mounting in a shelf in the rack,

Figure 5 shows, in an end view, an alternative embodiment of the heat exchanger,

Figure 6 shows a detailed view of a tube incorporated in the heat exchanger according to fig. 5 and cooling flanges arranged around this tube, with drop-collecting edges and drip tray

Figure 7 shows, in perspective viewed from in front and above, certain parts of a second embodiment of a rack utilizing the apparatus, in which another embodiment the shelf cooler provides the cooling function essentially autonomously, i.e. without the aid of e.g. the row cooler in figures 1—2,

Figure 8 shows, in a side view, shelves in a rack according to fig. 7 with apprutenant board magazines for printed circuit boards,

Figure 9 shows, in a horizontal section, the design of the board magazines, the shelf and the rack according to fig. 8 and

Figure 10 shows, in a horizontal section, the extent of the heat exchanger in the shelf according to Fig. 8.

Description of the Preferred Embodiment

Figures 1—6 illustrate an embodiment where the individual cooling function in the rack shelves has been combined with a row cooling function common to all racks. In figure 1, the premises or equivalent room are indicated by 1. The premises are bounded in this case by walls 2, a ceiling 3 and the floor 4. A number of racks 5 of known type with telecommunications equipment are set up on the premises. The racks 5 shown are arranged in rows and the premises may contain one or more such rows of racks. Each rack consists of a number of compartments arranged on top of one another, designated in the figure by 5a, 5b . . . 5g. The compartments are bounded by shelves, which are indicated by 6a', 6a, 6b . . . 6f. The shelves are designed to support various telecommunication units plus magazines with boards with printed wiring and various components (condensers, relays, resistors etc.). By "shelf" is meant here cantilevers and the like. The aforesaid boards are arranged on end in the magazines.

At the end of each row is a refrigeration module 7 that operates by means of, for example, direct expansion and is of known type. The refrigeration module in question here is sold on the market by Fläkt AB and carries the code designation KDAX. The refrigeration module comprises a filter 7a, a cooling coil 7b, compressors 7c and a fan 7d. The compressors are connected to a condenser 8 or water cooler located outside of the premises, and the regrigerant line is indicated by 9. One or more ducts 10 pass under the racks.

The air discharged by the fan 7d is blown into the aforesaid ducts and distributed in known manner between the racks, creating upward-directed and preferably dry air streams, indicated in the figure by 12.

Heat exchangers, described in greater detail below, are installed in the different shelves in the different racks. In the embodiment illustrated, most of the aforesaid shelves are equipped with a heat exchanger 13, and the new apparatus functions more effectively the more shelves are equipped with heat exchangers. 60—100% of the shelves should preferably be equipped with heat exchangers. In the middle racks shown in the figure, units 14 (not specified in any greater detail here) are installed in the lower parts. These compartments are not equipped with heat exchangers. The same applies to all bottom compartments 6a' in the various racks. Instead, the top compartment is equipped with two heat exchangers, one of which is mounted in the ceiling of the compartment, where it provides good cooling capacity, since the air is hottest at the top of the rack. The aforesaid heat exchangers are primarily installed in compartments or shelves that are intended to carry magazines of known type for boards with printed wiring and various types of components (relays, condensers, resistors etc.). In the case of the invention, however, most of the shelves in the racks (e.g. more than 75%) shall be equipped with a heat exchanger. The heat exchanger can thereby be integrated with the shelf or constitute a separate part in relation to the shelf, whereby in the latter case the heat exchanger is attached to the rack itself. The rack is equipped with two vertically running pipelines 15 and 16. The inlet connections to the heat exchangers' tubular coils are connected to the first pipeline 15, while the outlets from the heat exchangers' tubular coils are connected to the second pipeline 16. The pipelines 15 and 16 in the different racks are connected in parallel to a tubular evaporator or fluid cooler 17 of known type. In the fluid cooler, the tubular evaporator is indicated by 17a and the compressor by 17b. The compressor is connected to a condenser or water recooler located outside of the premises via a line 17c for the refrigerant. In the case shown here, the tubular evaporator has been connected to the same condenser or water recooler as the refrigerator module 7, but can otherwise be connected to a separate condenser or water recooler. The piping system that includes the first and second pipelines 15 and 16 to the various racks is connected to the tubular evaporator's inlet 17a' and outlet 17a". The pipelines can also be connected in series, depending on the diameter of the tubing used. A pump 18 of known type is incorporated on the pressure side of the piping system. The fan's 7d outgoing air flow is kept to a temperture of about +8°C in order to minimize the water content of the supply air to about 6.5 grammes of water per kg of air. This is done in order to dry the air so that water does not condense on the outer surfaces of the heat exchangers. The air entering the air duct from the fan outlet can, owing to the ejector effect, entrain room air and thereby assume a temperature of +15—16°C.

The coolant in the piping system should preferably consist of freon in the liquid state, which is thus cooled in the fluid cooler 17 and circulated in the piping system by the pump 18. In the case illustrated here, the fluid has an inlet temperature at 17a' of up to +30°C, while its temperature at the outlet 17a" can be about +15°C. Circulation in the system is indicated by arrows 19 and 20.

The upward and preferably dry air streams 12a, 12b, 12c and 12d in the different racks thereby pass the heat exchangers' cooling flanges on the shelves equipped with heat exchangers, while absorbing heat from the racks. As shown by arrows 12c and 12f, air eddies are also created in the rack. An example of the velocity of the fluid in the closed piping system is 0.2—0.4 m/s. The piping system and the tubular coils in the heat exchangers consist of copper tubing with an inside diameter of about 10 mm. The velocity of the upward air streams in the racks is 1—5 m/s. The temperature on the premises can be +23—24°C.

Figure 2 shows three compartments in a rack 5. For each shelf, a side piece 21 is shown, which is fitted at the rear of the rack with hook devices (not shown) by means of which the shelf can be attached to the rack posts 22 on either side of the shelf. The aforesaid rack posts are provided with slots into which the aforesaid hook devices can be inserted and secured in known manner. At its inner and outer ends, the shelf bears the bracket that supports the lower parts 23a and 23b of a board magazine that carries the boards standing on end. Figure 2 shows a board, indicated by 24. The boards are mounted in the magazine with intervening spaces, and each board carries components (not shown) and printed wiring on one or both sides. Since the constructions of the shelf, the board magazines and the boards are presumed to be known, they will not be described in any greater detail here. It can however be stated that the shelf consists of, besides the aforesaid side pieces 21, a load-bearing surface which, in the view shown in figure 2, is composed of two flat sections 21a and 21b which meet each other at an angle. For the future, a shelf design is being discussed with a single surface that runs essentially perpencular to the vertical direction of the rack. This surface can be assumed to coincide with the left-hand flat portion 21a of the shelf's load-bearing surface. The invention also applies for other shelf planes, however, for example planes that coincide essentially with the section 21b. The upward air stream in the rack is divided as shown in figure 2 into a number of substreams 12a, 12b and 12c. The rack can be equipped with a section 25, provided with a number of slots for air that flows upward from the lower parts of the rack and via the slots into the rack in question. In the embodiment shown, a slot section 25a, 25b, 25c is provided at each shelf. At the lowest shelf, the air flows in from the cooling duct via the slots into the adjoining section and then upward between the aforesaid boards. Air also enters at the overlying shelves via their slot sections. The air flowing upward in the rack passes the shelves with their heat exchangers 13 and continues up via the lower parts of the board magazines and up between the boards to the overlying parts, and so on. Except for the aforesaid rack posts 22, the rack is completely open in back, and a chimney duct 26 is provided at the rear of the rack, in which some of the air can escape on its passage upward through the rack. The upward and backward sweeping air stream created in this fashion provides good cooling, especially at the rear parts of the rack, which are the most critical from the viewpoint of cooling.

Figure 3 shows the shelf 21 in a horizontal view from above. As is evident from the figure, the flat or sloping portion 21a, 21b contains the cavity for the heat exchanger, which is of the type that incorporates a tubular coil 13a with attached cooling flanges 13b. The heat exchanger has a large extent in the plane shown in figure 3 and occupies a large part of the plane of the shelf in question. As an example of what is meant by "a large part" in this case, it can be mentioned that the heat exchanger shall occupy 60—90% of the plane of the shelf in question. The heat exchanger exhibits a depth A equal to about 2/3 of the depth B of the shelf. The heat exchanger extends from the rear parts of the rack (the beams 22) towards the front of the rack. Owing to its relatively far retracted position in the rack, the heat exchanger provides an effective cooling function at the same time as the shelf retains a reliable load-bearing function. The heat exchanger can be attached to the shelf and/or the rack. The points of attachment have extensive contact surfaces that ensure good heat conduction from the warmer parts of the heat exchanger to the colder surfaces on the shelf and/or the rack. The tubular coil 13a is fitted with an inlet connection 13c and an outlet connection 13d which, in accordance with the above, shall be connected to the first and second pipelines. The aforesaid first and second pipelines can be run in the aforesaid chimney duct 26. Parts of the first pipeline's 15 run in the aforementioned duct 26 have been indicated in figure 2. The tubes in the aforesaid tube coil in the heat exchanger extend primarily in the transverse direction of the shelf along most of the aforesaid transverse direction. In the embodiment shown, the aforesaid tubular coils extend along 90—95% of the shelf's transverse direction. The cooling flanges on the heat exchanger can be square, circular or of another shape.

Figures 5—6 are intended to illustrate an alternative n embodiment of the heat exchanger and its application to the shelf, whose sloping portion 21b' is indicated in figure 4. In this case, the hook devices 27 on the shelf, by means of which the shelf is attachable to the rack posts 22, are also shown. The heat exchanger's cooling flanges are of square shape in this case and are rotated 45° around their longitudinal axis in relation to the embodiment according to figure 4. In addition, the edges of the cooling flanges are upset or bent in order to provide drop collectors, and a drip tray 28 is provided under the heat exchanger. This embodiment may be appropriate when water can be expected to condense on the heat exchanger surfaces during the cooling process. This may occur when the air on the premises is relatively humid, for example because the outdoor humidity is in itself high and corridors and other passageways have stood or are standing open into the premises. Condensate may also form on the surfaces when high cooling capacities are desired from the cooling apparatus. Any moisture on the outside of the cooling flanges can run down along the edges of the cooling flanges to the lowermost corner, under which the drip tray 28 is arranged. The drip tray can lead to one or more containers (not shown).

The heat exchanger is also provided with a protective grille or screen 29 so that piping can be run within the rack regardless of the aforesaid heat exchangers. The grille or screen is thereby designed so that heat is conducted between the heat exchanger and the screen or grille, from which the heat can be conducted to the shelf and/or the rack. The contact surfaces between the grille/screen and the heat exchanger shall thereby be made relatively large in order to facilitate such heat conduction. The heat exchanger is made with an effective cooling surface of up to 2.0 m^2 or more.

The apparatus described above can also be employed to advantage in premises that have an air treatment unit separate from the apparatus, intended to bring about hygienically acceptable air conditions on the premises and, in some cases, to humidify and/or dehumidify the air on the premises.

The shelf cooling function can, however, also function completely separately or essentially separately as per the following. Parts of a rack row at a telecommunications exchange are indicated by 101 in figure 7. The rack construction is of known type and has been supplemented with a cooling apparatus according to the invention. The rack row contains a number of racks 102, 103, 104 etc. and may form single or double racks. In addition to its frame with load-bearing beams 105, 106 and end and front sheets 107, 108, each rack is provided with a number of shelves 109, arranged on top of each other, whose number in the case in question is eight. Telecommunications equipment is arranged on the shelves. The aforesaid telecommunications equipment can thereby consist of circuit boards 111 arranged on end in board magazines 110 in known manner. Most of the shelves in a rack may carry such board magazines, while one or more shelves may carry other types of equipment, such as transformers, power supply components etc.

Most of the shelves in each rack shall preferably be integrated each with its own heat exchanger 112. In the case in question, all shelves in the rack 102 are equipped with such a heat exchanger. Each shelf with appurtenant heat exchanger is thereby integrated in such a manner that the load-bearing capacity of the shelf is not unduly affected. Each heat exchanger is equipped with one or more tubular coils. In the case in question, each heat exchanger has been equipped with one tubular coil, whose inlets and outlets are designated 112a and 112b in figure 7. The aforesaid tubular coils are connected to lines 113 and 114, to which the corresponding inlets and outlets on the tubular coils in the other heat exchangers are also connected in such a manner that a common or continuous piping system is obtained.

The aforesaid piping system thereby comprises vertical first lines 113 and 114, which in turn are connected to horizontal second lines 115 and 116. The vertical lines link the heat exchangers within the same group together, while the horizontal lines connect the heat exchangers in the different racks. A coolant, which consists in the present case of chlorofluorocarbon in liquid form, circulates in the aforesaid tubular coils and lines 113, 114 and 115, 116. Naturally, other coolants may also be used for this purpose. The lines 115 and 116 are also connected to a fluid cooler 117, which is included in the refrigeration equipment employed, together with a compressor 118 and a condenser 119. The cooler 117 with compressor 118 may take the form of a tubular evaporator or a fluid cooler. The condenser (or water recooler) should preferably be located outside of the premises 120 in which the rack or racks in question are set up. The aforesaid refrigeration equipment is of the kind that is available on the general market, and an example of such equipment is Stal-VMR (series 100), which is supplied by Stal Refrigeration AB, Sweden. The refrigeration equipment does not include circulation devices for the coolant (not shown here), and the directions of circulation are indicated by arrows 121 and 122, whereby direction 121 is the outgoing direction from the fluid cooler 117 and 122 is the incoming direction. The flow velocity of the coolant is dependent upon the capacity with which the heat exchangers are to operate in the racks. 0.2—1.5 m/s is a typical velocity of the coolant. The piping system and the tubular coils in the heat exchangers consist of tubing of e.g. copper with an inside diameter of about 10 mm. The aforesaid coils and piping system are arranged in such a manner that the throughput rate in the different heat exchangers is roughly the same.

The aforesaid board magazines, which will be described in greater detail below, are applied to their respective shelves via brackets or cantilever supports 123, whose detailed design is also described below. Special arrangements have been made in order to achieve extensive contact surfaces between different parts of the rack. Thus, there are extensive metallic contact surfaces at each shelf between the rack post 105′ and the outgoing coolant line 113. These metallic contact surfaces are situated on a metal plate or a metal band or similar device which is attached both to the aforesaid post 105′ and the aforesaid line 113. A metal plate or a metal band 125 is also arranged at each shelf between the two lines 113 and 114 for the coolant. The device 125 is equipped with flanges 125a. The device 125 is further arranged so that it makes contact with the back surfaces of the board magazine when the board magazine with boards is placed on the shelf in question. A device 126 corresponding to device 124 is connected between the post 105 and the line 113 at each shelf.

Figure 8 shows two shelves 110′ and 110″ placed on top of each other in a rack according to figure 1, each shelf with its own heat exchanger 112′ and 112″, respectively. Each shelf incorporates two parallel load-bearing parts 127 and 128 (see also figure 9). Between these load-bearing parts, a cavity 129 is provided, as shown in figure 9. The heat exchanger 112′ or 112″ in question is mounted in this cavity. The heat exchanger is thereby mounted in a holding device, which preferably takes the form of a screen or grille 130 extending around the entire circumference of the heat exchanger. Along its long sides, the holding device 130 is provided with fastening pieces 130a and 130b, by means of which the holding device is fastened to the load-bearing parts 127 and 128. Thus, the fastening piece is fastened to the load-bearing part 127 by means of channels 131, which are screwed in place or otherwise fastened by means of screws 132 or the equivalent to the load-bearing part 127 so that an extensive contact surface is obtained between the fastening piece 130a and the load-bearing part 127. The aforesaid load-bearing part preferably consists of an extruded beam of aluminium or other metallic device. The fastening piece 130b is attached to the load-bearing part 128 by means of a long channel, which is attached to the load-bearing part 128 by means of screws 133 or equivalent fasteners, whereby the piece 130b is wedged between the clamping channel and the surface of the load-bearing part. The shelf is provided with bracket-shaped side pieces that incorporate projecting devices 134, 135, which are inserted in corresponding slots on the concerned rack post 105 and are designed in such a manner that, after being inserted in the slot, they lock to the post to provide appropriate support for the shelf with appurtenant heat exchanger, board magazines and boards.

The design of the cantilever support (see also figure 7) is shown in greater detail by figure 8. The cantilever support has a middle section 123a that reaches across the aforesaid cavity between the load-bearing parts 127 and 128 of the shelf. The cantilever support is anchored at its ends 123b and 123c to the load-bearing parts 127 and 128. The contact surfaces between the aforesaid free ends 123b and 123c on the one hand and the load-bearing parts 127 and 128 on the other hand are also extensive. The board magazines 110 are placed on the aforesaid cantilever supports 123a, whose top surfaces 123c′ have been made extensive in the horizontal plane. The board magazines have a back piece, to which box-shaped or cassette-shaped pieces are applied, in which the circuit boards are placed on end. Receptacles in which the boards can be connected via matching plugs are provided on the front side of the back piece. The box-shaped part on the board magazine is connected via its back surface 110a to the device 125, which thereby extends along a large part of the height of the board magazine.

The aforesaid back piece and board magazine boxes are made with slits and/or cutouts so that free air circulation is obtained between the circuit boards placed on end in the board magazines with their components. The convection that is brought about owing to, inter alia, the cold surfaces of the heat exchangers and the hot components gives rise to air currents past the components on the boards and removal of the heat generated in the components by the air currents. Such air currents between the boards in the board magazines are indicated in figure 8 by first arrows 136, which symbolize the upward air currents. Second arrows 137 symbolize downward air currents, while third arrows 138 symbolize circulating air currents between the boards. In addition, heat dissipation is obtained by direct radiation from the hot board surfaces onto the heat exchangers 112′ and 112″. The aforesaid extensive metallic contact surfaces contribute appreciably to heat transport from the boards/magazines to the colder surfaces on the heat exchangers.

The heat exchangers in a rack are thereby designed with such large effective cooling surfaces that cooling capacities in excess of 150 Watts are obtained from the heat exchangers. It can hereby be mentioned that cooling capacities of 500 Watts or higher from each heat exchanger are within the realm of possibility. Thanks to the specific integration between each shelf and its heat exchanger, a large effective cooling surface can be obtained in the shelf. 1.5—4 m^2 can be mentioned as an example of the size of an effective cooling surface.

The integrated cooling function with heat conduction via the contact surface, radiation and natural convection can be provided with the proposed principle in such a manner that, despite a high packing density of the boards and their components, the maximum component temperature on each board can be limited to about 75°C. In one proposed embodiment, however, it is proposed that the heat exchangers be arranged and operated in such a manner that maximum component temperatures of about 65°C or lower are obtained.

Figure 9 shows, inter alia, the connection of the devices 124 and 125 to the coolant lines 113 and 114. The devices 124 are connected via extensive contact surfaces 124a to the rack posts 105 and 105′. The side pieces on the shelf are indicated by 139 and 140 and the cavity between the load-bearing parts 127 and 128 is indicated by 141. The rack has a chimney-like duct 142 at the back, which contains the aforesaid lines 113 and 114 and the connections 112a, 112b on the heat exchangers (see also fig. 10). The figure is simplified in that the link between the connections 112, 112a, 112b and the vertical pipes 113 and 114 are symbolized by thin solid lines 143 and 144. The board magazines are shown schematically, since they are assumed to be of known type.

Figure 10 shows the extent of the heat exchanger in the horizontal plane of the shelf and its mounting in the cavity 141. The heat exchanger occupies a large part of the shelf's horizontal plane, for example 60—90%. The heat exchanger runs underneath the board magazines and extends over the greater part of their depth. Thus, the heat exchanger has a depth which is preferably 70—95% of the depth of the board magazines. In this manner, the heat exchanger can be made to act indirectly over the entire depth of the board. The temperature of the coolant entering the fluid cooler 117 can be up to about +30°C. The temperature of the outgoing coolant from the cooler can be down to +15°C.

The invention is not limited to the versions described above as examples, but can be subject to modifications within the framework of the following patent claims and invention concept. Values of temperature, cooling surface etc. can, for example, vary from one embodiment to another.

CLAIMS

1. Apparatus for cooling telecommunications equipment consisting of component-carrying printed circuit boards arranged in board magazines which in turn are mounted on shelves arranged on top of one another in a rack characterized in that one or more shelves are integrated each with its own heat exchanger which incorporate cooling flanges and at least one tubular coil which is connected via the heat exchanger to a cooler for a coolant running in the tubular coil; that each heat exchanger, viewed in the horizontal plane of its shelf, extends along the greater part of the shelf; that extensive heat-conducting metallic contact surfaces are arranged between each board magazine and the associated shelf as well as between the shelf and the heat exchanger; and that the aforesaid heat exchangers and cooler are arranged in such a manner that they contribute towards removing the heat given off by the telecommunications equipment by means of heat conduction via the aforesaid extensive contact surfaces, even at a high packing density of the aforesaid telecommunications equipment in the rack and a high rate of heat emission.

2. Apparatus according to claim 1, characterized in that half, and preferably the greater portion, of the rack's shelves are integrated each with its own heat exchanger, and that the heat exchanger's tubular coils are interconnected and connected to the cooler.

3. Apparatus according to Claim 2, in which the cooler is common to all heat exchangers.

4. Apparatus according to any of claims 1 to 3 characterized in that the heat exchangers and the cooler are arranged to ensure that the component temperatures for each card will not exceed about 75°C, preferably 65°C or lower.

5. Apparatus according to any of claims 1 to 4 characterized in that each heat exchanger has an effective cooling surface of up to 1.5 m^2, preferably between 1.5 and 4 m^2.

6. Apparatus according to one of the preceding claims, characterized in that each heat exchanger has a cooling capacity in excess of 150 Watts, preferably 500 Watts or higher.

7. Apparatus according to one of the preceding claims, characterized in that the rack(s), with its shelf-integrated heat exchangers. functions as a cooling buffer on the premises where the rack(s) is set up, while simultaneously bringing about cooling of the telecommunications equipment in the rack(s).

8. Apparatus according to one of the preceding claims, characterized in that it incorporates a cooling system, preferably open, arranged on the premises which in turn incorporates a compressed air source whose outgoing air flow is conducted in one or more air ducts underneath the rack(s) so that an upward air stream is created within each rack, where it passes different shelves arranged on top of each other with telecommunications equipment and from which it emerges at the top part of the rack into the atmosphere on the premises; that a heat exchanger with appurtenant cooling flanges and coolant-bearing tubular coil is mounted in each of at least most, e.g. 60—100%, of the rack's or racks' shelves that each heat exchanger extends over the greater part of the plane of the shelf that is perpendicular to the vertical direction of the rack; and that the aforesaid upward air stream in each rack passes the cooling flanges on the heat exchangers in the shelves provided with heat exchangers.

9. Apparatus according to one of the preceding claims, characterized in that each shelf is provided with a cavity extending in the horizontal plane, in which the heat exchanger belonging to that shelf is mounted by means of a holding device preferably consisting of a screen, which is connected to the body of the shelf via at least its two long sides.

10. Apparatus according to claim 9, characterized in that the shelf incorporates two essentially parallel load-bearing parts between which the aforesaid cavity extends; that cantilever supports are arranged to the load-bearing parts so that their middle sections extend across the cavity; that the cantilever supports carry on their middle sections one or more board magazines; that the extensive metallic contact surfaces include attachment surfaces between the holding device and the body of the shelf, between the cantilever supports and the body of the shelf and between the aforesaid middle sections and board magazines, providing for effective heat conduction between the interiors of the board magazines where the boards are placed on end and the heat exchanger.

11. Apparatus according to one of the preceding claims, characterized in that metallic heat-conducting devices are connected between all or different units of the heat exchangers, the coolant lines, the board magazines, the shelves, the racks and/or the holding devices.

12. Apparatus according to one of the preceding claims, characterized in that the coolant circulated in the cooling sdystem, preferably consisting of chlorofluorocarbon in liquid form, has a temperature entering the cooler of up to about +30°C and a temperature leaving the cooler of down to about +15°C.

13. Apparatus according to one of claims 8—12 characterized in that the aforesaid compressed air source or fan for generation of the aforesaid upward air stream in the rack(s) and the air circulation on the premises is incorporated in a refrigeration module of known type that preferably operates by direct expansion, which refrigeration module incorporates a cooling coil compressor(s) a fan and preferably a filter in addition to which a condenser or water recooler belonging to the regrigeration module is located outside the premises.

14. Apparatus according to claim 13, characterized in that the refrigeration module is arranged to create forced upward air streams in each rack, bringing about a cooling capacity of 150 Watts or higher from the heat exchanger on each shelf.

15. Apparatus according to one of the preceding claims, characterized in that the shelves on the racks are arranged to carry, in addition to various telecommunications units the aforesaid magazines with boards with printed wiring and telecommunications components; that the upward air streams pass upwards between the boards and are directed out into a chimney duct via the rear of the rack; and that the heat exchangers extend from the rear of the rack in the direction towards the front of the rack in such a manner than they occupy preferably about 2/3 of the shelf's depth, providing favourable cooling of and at the rear portions of the rack as well.

16. Apparatus for cooling telecommunications equipment substantially as hereinbefore described with reference to and as shown in the accompanying drawings.

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(54) Schaltschrank

(57) Die Erfindung betrifft einen Schaltschrank mit einem Schrankkorpus, dessen offene Korpusseite zumindest teilweise mit einer oder mehreren Seitenwänden verschließbar ist, wobei im Bereich wenigstens einer Seitenwand eine Klimatisierungseinrichtung angeordnet ist, über die Wärme aus dem Innenraum des Schrankkorpus abgeführt oder diesem zuleitbar ist. Zur Vereinfachung des Aufbaus der Klimatisierungseinrichtung ist es erfindungsgemäß vorgesehen, daß parallel beabstandet zu der Seitenwand dem Innenraum des Schrankkorpus zugekehrt eine Zwischenwand angeordnet ist, und daß in dem Zwischenraum, der von der Seitenwand und der Zwischenwand umschlossen ist, die Klimatisierungseinrichtung angeordnet ist.

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(54) **Cooling system for communication station**

(57) A casing (103) of a communication station, accommodating communication equipment including heat components, is cooled by a boiling-type cooler (121) in a natural circulation refrigerating circuit and an evaporator (113) in a forced circulation refrigerating circuit (109), activated by a compressor (110). A common airflow path (130) has a heated air intake port (132) for taking heated air into the casing (103) and a cooled air exhaust port (133) for blowing cooled air into the casing (103). A common fan (131) sends air to the boiling-type cooler (121) and the evaporator (113), all these items being built in the common airflow path (130). A temperature detector (134) detects at least an outer air temperature, and a compressor control (140) stops the operation of the compressor (110) in the forced circulation refrigerating circuit (109) based on the detected temperature.

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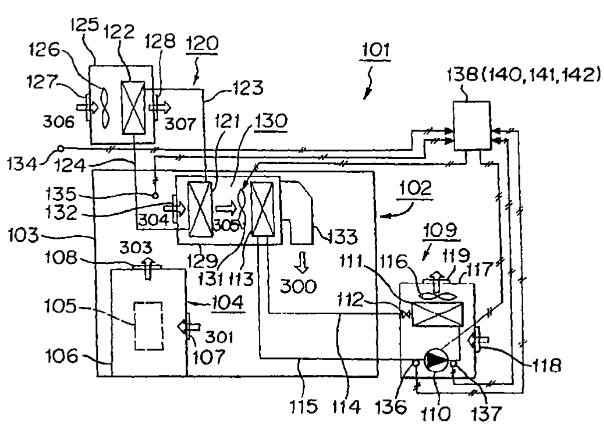

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Description

FIELD OF THE INVENTION

[0001] The present invention relates to a cooling system to cool a communication station, wherein an inside of the communication station, accommodating communication equipment including heat elements such as boards, are cooled by a cooling device such as an air conditioner.

[0002] The present invention also relates to an improvement of a system for cooling a casing of a communication station, wherein an inside of the casing, accommodating communication equipment including heat components, is cooled by a boiling type cooler in a natural circulation refrigerating circuit and an evaporator in a forced circulation refrigerating circuit, circulated by a compressor.

DISCUSSION OF BACKGROUND

[0003] In recent years, communication stations including a large number of electronical boards for communication are located in various places for relaying communications along with expanding popularization of portable communication apparatuses. Dimensions of such communication relay stations are, for example, a width of about 6 m, a depth of about 1.7 m, and a height of about 1.7 m. Although the communication relay stations are relatively small, a gloss colorific value of electronical boards, equipped in the communication stations, are several kW through several dozens of kW. Therefore, air conditioners are used to cool these electronic boards by cooling board casings of the communication stations. Figure 33 illustrates a structure of a conventional method for controlling to cool a communication station. In Figure 33, numerical reference 1 designates a rack accommodating a communication equipments including a large number of electronic boards and so on; numerical reference 3 designates a fan; numerical reference 4 designates an indoor unit including an indoor heat exchanger 4a and an indoor fan 4b; numerical reference 5 designates an outdoor unit including a compressor 5a and an outdoor heat exchanger 5b; numerical reference 6 designates a suction air into the indoor heat exchanger 4a; numerical reference 7 designates a blown-out air from the indoor heat exchanger 4a; numerical reference 8 designates a suction air for cooling the communication equipment 2; numerical reference 9 designates a suction air temperature detector for detecting a temperature of the suction air 6; numerical reference 10 designates a casing for accommodating the lack 1 and the indoor unit 4; and numerical reference 11 designates a cooling controller for controlling a cooling capability of the compressor 5a.

[0004] In the next, an operation of the conventional method for controlling to cool the communication station will be described. The number of operating communication equipments 2 is changed in response to a frequency of communication, and a colorific value is increased or decreased in response to the number of operating communication equipments 2. The suction air 8 to the communication equipments 2 is sent by the fan 3 to cool the communication equipments 2, is heated after cooling, and is taken in the indoor unit 4 as the suction air 6 into the indoor heat exchanger 4a. The suction air 6, taken into the indoor unit 4, is cooled by the indoor heat exchanger 4a, is blown out into the casing 10 as the blown-out air 7 from the indoor heat exchanger 4a, and is served as the suction air 8 into the communication equipments 2. On the other hand, the cooling controller 11 controls a cooling capability of the compressor 5a based on an output temperature of the suction air temperature detector 9 so that the suction air 8 into the communication equipments 2 becomes a predetermined temperature, for example, 35°C or less.

[0005] Further, a large number of communication stations for handy personal phones and so on are located in cities, rooftops of condominiums and office buildings, mountain tops in the suburbs, and wilds. Communication equipments are generally accommodated in a sealed casing in the communication stations. However, some of casings have a space too narrow to receive a person. Therefore, the casings are adequately cooled because heat components are included in the communication equipments.

[0006] As a system for cooling such a casing is disclosed in Japanese Unexamined Patent Publication JP-A-11-135972. Figure 34 illustrates this system. The casing cooling system 151 for a communication station 152 is composed of a boiling type cooler 121 in a natural circulation refrigerating circuit 120 and an evaporator 113 in a forced circulation refrigerating circuit 109 so as to cool an inside of the casing 103 as the sealed space. The forced circulation refrigerating circuit 109 is constructed to forcibly circulate a refrigerant by a compressor 110, which mechanism is generally used in an air conditioner and so on. Communication equipments 104 including heat components 105 are accommodated in the casing 103. In generally used communication equipments 104, a fan (not shown) is located inside an equipment case 106 having built-in heat components 105 to take an air from an intake port 107, located on a side surface or a bottom surface of the equipment case, and to blow a heat out of an exhaust port 108, positioned on a back of the equipment case.

[0007] In a case of an evaporator, an intake port 155 for taking an air inside the casing 103 and an exhaust port for blowing a cooled air into the casing 103 are formed. In the case 153 of the evaporator, the evaporator 113 and a fan 154 are built. On the other hand, on a back surface of the equipment case 106, a heated air guide path 157, connected to the exhaust port 108, is formed. The heated air guide path 157 is connected to an air path 167 having a heated air intake port and a heated air exhaust port. A condenser 122 and a fan 163 are built in the air path 167.

[0008] A condenser 111 in the force circulation refrigerating circuit 109 is located in a case of a condenser as an outdoor unit of an air conditioner. The case 117 of the condenser is formed like a box having an outer air intake port 118 and an exhaust port 119. The condenser 111, the compressor 110, a choke valve 112 for refrigerant, and a fan 116 are housed in the case 117 of the condenser. The forced circulation refrigerating circuit 109 is constructed by sequentially connecting the compressor 110, the condenser 111, the refrigerant choke valve 112 in the condenser case 117 with the evaporator 113 in the casing 103 via tubes 114, 115 for the refrigerant so as to be shaped like a ring. Further, the condenser 122 in the natural circulation refrigerating circuit 120 is disposed in the condenser case 159 as an outdoor unit. The condenser case 159 is shaped like a box having an outer air intake port 160, an exhaust port 161, the condenser 122, the fan 162. The natural circulation refrigerating circuit 120 is constructed by connecting the condenser 122 in the condenser case 159 with the boiling type cooler 121 in the airflow path 167 via a refrigerant evaporation tube 123 and a liquid refrigerant return tube 124 so as to be shaped like a ring.

[0009] In the conventional cooling system, a cooling capability is determined in conformity with a maximum load of the heat components 105. Because the casing 103 generally has a structure having an extremely small heat transfer through solid conductors, there are very small variations of a cooling load inside the casing 103 in response to variations of an outer air temperature.

[0010] In the next, an operation of the conventional system will be described. An air in the casing 103 is taken in the equipment case 106 through the air intake port 107 when a fan in the communication equipments 104 (not shown) is driven. A cooling air, taken in, cools the heat components 105 and is changed to be a heated air. Thereafter, the heated air is blown out of the exhaust port 108 in the back surface of the case into the heated air guide path 157. Thus blown-out heated air is sucked in the airflow path 167 through the heated air intake port 158 by the fan 163. The heated air passes through the boiling type cooler 121 in the airflow path 167 and primarily cooled by changing heat with a refrigerant in the natural circulation refrigerating circuit 120. The air subjected to the primary cooling is sucked by the fan 163 and blown into the casing 103 through the exhaust port 164. At least a part of the air subjected to the primary cooling is sucked into the evaporator case 153 through the intake port 155 by the fan 154 and passes through the evaporator 113, whereby the air is cooled by changing heat with a refrigerant in the forced circulation refrigerating circuit 109. Thus cooled air is blown out of the cooling air exhaust port 156 into the casing.

[0011] In the natural circulation refrigerating circuit 120, a refrigerant in the boiling type cooler 121 is boiled by changing heat with the heated air so as to be a gas refrigerant. The gas refrigerant passes through the refrigerant evaporation tube 123 and reaches the condenser 122. The gas refrigerant in the condenser 122 is changed to a liquid refrigerant by changing heat with an outer air passing from the outer air intake port 160 to the exhaust port 161 in a condenser case 159, wherein the gas refrigerant is cooled. The liquid refrigerant returns to the boiling type cooler 121 through the liquid refrigerant return tube 124 by a gravity flow caused by a difference of weight densities between the liquid refrigerant and the gas refrigerant. On the other hand, in the forced circulation refrigerating circuit 109, a high-temperature high-pressure gas refrigerant, forcibly discharged out of the compressor 110, flows into the condenser 111 and is changed to be a liquid refrigerant by exchanging heat with an outer air flowing from the outer air intake port 118 to the exhaust port 119 in the condenser case 117 by the fan 116, wherein the high-temperature high-pressure gas refrigerant is cooled. The liquid refrigerant is depressurized by the refrigerant choke valve 12 to be a gas-liquid two-phase state. Thereafter, the liquid refrigerant reaches the evaporator 113 through the refrigerant tube 114. The refrigerant exchanges heat with an air flowing through the evaporator case 153 in the evaporator 113 so as to be a low-pressure gas refrigerant. The refrigerant returns to an intake side of the compressor 110 through the refrigerant tube 115.

[0012] In the conventional cooling control method for communication stations, because an ordinary wall-hang or dangling-type package air conditioner is used as disclosed in Japanese Unexamined Patent Publication JP-A-4-98038, a suction temperature 6 of the indoor heat exchanger 4a is detected by the suction temperature detecting means 9. However, in case that an air distribution in the casing is not preferable, there occur phenomenons that exhausted heat from the communication equipments resides and an air blown out of the indoor unit causes a short cycle. Accordingly, a calorific value of the communication equipments, i.e. a real cooling load, does not in conformity with the suction air temperature 6. Accordingly, the air conditioner does not deal with the real cooling load, whereby a temperature in the casing is increased or decreased; and environmental conditions of working temperature of the communication equipments are resultantly unsatisfied, and vapor is condensed in the air conditioner.

[0013] Meanwhile, in the conventional cooling system, because the boiling type cooler 121 and the evaporator 113 are located in the different airflow paths, it is necessary to locate fans 163 and 154 respectively for the airflow paths.

[0014] Further, because a density of components installed in the casing 103 is high in order to pursue compactness, it is impossible to provide a space for additional fans. Therefore, there is a problem that a large-sized fan can not be used, for example, a large airflow rate can not be supplied when the space of the casing 103 is unchanged.

[0015] Incidentally, because the air, primarily cooled in the boiling type cooler 121, diffuses in the casing 103 after passing through the exhaust port 164, a part of the airflows toward the evaporator case 153, like an arrow C, and the other parts are sucked in the air intake port 107 of the communication equipment 104 by bypassing like an arrow B. When an airflow rate of the fan 154 is excessively large, the cooled air, blown out of the cooling air exhaust port 156, may return to the intake port 155 by a short cycle, whereby cooling efficiency is deteriorated.

[0016] Further, in order to take the heated air in the boiling type cooler 121, it is necessary to locate the heated air guide path 157 and the heated air intake port 158, whereby a structure of airflow path becomes complicated. If the heated air guide path 157 is not located, a high-temperature heated air, blown out of the exhaust port 108 of the communication equipments 104, is directly sucked into the suction port 155 of the evaporator 113 by bypassing the boiling type cooler 121, whereby there is a danger that the forced circulation refrigerating circuit 109 is broken.

SUMMARY OF THE INVENTION

[0017] It is an object of the present invention to solve the above-mentioned problems inherent in the conventional technique and to provide a cooling control method for communication stations, by which communication equipments can be cooled in response to a variation of a calorific value, caused by the number of operations of the communication equipments. Further, it is possible to control with a good follow-up capability; a cooler can be highly efficiently operated by saving an energy; moisture condensation can be prevented; frequent turning-ons and turning-offs of an air conditioner can be prevented; it is possible to deal with environmental changes; and a COP of the cooler can also be increased; and other improvements can be obtained.

[0018] Further, another object of the present invention is to provide a cooling control method for communication stations, by which moisture condensation, caused by excessive drop of a temperature of a blowing-out air from an indoor unit can be prevented.

[0019] Another object of the present invention is to provide a cooling system for a casing of communication stations, by which a capacity of the total volume of the cooling system is optimized, an energy is saved, and reliability of the system can be improved by appropriately combining a boiling-type cooler in a natural circulation refrigerating circuit, an evaporator in a forced circulation refrigerating circuit, and fans.

[0020] According to a first aspect of the present invention, there is provided a cooling system for a communication station for cooling a casing of the communication station, accommodating communication equipments including heat components, by a boiling-type cooler in a natural circulation refrigerating circuit and an evaporator in a forced circulation refrigerating circuit, activated by a compressor comprising:

a common airflow path having a heated air intake port for taking a heated air into the casing and a cooled air exhaust port for blowing a cooled air into the casing; and
a common fan for sending an air to the boiling-type cooler and the evaporator,

wherein the boiling-type cooler, the evaporator, and the common fan are built in the common airflow path.

[0021] According to a second aspect of the present invention, there is provided the cooling system for the communication station,

wherein the common airflow path is constructed by an air path on a cooler side, inside which the boiling-type cooler is installed, an air path on an evaporator side, in which the evaporator is installed, and a connection air path for connecting the air path on the cooler side and the air path on the evaporator side.

[0022] According to a third aspect of the present invention, there is provided the cooling system for the communication station further comprising:

a temperature detecting means for detecting at least an outer air temperature; and
a compressor control means for stopping an operation of the compressor in the forced circulation refrigerating circuit based on a detected temperature received from the temperature detecting means.

[0023] According to a fourth aspect of the present invention, there is provided the cooling system for the communication station further comprising:

a malfunction detecting means for detecting malfunctions of the forced circulation refrigerating circuit; and
a driving means for keeping the common fan in a running state when the malfunctions in the forced circulation refrigerating circuit are detected by the malfunction detecting means.

BRIEF DESCRIPTION OF THE DRAWINGS

[0024] A more complete appreciation of the invention and many of the attendant advantages thereof will be readily obtained as the same becomes better understood by reference to the following detailed description when considered in connection with the accompanying drawings, wherein:

Figure 1 illustrates a structure of a cooling control method for a communication station according to Embodiments 1 and 6;

Figure 2 is a block chart illustrating a cooling control means according to Embodiments 1 and 6;

Figure 3 is a flowchart showing a control by the cooling control means according to Embodiments 1 and 6;

Figure 4 is a flowchart illustrating a control by another cooling control means according to Embodiments 1 and 6 of the present invention;

Figure 5 is a flowchart illustrating a control by another cooling control means according to Embodiment 1;

Figure 6 illustrates a structure of a cooling control method for communication stations according to Embodiments 2 and 7;

Figure 7 is a block chart illustrating a cooling control means according to Embodiment 2;

Figure 8 is a flowchart illustrating a control by the cooling control means according to Embodiment 2;

Figure 9 is a flowchart illustrating a control by another cooling control means according to Embodiment 2;

Figure 10 is a block chart illustrating another cooling control means according to Embodiment 2;

Figure 11 is a flowchart illustrating a control by another cooling control means according to Embodiment 2;

Figure 12 is a flowchart illustrating a control by another cooling control means according to Embodiment 2;

Figure 13 illustrates a structure of a cooling control method for a communication station according to Embodiment 3 of the present invention;

Figure 14 is a block chart illustrating a cooling control means according to Embodiment 3;

Figure 15 is a flowchart illustrating a control by the cooling control means according to Embodiment 3;

Figure 16 illustrates a relationship between a suction temperature and a cooling capability of an air conditioner;

Figure 17 illustrates a relationship of various temperatures of the air conditioner according to Embodiment 3;

Figure 18 is a flowchart illustrating a control by another cooling control means according to Embodiment 3;

Figure 19 illustrates a state of thermo-off according to Embodiment 3;

Figure 20 is a block chart illustrating another cooling control means according to Embodiment 3;

Figure 21 is a flowchart illustrating a control by another cooling control means according to Embodiment 3;

Figure 22 illustrates a structure of a cooling control method for a communication station according to Embodiment 4;

Figure 23 illustrates a structure of another cooling control method for a communication station according to Embodiment 4;

Figure 24 illustrates a structure of a cooling control method for a communication station according to Embodiment 5;

Figure 25 is a block chart illustration a cooling control means according to Embodiment 7;

Figure 26 is a flowchart illustrating an operation of the cooling control means according to Embodiment 7;

Figure 27 is a flowchart illustrating an operation of a cooling control method according to Embodiment 6;

Figure 28 illustrates a structure of a cooling control method for communication station according to Embodiment 8;

Figure 29 is a block chart illustrating a cooling control means according to Embodiment 8;

Figure 30 schematically illustrates a structure of a cooling system for a casing of a communication station according to Embodiments 9, 11 and 12, in accordance with the invention;

Figure 31 schematically illustrates a structure of a cooling system for a casing of a communication station according to Embodiment 10, in accordance with the invention;

Figure 32 is a graph showing a relationship among a capability of a natural circulation refrigerating circuit, an outer temperature, and so on in a cooling system for a casing of a communication station according to Embodiment 11;

Figure 33 illustrates a structure of a conventional cooling control method in a communication station; and

Figure 34 schematically illustrates a structure of a conventional cooling system for a casing of a communication station.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

[0025] A detailed explanation will be given of preferred embodiments in reference to Figures 1 through 32 as follows, wherein the same numerical references are used for the same or similar portions and description of these portions is omitted.

EMBODIMENT 1

[0026] An example of a cooling control method for a communication station and a communication relay station (hereinbelow referred to as communication station) according to Embodiment 1 will be described. Figure 1 illustrates a structure of a cooling control method for a communication station according to Embodiment 1. In Figure 1, numerical references same as those in Figure 33 designate the same or similar portions and description of these portions are omitted. Numerical reference 11a designates a cooling control means for controlling a cooling capability of a compressor 5a depending on cooling conditions of communication equipments 2. Numerical reference 12 designates an electrical power detecting means, such as an electric power meter, for detecting an electric power consumption, consumed by operating communication equipments 2. Numerical reference 13 designates a suction temperature detecting means for the communication equipments, which means detects a temperature of a suction air 8 into the communication equipments. Figure 2 is a block chart illustrating the cooling control means according to Embodiment 1 of the present invention. In Figure 2, numerical reference 20 designates a means for setting a target value of the suction temperature into the communication equipments. Numerical reference 21a designates a means for controlling air conditions, which means controls a cooling capability of the air conditioner having an indoor unit 4, an outdoor unit 5, and so on. Numerical reference 22 designates a means for controlling a frequency, which means controls the frequency of a power source of a compressor motor. The cooling control means 11a is constructed by the target suction temperature value setting means 20 for the communication equipments, the air condition control means 21a, and the frequency control means 22.

[0027] In the next, an operation of the cooling control method for the communication station according to Embodiment 1 will be described in reference of Figures 1 and 2. By the cooling control method for the communication station, a suction air temperature in the communication equipments 2 is controlled to be within a predetermined temperature by supplying a requisite amount of the suction air for the communication equipments by a fan 4b toward the communication equipments 2. In general, a temperature of the suction air 8 for the communication equipments is controlled to be 35°C or less. The suction air 8 is heated after cooling the communication equipments 2, is sucked into the indoor unit 4 so as to be cooled by an indoor heat exchanger 4a, is returned to a casing 10 as a blowing-out air 7, and cools the communication equipments 2 again as the suction air 8 for the communication equipments.

[0028] The cooling control means 11a controls the temperature of the suction air 8 for the communication equipments so as to be a temperature set by the target suction temperature value setting means 20 for the communication equipments, for example, 20°C or less based on outputs from the target suction temperature value setting means for communication equipment 13 and electric power detecting means 12. Provided that a flow rate by the fan 3 is 40 m^3/min, and an electric power detected by the electric power detecting means 12 is 12 kW, because most of the electric power consumption used by an electronic circuit board is occupied by the electric power consumption used by the communication equipments 2, and therefore the consumption electric power of the communication equipments 2 and a calorific value of the communication equipments 2 are substantially the same, a temperature difference between the suction air 6 for the indoor heat exchanger and the suction air 8 for the communication equipments is expressed by the following equation:

$$\Delta T = (\text{electric power consumption})/(\text{airflow rate} \times \text{air density} \times \text{specific heat of air at constant pressure}).$$

[0029] When the electric power consumption is 12 kW; the airflow rate is 0.67 m^3/sec; the air density is 1.2 kg/m^3; and the specific heat at constant pressure of the air is 1.01 kJ/kg·K, ΔT=15deg. When the temperature of the suction air 8 for the communication equipments is 20°C, the temperature of the suction air 6 for the indoor heat exchanger is 20°C + ΔT=20°C + 15deg=35°C. Provided that the airflow rate of the fan 4b is 40 m^3/min, in order to supply the suction air 6 for the indoor heat exchanger as the suction air 8 for the communication equipments after cooling to be 20°C, an electric power of 12 kW is necessary. By controlling a capability of the air conditioner upon a detection of the electric power consumption by the communication equipments, it is expected that the capability follows an actual heat load. However, the expected temperature is not attained because the blown-out air 7 is mixed with an atmosphere around the blown-out air 7 from the indoor unit in the casing, and the mixture becomes the suction air 8 for the communication equipments. Further, when the calorific value of the communication equipments 2 is abruptly changed, a very short time difference occurs until an influence of the change affects the output from the suction temperature detecting means 13 for the communication equipments. In order to correct the time difference, a requisite capability of the compressor is basically calculated by comparing a temperature, outputted from the suction temperature detecting means 13 for the communication equipments, with the set target value so that the suction air temperature 8 for the communication equipments becomes the set target value, obtained by the target suction temperature value setting means 20 for the communication equipments. Thereafter, an upper limit of maximum frequency of the compressor is calculated based on the detected electric power from the electric power detecting means 12, and the compressor 5a is controlled through the frequency control means 22 upon a command from the air condition control means 21a, which outputs the command about the frequency after correcting the above-mentioned requisite capability.

[0030] Figure 3 is a flowchart illustrating a control operation by the cooling control means 11a. The cooling control means 21a checks a current frequency f, currently outputting to the compressor 5a, in step S1, hereinbelow referred to as S1. In S2, a temperature Tm of the suction air 8 of the communication equipments, detected by the suction temperature detecting means 13 for the communication equipments, and a set temperature Ts of the suction air 8 for the communication equipments, set by the target suction temperature value setting means 20 for the communication equipments are checked. In S3, when the temperature Tm and the set temperature Ts are not equal, it is checked whether or not the temperature Tm exceeds the set temperature Ts. In S5, when the temperature Tm exceeds the set temperature Ts, the frequency of the power source for the compressor 5a is increased by a predetermined value through the frequency control means 22. In step S6, when the temperature Tm does not exceed the set temperature Ts, the frequency of the power source for the compressor 5a is decreased by a predetermined value through the frequency control means 22. In S4, when the temperature Tm is equal to the set temperature Ts in S2, the frequency is maintained without change. The air condition control means 21a receives an output from the electric power detecting means 12 and operates an upper limit fmax of the frequency of the compressor in S7. The upper limit fmax is obtained by a function f(w) having a variable of the output w from the electric power detecting means 12. The function is, for example, as follows:

$$f(w) = 13.7\,(w-6) + 30$$

[0031] This function is for a case that characteristics of the compressor 5a are 12 kW at 112 Hz and 6 kW at 30 Hz, and a capability of the compressor is changed so as to be linear in frequencies between 112 Hz and 30 Hz. In other words, a range of frequency is sufficient for making the compressor demonstrate cooling a capability for cooling the consumption electric power, i.e. calorific value, of the communication equipments 2. In S8, fmax, operated based on the function f(w), and f1, calculated as above, are compared. In S9, S10, and S11, when f1 is larger then fmax, the frequency of the compressor is set to be fmax; and when f1 is fmax or less, the frequency of the compressor is set to be f1. As described, the frequency of the compressor is controlled.

[0032] Needless to say that, in S2 and S3 in Figure 3, it may be determined whether the temperature Tm of the suction air 8 for the communication equipments is equal to or larger than a value obtained by adding or subtracting a predetermined value from the set temperature Ts of the suction air 8 for the communication equipments. In other words, by giving a predetermined range to the set temperature Ts and determining whether or not the temperature Tm is in this range, higher than an upper limit of the range, lower than a lower limit of the range, or in other positions, the temperature Tm may be brought into the range of the set temperature Ts.

[0033] In Embodiment 1, it is possible to provide the cooling control method, which makes the temperature of the suction air for the communication equipments stable, and by which a load and a change of the calorific value in accordance with the number of operations of the communication equipments 2 are dealt with. This control is realized by detecting the temperature of the suction air 8 for operating the communication equipments 2 by the suction temperature detecting means 13 for the communication equipments and controlling the air conditioner by the cooling control means to bring the temperature of the suction air 8 within the range of the set temperature Ts. Further, there is a case that the calorific value of the communication equipments 2 is abruptly changed. In this case, the control depending on only the output signal from the suction temperature detecting means 13, obtained as a result of the change, is insufficient. Therefore, it is possible to achieve a stable control with a good follow-up capability by previously obtaining the output signal from the electric power consumption detecting means 12, being a factor for changing the temperature Tm of the suction air 8, and controlling the frequency of the compressor.

[0034] Although, in Figure 3, the upper limit fmax of the frequency of the compressor is set in S7, the following steps may be adopted. At first it is judged whether the electric power consumption is increased or decreased by comparing detected values of the electric power before and after a receipt from the electric power detecting means 12 in S4, S5, S6, and S7. When the electric power consumption is decreased, a process similar to that described in Figure 3 is proceeded. When the electric power consumption is increased. In S7, the upper limit fmax is changed to the lower limit fmin=f(w) of the frequency of the compressor, and the lower limit fmin of the frequency of the compressor is changed to f(w) shown in Figure 3. In S8, it is judged whether or not f1<fmin. In S9, when f1<fmin, f1=fmin is established. In S10, when f1≧fmin, f1=f1 is established. Thereafter, S11 is selected.

[0035] In such a case, when the calorific value of the communication equipments 2 is abruptly increased or decreased, it is possible to rapidly deal with the increment and the decrement, and a stable control with a good follow-up capability is obtainable.

[0036] Meanwhile, the control method, illustrated in Figure 3, may be modified and simplified using only S4, S5, and S6 through S11 and removing S7, S8, S9, and S10.

[0037] Figure 4 is a flowchart illustrating another example of the cooling control method according to Embodiment 1.

[0038] A procedure until S9 and S10 is the same as above.

[0039] A structure of the cooling method is illustrated in Figure 1, and a block chart of the cooling method is illustrated in Figure 2. In Figure 4, after calculating the frequency f1 in step S9 or S10, an electric power consumption W, i.e. an output from the electric power detecting means 12, at a time of calculating the frequency f1 is compared with the set value Ws of a previously set consumption electric power in S21. In S22, when W>Ws, the frequency of the compressor 5a is set to be f1. In S23, when W≦Ws, the capability of the compressor is minimized. This means that the frequency in the compressor is rendered to be smallest frequency, in which the compressor can be driven. Or it may be possible to form a refrigerating circuit for bypassing a part of a refrigerant, flowing into the outdoor heat exchanger 5b on a suction side of the compressor, to further degrade the cooling capability by making the frequency a minimum frequency enabling the operation of the compressor. The bypass circuit is not illustrated in Figure 1.

[0040] The set value Ws is made a little larger than a minimum capability of the air conditioner. For example, when the minimum capability is 6 kW, the set value Ws is 7 kW. In the first example of Embodiment 1, when the calorific value of the communication equipments is reduced, there is a case that the compressor is stopped by a thermo-off because the capability of the air conditioner is sequentially reduced in accordance with the reduced calorific value, and finally the calorific value becomes smaller than the minimum capability of the air conditioner. In this example, step S21 is added to judge to minimize the capability of the compressor before the decrement of the calorific value less than the minimum capability of the air conditioner, whereby the thermo-off does not easily occur. Repetition of the thermo-ons and the thermo-offs does not only shorten a lifetime of a compressor but also causes moisture condensation in the casing.

[0041] Further, a process of designating the upper limit value of the frequency in Figure 4 may be changed. As in a flowchart illustrated in Figure 5, the output W from the electric power detecting means 12 is compared with the set value Ws in S51, and the capability of the compressor is minimized in S53 when W≦Ws is established. It is possible to simplify the structure of the cooling control method while maintaining a function of preventing the moisture condensation, caused by the repetition of thermo-ons and thermo-offs. A structure and a block chart of this modification are respectively illustrated in Figure 1 and Figure 2.

EMBODIMENT 2

[0042] Hereinbelow, an example of a cooling control method for a communication station according to Embodiment 2 will be described. Figure 6 illustrates a structure of the cooling control method for the communication station according to Embodiment 2.

[0043] Figure 7 is a block chart of the cooling control method. In Figures 6 and 7, the same numerical references as those in Figures 25, 1 and 2 designate the same or similar portions and description of these portions is omitted. Numerical reference 11b is a cooling control means for controlling a cooling capability of an air conditioner. Numerical reference 9 designates a suction temperature detecting means for an indoor unit of the air conditioner, which detects a suction air temperature into the indoor unit.

[0044] Figure 8 illustrates a flowchart illustrating an operation of the cooling control method. A processes until S5 and S6 is similar to that described in Embodiment 1. In S31, the upper limit fmax of the frequency is obtained as a function f(Tin) having a variable of a suction temperature Tin detected by the suction temperature detecting means 9. When a temperature Tm of the suction air 8 for the communication equipments is constant, there is a relationship between the calorific value of the communication equipments and the suction temperature Tin of the indoor unit:

$Tin \propto$ (calorific value of communication equipment)

[0045] Accordingly, it is possible to substitute the suction temperature for the electric power consumption, described in Embodiment 1 in reference of Figure 3. For example, provided that Tm=20°C (constant), and the airflow rate of the blower 3 is 40 m^3/min, the following equation is established:

$$Tin = Tm + \Delta T = 20 + 1.23W,$$

where the consumption electric power is denoted by W, the airflow rate is 0.67 m^2/sec, the air density is 1.2 kg/m^3, and the specific heat at constant pressure of the air is 1.01 kJ/kg·K,
wherein

$$\text{the equation } \Delta T = (\text{electric power consumption}) / (\text{airflow rate} \times \text{air density} \times \text{specific heat of air at constant pressure}),$$

described in Embodiment 1, is used.

[0046] Further by substituting thus obtained W for W in the equation f(W)=13.7(W-6)+30, described in Embodiment 1, the following equation is obtained:

$$f(Tin) = 11.2Tin - 277$$

[0047] For example, fmax is calculated by such a function.

[0048] In case that the suction temperature Tin is 35°C, the following equation is established:

$$f(Tin) = 115 \text{ Hz},$$

where Tin=35°C.

[0049] This represents a case that a capability of the compressor 5a is linearly changed on a premise that characteristics of the compressor 5a is 12 kW at 112 Hz and 6 kW at 30 Hz. Thus operated fmax from f(Tin) and f1 calculated above are compared in S32. In S33, S34 and S35, the frequency of the compressor is set to be fmax, in case that f1 is larger than fmax; and the frequency is set to be f1 in case that f1 is fmax or less.

[0050] Thus the frequency of the compressor is controlled.

[0051] Further, although it is not illustrated in the flowchart, when thus calculated f1 is out of a frequency band enabling to use the compressor, it is possible to add a function of limiting the frequency at upper and lower limit values.

[0052] There is a case that the calorific value of the communication equipments 2 abruptly changes. In such a case, it is insufficient to control using only the output signal from the suction temperature detecting means 13 for the communication equipments. By checking and controlling the suction temperature Tin reflecting a change of the calorific value of the communication equipments, being a factor for changing the temperature Tm, it is possible to stably control with a good following-up capability.

[0053] In the next, a cooling control method for communication relay stations according to an another example in Embodiment 2 will be described in reference of Figures. In Figures 6 and 7, a structure and a block chart are respectively illustrated.

[0054] Figure 9 is a flowchart illustrating an operation of the cooling control means 11b. In Figure 7, a procedure according to this example of Embodiment 2 is the same until the steps S33 and S34, described in the former example. After calculating the frequency f1 in the step S33 or S34, a suction temperature Tin of the indoor unit, outputted from the suction temperature detecting means 9, is compared with a previously set value Tins of the suction temperature in S41. When Tin>Tins, the frequency of the compressor is set to be f1 in S42. When Tin≦Tins, the capacity of the compressor is minimized in S43. This means that the frequency of the compressor is decreased to a minimum frequency enabling the operation of the compressor. Further, a refrigerating circuit is constructed so as to bypass a part of a refrigerant flowing into the outdoor heat exchanger 5b on a suction side of the compressor (not shown in Figure 6). In this case, a cooling capability can be degraded by making the frequency of the compressor to be the minimum frequency enabling the operation of the compressor. Such an application can also be adopted.

[0055] The set value Tins is determined as follows. Step S41 in Figure 9 has a significance similar to the step S21 in Embodiment 1 with reference to Figure 4. When the temperature Tm of the suction air 8 into the communication equipments is constant, there is the following relationship between the calorific value of the communication equipments and the suction temperature Tin of the indoor unit:

Tin∝ (calorific value of communication equipment)

[0056] Accordingly, it is possible to substitute the suction temperature for the electric power consumption described in reference of Figure 4. Because the set temperature Tins is little higher than the minimum capability of the air conditioner, when the minimum capability is 6 kW as in Figure 4, for example, Tin with respect to the electric power consumption of 7 kW is determined. At this time, a temperature difference ΔT between the suction air 6 and the suction air 8 for the communication equipments with respect to the electric power consumption of 7 kW becomes ΔT=8.7deg, in use of the above-mentioned constants and the equations. When Tm is 20°C, Tin=Tm+ΔT=28.7°C, wherein Tins=28.7°C.

[0057] In Figure 9, as described in Embodiment 1 with reference to Figure 5, a structure of the method can be simplified by directly connecting S4, S5 and S6 to S32 and removing S31, S32, S33, and S34 in Figure 9, whereby a function of preventing moisture condensation caused by repetitions of thermo-ons and thermo-offs.

[0058] Further, it is possible to substitute S21, S22, and S23, described in Embodiment 1 with reference to Figure 4, for S41, S42, and S43 in Figure 9. In other words, it is controlled as illustrated in a flowchart of Figure 11, whereby when the calorific value of the communication equipments is reduced, the compressor is seldom stopped, i.e. thermo-off by sequentially reducing the capability of the air compressor and thereby the calorific value becomes smaller than the minimum capability. Thus a lifetime of the compressor caused by the repetition of thermo-ons and thermo-offs is prevented from reducing, and the moisture condensation can be prevented. In this case, the electric power detecting means 12 is added to Figure 6, and a block chart illustrating this case is in Figure 10. Further, it is possible to substitute S41, S42, and S43, illustrated in Figure 9, for S21, S22, S23, described in Embodiment 1 with reference to Figure 4. In other words, by controlling like a flowchart illustrated in Figure 12, similar function and effect are obtainable. In this case, the electric power detecting means 12 is added to Figure 6, and a block chart illustrating this case is in Figure 10 as above.

[0059] Further, although in Embodiments 1 and 2, the cases of detecting electric power consumption of electronic boards of the communication equipments is detected by the electric power detecting means 12, the electric power consumption may be substituted by a current of the communication equipments. There is an effect of detecting the electric power consumption by calculating the electric power consumption from a detected total value of the current of the communication equipments, accommodated in the casing, using an ampere meter, of which cost is lower than that of the electric power meter, so as to form the electric power detecting means.

EMBODIMENT 3

[0060] Hereinbelow, an example of a cooling control method for communication stations according to Embodiment 3 will be described. Figure 13 illustrates a structure of the cooling control method for the communication station according to Embodiment 3.

[0061] Figure 14 is a block chart of the cooling control method. Figure 15 is a flowchart illustrating a control by the cooling control method. In Figures 13 and 14, numerical references same as those described in Embodiments 1 and 2 designate the same or similar portions and description of these portions is omitted. Numerical reference 11c designates a cooling control means for controlling a cooling capability. Numerical reference 20a designates an initial target suction temperature value setting means for the communication equipments. Numerical reference 20b designates an initial target suction temperature value setting means for the indoor heat exchanger. Numerical reference 20d designates a target value determining means for determining a target value of the suction temperature for the indoor heat exchanger and a target value of the suction temperature for the communication equipments. Numerical reference 21d designates an air condition control means for controlling a capability of the air conditioner based on the target value f the suction temperature for the communication equipments determined by the target value determining means 20d. Numerical reference 20e designates a memory for memorizing outputs detected by the suction temperature detecting means 9 and the suction temperature detecting means 13. Numerical reference 20f designates a timer.

[0062] The cooling control means 11c is formed by the initial target suction temperature setting means 20a, the initial target suction temperature value setting means 20b, the target value determining means 20d, the memory 20e, the timer 20f, the air condition control means 21d, the frequency control means 22, and so on.

[0063] In general, when an air conditioner is operated by increasing the suction temperature less than a limit value, ordinarily 40°C or less, an efficiency is improved as in Figure 16, wherein a sensible cooling capability in the ordinate is increased along with an increment of the suction temperature into the air conditioner in the abscissa. In Figure 13, when the suction temperature detected by the suction temperature detecting means 9 is maintained as high as possible, the efficiency is improved. In order to attain this state, the following control is performed.

[0064] A controlling operation by the cooling control means 11c will be described in reference of a flowchart illustrated in Figure 15. The target initial value Tinso, set by the suction temperature initial target value setting means 20b, and the initial target value Tso, set by the suction temperature initial target value setting means 20a for the communication equipments, are read out in S101. In S102, the read out values are used respectively as the suction temperature target value Tins and the suction temperature target value Ts for the communication equipments. In S103, the suction temperature target value Ts, being equal to Tso, of the communication equipments is inputted into the air condition control means 21d. The air condition control means 21d compares the output Tm from the suction temperature detecting means 13 for the communication equipments with the suction temperature target value Ts for the communication equipments, controls the frequency control means 22, and controls a capacity of the compressor 5a, whereby the capability of the air conditioner is controlled. The capability of the air conditioner is controlled in accordance with, for example, S1 through S6 illustrated in Figures 3, 4, 5, 8, 9, 11 and 12 and described in Embodiments 1 and 2. The suction temperature Tin of the indoor unit and the suction temperature Tm of the communication equipments, obtained as a result of the control, are respectively detected by the detecting means 9 and 13, and the detected values are memorized every minute in S105. In S104 and S106, ten minutes are counted. In S107, the suction temperature detected values Tin for the ten minutes are read out from the memory 20e, and an average value Tin10 of the suction temperature detected values is calculated by the target value determining means 20d. The average value and the suction temperature target value Tins are compared in S108. When a difference between the average value and the suction temperature target value Tins is smaller than a range of ±1°C, the suction temperature target value Ts for the communication equipments is unchanged in S109. In S110, it is judged whether or not the average value Tin10 is higher than the suction temperature target value Tins by 1°C or more. When higher by 1°C or more, the suction temperature target value Ts for the communication equipments is decreased by 1°C in S111. On the other hand, when the average value is lower than the suction temperature target value Tins plus 1°C in S110, the suction temperature target value Ts for the communication equipments is increased by 1°C in S112. In S103, thus determined suction temperature target value Ts for the communication equipments are inputted into the air condition control means 21d again. These operations are repeated.

[0065] By setting the suction temperature target value Tins as high as possible so as not to exceed the limit value, it is possible to use the air conditioner at a high temperature range as close as possible to the suction temperature target value Tins, whereby the air conditioner is in a highly efficient state. Further, moisture condensation can be prevented because the blowing-out temperature from the air conditioner is increased. As an example of this embodiment, Figure 17 illustrates a case that the initial value of the suction temperature target value Ts for the communication equipments is 30°C and the initial value of the suction temperature target value Tins is 35°C. In Figure 17, the capability of the air conditioner is balanced with the loads, and the suction temperature tin, the suction temperature Tm for the communication equipments, and the blowing-out temperature for the air conditioner becomes even with a lapse of time. The reason why the average value of the detected values of the suction temperature for every ten minutes is used for controlling the air compressor is to avoid disturbance in the control, caused by a temporary change of the temperature in the communication station. In general, there is a case of controlling the frequency by every one minute for controlling a capability of an air conditioner by an air condition control means. In this case, frequent changes in a suction temperature target value Ts for the communication equipments make the control unstable. As described in this embodiment, by automatically setting and changing the suction temperature target value Ts for the communication equipments, it is possible to use the air conditioner in a most preferable manner in conformity with actual conditions of the loads.

[0066] In the next, another example will be described. Figures 13 and 14 respectively illustrate a structure and a block chart of this example. A flowchart for controlling is illustrated in Figure 18. As described in reference of Figure 17, when the calorific value of the communication equipments and the cooling capability of the air conditioner are balanced, the control, described in the prior example, can be adopted. However, the air condition control means 21d of the air conditioner is in a thermo-off condition when the suction temperature Tm for the communication equipments becomes lower than the suction temperature target value Ts for the communication equipments by a predetermined value. This state is illustrated in Figure 19. In this case, there are possibilities of:

(1) causing the suction temperature Tm for the communication equipments to temporarily outstrip the suction temperature target value Ts for the communication equipments; and

(2) adversely affecting the lifetime of the air conditioner because frequent thermo-ons and thermo-offs occur.

[0067] Especially, as for (2), when the suction temperature Tin is increased as high as possible so as to attain the aim of the control according to this embodiment, the suction temperature Tm into the communication equipments is resultantly increased. When the suction temperature target value Ts for the communication equipments is high, a time that the suction temperature Tm for the communication equipments reaches by decreasing to the suction temperature target value Ts for the communication equipments after starting to cool by making the air conditioner thermo-on.

[0068] Further, it is generally necessary to control the air conditioner for protecting the compressor, wherein the thermo-on should be prevented for three minutes. Because a cooling function is stopped for the three minutes, a temperature of the communication station is increased as illustrated in Figure 19. Although a tendency of the increment depends on the calorific value of the communication equipments inside the communication relay station, when the calorific value is constant, a temperature to be gained inside the communication relay station is high as the suction temperature target value Ts for the communication equipments is high because of the period between the thermo-off and the thermo-on. In general, when an actually measured value is a target value such as the suction temperature target value Ts for communication equipments plus 1°C, an air conditioner is in a state of thermo-on, and when the actually measured value is the target value minus 1°C, the air conditioner is in a state of thermo-off, wherein even though criteria for the thermo-on and the thermo-off are changed, the above-mentioned tendency similarly occurs.

[0069] In order to solve the above-mentioned problems (1) and (2), the following process is added to the above control. When at least one of conditions that the suction temperature Tm once exceeds a certain limit value, for example 35°C, within the ten minutes and that the thermo-on occurs twice or more within the ten minutes, the suction temperature target value Tins is decreased by 1°C.

[0070] Such an operation will be described in reference of Figure 18. An explanation, already have described in the example in Figure 15, is omitted. The air condition control means 21d judges the thermo-on and thermo-off in the air conditioner. In case of the thermo-on, the thermo-on is reported to the target value determining means 20d, wherein the target value determining means 20d counts the number of the thermo-ons, and the memory 20e memorizes the information and a detected value of the suction temperature Tm for the communication equipments in S105b. After a lapse of ten minutes, in S201, the target value determining means 20d checks whether or not Tm at least once exceeds 35°C during the ten minutes or not by calling out the detected value out of the memory. In S202 and S203, Flag is set to be 1 when Tm has thus exceeded. In S204, the target value determining means 20d checks whether or not two times or more thermo-ons have occurred during the ten minutes. In case of the two times or more, Flag 2 is set to be 1 in S205 and S206. In S207, it is judged whether or not at least one of Flag and Flag2 is 1. If so, in S209, the target value Tins of the suction temperature is decreased by 1°C. Tins is maintained to be the same when both of the flags are 0 in S208. After determining the suction temperature target value Tins by the target value determining means 20d, the suction temperature target value Ts for the communication equipments for controlling next ten minutes is determined in S107 through S112, similar to those in the above embodiment, based on the average value Tin10 of the suction temperatures in the ten minutes after determining the suction temperature target value Tins by the target value determining means 20d. This procedure is repeated in a similar manner to that described in the above embodiment.

[0071] In this example, it can be misunderstood that a running efficiency is in a tendency of deteriorating by decreasing the suction temperature target value Tins. However, this makes the air conditioner run with a best efficiency and without a danger of generating moisture condensation if a presupposition of controlling in the present invention that the suction temperature for the communication equipments is maintained to be a predetermined value or less is satisfied, wherein the control is based on a worm's eye view.

[0072] In the next, another example of this embodiment will be described.

[0073] In the above-described two examples of Embodiment 3, the control means for determining the target value, by which a stable usage is obtainable for a certain condition of the load. In an actual application, because there is a case that the operation is troubled, for example, moisture condensation occurs in the communication station and a usable temperature range of the air conditioner is escaped, by decreasing or increasing the suction temperature target value Tins and/or the suction temperature target value Ts without limit.

[0074] A structure of the system is illustrated in Figure 13, and a block chart is illustrated in Figure 20 as in the above examples. In Figure 20, numerical reference 20g designates a suction temperature target lower limit value setting means; and numerical reference 20h designates a suction temperature target upper limit and lower limit values setting means. The cooling control means 11d is constructed by adding the suction temperature target lower limit value setting means 20g and the suction temperature target upper and lower limit value setting means 20h to the above-mentioned 11c. In use of a flowchart illustrated in Figure 21, points different from described above will be described. An explanation of the portions same as described above is omitted. In S101b, a lower limit value Tinsmin of the suction temperature target value, set by 20g, and a lower limit value Tsmin and an upper limit value Tsmax of the suction temperature target value, both set by 20h, are read. It is detected for ten minutes like the above example. After determining the suction temperature target value Tins depending on a value of the flag, it is compared whether or not the determined value Tins is smaller than the lower limit value Tinsmin in S210. If small, Tsin=Tsmin in S212. If large, the determined value Tins is used as in S212. After determining the suction temperature target value Ts by comparing an average value of the suction temperatures for the ten minutes with the suction temperature target value, the suction temperature target value Ts is compared with the upper limit value Tsmax and the lower limit value to determine whether or not the suction temperature target value Ts exceeds the upper limit value Tsmax or is smaller than the lower limit value Tsmin, respectively in S304 and S301. If exceeds the upper limit value, in S306, the target value Ts is equal to Tsmax. If smaller than the lower limit value, in S303, Ts is equal to Tsmin. If between the upper limit value and the lower limit value, in S302 and S305, the determined Ts is used. Meanwhile, the upper limit value of the suction temperature target value of the suction temperature target value corresponds to the target initial value Tinso, set by the suction temperature target initial value setting means 20b.

[0075] In the above examples, the control means determines the target values enabling the stable operation. However, there is a possibility that these target values in the stabilized condition do not always realize a most stable operation depending on hours and seasons. This is because, the capability of the air conditioner is influenced by an outer air temperature and there is a change, caused by heat and so on penetrating through a wall of the casing 10. In order to deal with these problems, as illustrated in S401 and S402 in the flowchart illustrated in Figure 21, all values are set back to initial values by every predetermined time, for example, every six hours, and a most suitable target value for the time point is searched.

EMBODIMENT 4

[0076] Even though other air conditioners and auxiliary cooling devices such as heat pipes and boiling type cooling devices in addition to a main cooling device of the above-described air conditioner as the cooling means, the process, described in Embodiment 3, can be similarly proceeded. A system structure in this case is illustrated in Figure 22. In Figure 22, numerical reference 30a designates an evaporator, i.e. an air cooler of the auxiliary cooling device. Numerical reference 30b designates a condenser, i.e. a radiator, of the auxiliary cooling device. Numerical reference 30c designates an intake air into the evaporator. Numerical reference 30d designates a blowing-out air. The auxiliary cooling device 30 is constructed by the evaporator 30a, the condenser 30b, and so on. The auxiliary cooling device 30 may be located at any place inside the casing 10. As long as the auxiliary cooling device 30 is constantly and independently controlled, it is possible to process in a manner completely similar to that described in Embodiment 3, wherein the calorific value of the communication equipments 2 and removed heat by the auxiliary cooling device 30 are treated as the load inside the communication station and the removed heat is treated as an added heat. In other words, all examples described in Embodiment 3 can be applied to this structure in Embodiment 4.

[0077] Another Example of Embodiment 4 will be described. A structure is illustrated in Figure 23. As illustrated in Figure 23, by controlling a shut-down of the auxiliary cooling device 30 by a cooling control means 11f of the main cooling device, it is possible to more effectively operate the air conditioner. A cooling capability of a main cooling device is generally larger than that of an auxiliary cooling device. However, when both of these are independently operated, there is a case that an input is excessively increased by combining these operations, for example, the auxiliary cooling device is continued to run even though an internal load can be covered by the main cooling device. Further, when the load is small, there is a case that only the auxiliary cooling device can cover the load. In this case, by appropriately selecting these devices, an effective operation is demonstrated in consideration of an entire system.

EMBODIMENT 5

[0078] A structure according to Embodiment 5 is illustrated in Figure 24. The auxiliary cooling device, described in Embodiment 4, is substituted by a boiling type cooling device 31 including an evaporator 31a and a condenser 31b, wherein the evaporator 31a is located on an upper stream side in the airflow path, in which the main cooling device is located. The boiling type cooling device 31 has a characteristic that as a difference between an evaporative temperature in the evaporator 31a and a condensing temperature in the condenser 31b located in an outdoor unit of the boiling type cooling device increases, a capability is further demonstrated, basically in proportion thereto. Further, the boiling type cooling device 31 is inputted from only a blower 31c. Therefore, it is possible to highly efficiently use the air conditioner by making the air path of the indoor unit commonly use for that of the air conditioner and the air path of the indoor unit commonly use for the boiling type cooling device and the air conditioner.

[0079] Therefore, for example, a suction temperature into the indoor unit, i.e. the evaporator 31a, of the boiling type cooling device 31 is set to be as high as possible while satisfying conditions that (suction temperature for communication equipment Tm)$\leqq$35°C and (suction temperature for indoor heat exchanger Tin)$\leqq$40°C.

[0080] By controlling in a manner similar to that described in Embodiment 3, the above example can be realized.

However, it is necessary to locate a detecting portion of the suction temperature detecting means 9 ahead the evaporator, i.e. the indoor heat exchanger 4a, of the air conditioner, being the main cooling device.

[0081] When the detected value by the suction temperature detecting means 9 is smaller than a predetermined value, for example 20°C. The suction air temperature into the communication equipment is controlled to be 35°C or less. However, when it is excessively lowered, the following problems occur:

(1) There is a lower limit in a circumstance of a temperature, at which accommodated equipments are used. Generally, it is 0°C or more, but 20°C or more is preferable for a battery;
(2) As an indoor temperature is low, the suction air temperature approaches a dew point of an indoor air and moisture condensation is apt to occur; and
(3) An energy is not properly saved. Further, other problems may occur.

[0082] In such a case, the blower of the outdoor unit of the auxiliary cooling device 31 is stopped. The shut-down of the blower of the outdoor unit demonstrates various effects in the following cases in addition to the above-mentioned situation, where a detecting means, a determining means, and so on are not specifically mentioned.

(1) An outer air temperature is lower than a predetermined temperature;
(2) A difference between detected values of the suction temperature for the boiling type cooling device and of the outer air temperature is larger than a predetermined value;
(3) A difference between the suction temperature and the blowing-off temperature, both of the heat exchanger in the outdoor unit of the boiling type cooling device, is larger than a predetermined value;
(4) A difference between an inlet temperature and an outlet temperature, both of a refrigerating tube for the outdoor unit of the boiling type cooling device, is larger than a predetermined; and
(5) A detected value by the suction temperature detecting means for the communication equipments is lower than a predetermined temperature.

[0083] Further, in the following cases, corresponding measures are adopted instead of stopping the blower in the outdoor unit.

[0084] At first, if there is a danger that an air conditioner can not be restarted after once stopping a blower of an outdoor unit because snow falls and piles up, the following measures are adopted.

(1) Decreasing but not stopping the number of revolution of the blower in the outdoor unit;
(2) Without stopping the blower in the outdoor unit, the blowing by the indoor unit of the main cooling device is intermittently operated; and
(3) Without stopping the blower in the outdoor unit, the number of revolution of the blower in the indoor unit of the main cooling device is decreased.

[0085] Further, when a difference between the outdoor air temperature and the suction air temperature for the communication equipments is not sufficiently kept, because there is a danger that the capability of the boiling type cooling device is inferior to the input, namely a COP becomes less than 1, the blower in the outdoor unit of the boiling type cooling device is stopped, whereby an energy is saved. For example, when the outdoor air temperature is 10°C or more and snow does not pile up, a temperature difference is measured, and if it is 1°C or less, the outdoor unit is stopped.

[0086] In Embodiments 3, 4, and 5, even though a plurality of air conditioners, i.e. main cooling devices, exist in the casing 10 of the communication relay station, or a total number of air conditioning apparatuses, i.e., main cooling devices, and auxiliary cooling devices is plural, it is possible to determine a most suitable operation with respect to a condition of heat from communication equipments accommodated in the communication relay station by starting a process from initial values respectively of the main cooling devices. In case of the plurality of cooling devices, there is a possibility that cooling functions interfere each other. However, the most suitable state can be automatically determined in consideration of a relationship between the plurality of cooling devices.

[0087] However, the suction temperature target value Tins of the main cooling device can only be decreased for the operation. This is because, a characteristic of the control is to decrease the suction temperature target value Tins from the initial value, for example, 35°C, of the suction temperature target value Tins while searching a maximum value, which does not causes problems.

[0088] There is a case that a calorific value in the communication relay station abruptly changes depending on a condition of the communication. However, it does not largely change in general, even though some changes stationary occur. Further, a capability of boiling type cooling device, used as the auxiliary cooling device, changes depending on an outer air temperature. Accordingly, a state of various target values, attained after the above-mentioned process started at a certain time point, does not always most suitable values at other time points. Since it is not processed to increase the suction temperature target value Tins, all target values are returned to initial values at a certain time point, e.g. after six hours from starting a previous process, and after six hours from any change of the suction temperature target value Tins, and a process for acquiring the most suitable state, whereby the most suitable operation is realized.

[0089] In the above Embodiments 1, 2, 3, 4 and 5, all values concerning the temperatures, the times, and so on are given as examples and may be changed depending on conditions.

EMBODIMENT 6

[0090] An operation of the cooling control method for the communication station according to Embodiment 6 will be described in reference of Figures 1 and 2. In the cooling control method for the communication station, the suction air temperature in the communication equipments 2 is controlled to be within a normal temperature by supplying a predetermined amount of the suction air 8 to the communication equipments 2 by the fan 4b. In general, the temperature of the suction air 8 is controlled to be 20°C or less. The suction air 8 is heated after cooling the communication equipments 2, sucked into the indoor unit 4 as the suction air 6, cooled by the indoor heat exchanger 4a, returned to the casing 10 as the blowing out air 7, and again provided to cool the communication equipments 2 as the suction air 8 for the communication equipments. The cooling control means 11a controls the temperature of the suction air 8 to be the set temperature, e.g. 20°C, or less based on outputs from the suction temperature detecting means 13 for the communication equipments and the electric power detecting means 12. Provided that the airflow rate by the fan 3 is 40 m^3/min and the electric power detected by the electric power detecting means 12 is 12 kW, the consumption electric power and the calorific value is substantially the same in the communication equipments 2, of which electric power consumption is consumed mostly by electronic circuit board, whereby a temperature difference ΔT between the suction air 6 and the suction air 8 becomes ΔT=(electric power consumption)/(airflow rate x air density x specific heat of air at constant pressure). The electric power consumption is 12 kW, the airflow rate is 40 m^3/min, the air density is 1.2 kg/m^3, and the specific heat of the air at constant pressure is 1.01 kJ/(kg·K), whereby ΔT=15deg. Provided that the temperature of the suction air 8 is 20°C, the temperature of the suction air 6 becomes 20°C+ΔT=20°C+15deg=35°C. When the airflow rate by the fan 4b in the indoor unit is 40 m^3/min, in order to supply the suction air 6 as the suction air 8 after cooling to be 20°C, a capability as much as 12 kW becomes necessary. It is supposed that the capability of the air conditioner follows an actual calorific load by detecting the electric power consumption by the communication equipments and controlling the capability. However, the blowing-out air 7 in the indoor unit is mixed with an ambient atmosphere in the casing and served as the suction air 8 for the communication equipments, whereby the temperature of the suction air 8 does not become the above-mentioned temperature. In order to correct this, a requisite capacity of the compressor is calculated so a to make the suction air 8 basically have the set target value from the suction air temperature target value setting means 20 by comparing the temperature outputted from the suction air temperature detecting means 13 for the communication equipments and the set target value. Further, an upper limit of a maximum frequency of the compressor is calculated from an electric power detected by the electric power detecting means 12. Then the requisite capacity is corrected and the compressor 5a is controlled through the frequency control means 22 upon a command of frequency outputted from the air condition control means 21a.

[0091] Figure 3 is a flowchart for illustrating a control operation by the cooling control means. The air condition control means 21a confirms a frequency f currently outputted to the compressor 5a.

[0092] In the next, the suction temperature detecting means 13 for the communication equipments checks the temperature Tm of the suction air 8 and the set temperature Ts of the suction air 8 in step S2. When these are not equal, in step S3, it is compared whether or not the temperature Tm exceeds the set temperature Ts. When the temperature Tm exceeds the set temperature Ts, in step S5, the frequency of the power source for the compressor 5a is increased through the frequency control means 22. When the temperature Tm does not exceed the set temperature Ts, in step S6, the frequency of the power source for the compressor 5a is decreased through the frequency control means 22. When the temperature Tm equals to the set temperature Ts in step S2, in S4, the frequency is unchanged in step S7, an upper limit fmax of the frequency of the compressor is operated by the air condition control means 21a.

[0093] The upper limit fmax is obtained from a function f(w) having a variable of the output W from the electric power detecting means 12. This relation is set, for example, as follows:

$$f(w) = 13.7(W-6) + 30$$

[0094] This is a case that a characteristic of the compressor 5a is such that 12 kW at 112 Hz and 6 kW at 30 Hz, and the capacity of the compressor linearly changes in frequencies between 112 Hz and 30 Hz. In step S8, fmax operated from f(w) and the calculated f_1 are compared. When f_1 is larger than fmax, the frequency of the compressor is fmax, and when f_1 is fmax or less, the frequency of the compressor is f_1 in steps S9, S10, and S11. The frequency of the compressor is controlled as described.

[0095] There is a case that the calorific value of the communication equipments 2 abruptly changes. In such a case, it is insufficient to control using only the output signals from the suction temperature detecting means 13 for the communication equipments, which output signal is obtained as a result of the change. Therefore, the output signal from the electric power consumption detecting means 12, causing a change of the temperature Tm, is previously obtained to control the air condition, whereby a stable control with a good follow-up capability is achieved.

[0096] Figure 4 is a flowchart illustrating the cooling control method according to another example of Embodiment 6. The procedure until steps S9 and S10 is similar to that described above. The cooling control means in this cooling control method is illustrated in Figure 2. After calculating the frequency f_1 in the step S9 or S10, in step S21, the electric power consumption W, i.e. output from the electric power detecting means 12, at the time point is compared with the set value f_1 of the electric power consumption previously set. When W>Ws, in step S22, the frequency of the compressor 5a is set to be f_1. When W>Ws, in step S22, the frequency of the compressor 5a is set to be f_1. When W≦Ws, the capacity of the compressor is minimized. This means the frequency of the compressor is minimized without avoiding the operation of the compressor, or a refrigerating circuit is formed to bypass a part of a refrigerant flowing into the outdoor heat exchanger 5b, wherein a bypass circuit is not illustrated in Figure 1. The set value Ws is made a little larger than the minimum capacity of the air conditioner. For example, when the minimum capacity of the air conditioner is 6 kW, the set value Ws is 7 kW. When the calorific value of the communication equipments is reduced in the first example of Embodiment 6, there is a case that the compressor is stopped, i.e. thermo-off, when the capacity of the air conditioner is succeedingly reduced and the calorific value becomes less than the minimum capacity. However, in the example, a determination in the step S21 is further added, and the capacity of the compressor is minimized before the calorific value is reduced less than the minimum capacity of the air conditioner, whereby the thermo-off seldom occurs even in such a case.
Repetitions of thermo-ons and thermo-offs do not only shorten a lifetime of a compressor but also causes moisture condensation in a casing, these problems can be prevented in this example.

EMBODIMENT 7

[0097] Hereinbelow, an example according to Embodiment 7 of the present invention will be described.

[0098] Figure 6 illustrates a structure of the cooling control method for the communication station according to Embodiment 7. Figure 25 is a block chart illustrating the cooling control method according to Embodiment 7. In Figures 6 and 25, numerical references the same as in Figures 1 and 2 designate the same or similar portions and description of these portions is omitted. Numerical reference 11b designates the cooling control means for controlling the cooling capability of the air conditioner.

[0099] An operation of the cooling control method for the communication station according to Embodiment 7 will be described in reference of Figures. Figure 26 is a flowchart illustrating an operation of the cooling control method 11b. A procedure until the steps S9 and S10 is similar to that in the examples of Embodiment 6. After calculating the frequency f_1 in the step S9 or S10, in step S31, a suction temperature Tin, i.e., an output from the suction temperature detecting means 9, of the indoor unit at that time is compared with a set value Tins of the suction temperature previously set. When Tin>Tins, in step S32, the frequency of the compressor is set to be f_1. When Tin≦Tins, a capacity of the compressor is minimized. This means the frequency of the compressor is minimized within an operable range for the compressor, or a refrigerating circuit is formed to bypass a part of a refrigerant flowing into the outdoor heat exchanger 5b, wherein a bypass circuit is not illustrated in Figure 6.

[0100] The set value Tins is set as follows. Step S31 in Figure 26 has similar significance to the step S21 in Figure 4. In other words, as long as the temperature Tin of the suction air 8 for the communication equipments is constant, there is a relationship between the calorific value of the communication equipments and the suction air Tin for the indoor unit:

Tin∝ (calorific value of communication equipment)

[0101] Accordingly, it is possible to substitute the suction temperature for the electric power consumption, described in the example of Embodiment 6 in reference of Figure 4. For example, the set value Tins is Tin corresponding to an electric power consumption of 7 kW. A temperature difference ΔT between the suction air 6 and the suction air 8 with respect to the electric power consumption 7 kW becomes ΔT=8.7deg based on the above-mentioned constants and equations. When Tm=20°C, Tin=Tm+ΔT=28.7°C. Accordingly, Tins may be set to be 28.7°C.

EMBODIMENT 8

[0102] In the above-mentioned examples of the embodiments, there are relationships between the calorific value of the communication equipments and the suction temperature. Therefore, the output from the electric power detecting means 12 and the output from the suction temperature detecting means may be exchanged. This case is illustrated in Figures 28 and 29 as Embodiment 8.

[0103] It is also possible to modify the process of designating the upper limit value of the frequency in the example of Embodiment 6 to be step S51, in which the output W from the electric power detecting means 12 is compared with the set value Ws, and when W≦Ws, the capacity of the compressor is minimized.

[0104] By this, while maintaining the function of preventing moisture condensation caused by repetitions of thermo-ons and thermo-offs, the structure of the cooling control method is simplified. Also in this example, the judgement in the step S41 may be based on the suction temperature not the electric power as described.

EMBODIMENT 9

[0105] Figure 30 schematically illustrates a structure of a cooling system of a casing in a communication station according to Embodiments 9, 11, and 12. Constitutional elements corresponding to the conventional cooling system 151, illustrated in Figure 34, are attached with the same numerical references, and description of these constitutional elements is omitted.

[0106] In Figure 30, the cooling system 101 for the casing is constructed to cool inside the casing 103 in the communication station 102 forming an enclosed space using a boiling type cooler 121 of a natural circulation refrigerating circuit 120 and an evaporator 113 in a forced circulation refrigerating circuit 109, through which a refrigerant is forcibly circulated by a compressor 110. Communication equipments 104 including heat components are accommodated in the casing 103.

[0107] In the casing 103 of the cooling system 101, a common airflow path 130 is located. The common airflow path 130 includes a common case 129 shaped like a hollow box having a heat air intake port 132 for taking a heat air in the casing 103 and a cold air exhaust port 113 for blowing a cold air into the casing 103. In the common airflow path 130, the boiling type cooler 121 in the natural circulation refrigerating circuit 120, the evaporator 113 in the forced circulation refrigerating circuit 109, and a common fan 131 for blowing an air to the boiling type cooler 121 and the evaporator 113 are built.

[0108] A condenser 111 in the forced circulation refrigerating circuit 109 is installed in a condenser case 117 as an outdoor unit. The condenser case 117 is shaped like a box having an outer air intake port 118 and an outer air exhaust port 119. Further, the condenser case includes the condenser 111, the compressor 110, a refrigerant choke valve 112, and a fan 116. The forced circulation refrigerating circuit 109 is constructed by sequentially connecting the compressor 110 in the condenser case 117, the condenser 111, the refrigerant choke valve 112, and the evaporator 113 in the common airflow path 130 through refrigerating pipes 114, 115 in an annular shape.

[0109] The condenser 122 in the natural circulation refrigerating circuit 120 is located in the condenser case 125 as the outdoor unit. The condenser case 125 is shaped like a box having an outer air intake port 127 and an outer air exhaust port 128 and includes the condenser 122 and a fan 126. In other words, the natural circulation refrigerating circuit 120 is constructed by connecting the condenser 122 in the condenser case 125 and the boiling type cooler 121 in the common airflow path 130 through a refrigerant evaporating pipe 123 and a liquid refrigerant return pipe.

[0110] An operation of the cooling system 101 for the casing in the communication relay station 102 according to the above-mentioned structure will be described.

[0111] A cooling air in a position 301 on a side of an equipment case 106 is taken into the equipment case by a fan (not shown) in the communication equipments 104. The cooling air cools heat components 105, is changed to a heat air, and is blown out of the exhaust port in an upper portion of the equipment case into a position 303 inside the casing 103. Thus blown-out heat air is taken into the common airflow path 130 from a position 304 through the heat air intake port 132 by the common fan 131. In the common airflow path 130, the heat air passes through the boiling type cooler 121 to primarily exchange heat with a refrigerant in the natural circulation refrigerating circuit 120. An air subjected to the primary cooling in position 305 is sucked by the common fan 131, and passes through the evaporator 113 in its entirety, whereby the air exchanges heat with a refrigerant in the forced circulation refrigerating circuit 109. Thus cooled air is blown out into a position 300 in the casing 103 from the cooled air exhaust port 133 as a cooing air. In other words, the air cools inside the casing 103 by sequentially circulating the positions 300, 301, 302, 304 and 305 in this order.

[0112] In a refrigerant system in the natural circulation refrigerating circuit 120, a refrigerant in the boiling type cooler 121 is boiled by exchanging heat with the heat air and reaches the condenser 122 through the refrigerant evaporating pipe 123. In the condenser 122, a gas refrigerant changes to a liquid refrigerant by condensing as a result of the heat exchange with an outer air passing through the condenser case 125 from the outer air intake port 127 to the exhaust port 128 by the fan 126. The liquid refrigerant returns to the boiling type cooler 121 through the liquid refrigerant return pipe 124 by a gravity flow caused by a difference of weight density from the gas refrigerant.

[0113] On the other hand, in a refrigerant system of a forced circulation refrigerating circuit 109, a high-temperature high-pressure gas refrigerant forcibly discharged from the compressor 110 flows into the condenser 111 and cooled by heat exchange with an outer air passing through the condenser case 117 from the outer air intake port 118 to the exhaust port 119. The liquid refrigerant is depressurized by the refrigerant choke valve 112 so as to be in a gas liquid two-phase state, and reaches the evaporator 113 through the refrigerant pipe 114. The refrigerant exchange heat with the air subjected to the primary cooling in the evaporator 113 so as to be a low-pressure gas refrigerant, and returns on an intake side of the compressor 110 through the refrigerant pipe 115. The capacity of the compressor 110 is controlled based on an air temperature at the position 305 inside the common airflow path 130.

[0114] As described, according to the cooling system 101 for the casing according to this embodiment, since the boiling type cooler 121 and the evaporator 113 are positioned inside a single common airflow path 130, it is sufficient to locate one common fan 131 inside the common airflow path 130. Further, because a space for locating the fan, which is required for two units of fans in the conventional technique, is for one unit. Therefore, a fan providing a high airflow rate can be adopted as the common fan 130. Accordingly, a capability of the natural circulation refrigerating circuit 120 can be expansibly utilized. Meanwhile, the air subjected to the primary cooling by the boiling type cooler 121 is always introduced into the evaporator 113 to be sufficiently cooled. Therefore, there is no danger that bypassing of an air and a short cycle occurs in the casing 103 as in the conventional technique, whereby a cooling efficiency becomes high and an energy can be saved. Further, because the heat air is not directly sucked into the evaporator 113, it is possible to prevent failures of the forced circulation refrigerating circuit 109. Further, the heat air guiding path 157, illustrated in Figure 34 required in the conventional system, is not always necessary.

[0115] As described, it is possible to reduce the number of fans and a running cost, whereby the energy can be saved; a total cooling capacity in the air conditioning system is reduced; reliability of the total cooling system is improved; and a noise can be reduced by reducing a running capacity of the compressor 110.

[0116] Incidentally, in order to improve the efficiency of the system, or preventing moisture condensation in the casing 103, it is effective to operate the system by increasing the temperature inside the casing as high as possible as long as an ambient temperature required in using the communication equipments 104 and so on. However, when the temperature inside the casing is increased, for example 30°C or more, an air temperature in the position 303, where the air is blown out of the equipment case 106, is increased, for example 40°C. If the air having such a temperature is directly sucked into an ordinary indoor unit of an air conditioner, for example the evaporator case 153 in Figure 34, an assured range for operation of the air conditioner indoor unit, for example 35°C, overpassed. As described, by pairing and arranging the boiling type cooler 121 and the evaporator 113, the suction temperature for the evaporator 113 is kept within the assured range, and succeedingly the reliability of a cooling system is improved.

[0117] Further, the boiling type cooler 121 in the common case 129 as a function of demonstrating a high capability as a difference between a temperature of the suction air in the position 304 and a temperature of an outer air sucked into the condenser 122 in the position 306 as illustrated in Figure 32. Therefore, it is efficient to exchange heat with a heat air positioned possibly closest to the heat components 105. For this, it is preferable to arrange the heat air intake port 132 in the common case 129 at a position just above the heat components in this case, the temperature of a heat air blown out of the exhaust port 108 of the communication equipments 104 is substantially maintained and becomes the same as that of an air, sucked into the common airflow path 130 in the position 304.

EMBODIMENT 10

[0118] Figure 31 schematically illustrates the cooling system for a casing of the communication station according to Embodiment 10. The cooling system 101a of a communication station 102a differs from the above-described cooling system 101 at a point that the common airflow path is constructed by an airflow path 148 having a built-in boiling type cooler 121, an airflow path 150 on the evaporator side having a built-in evaporator 113, and a connecting airflow path 149 for connecting the airflow path 148 and the airflow path 150. More specifically, an exhaust port 146 of a cooler case 143 accommodating the boiling type cooler 121 is connected to an intake port 147 of an evaporator case 145 accommodating the evaporator 113 through a connecting duct 144, wherein a common airflow path 130a is formed inside these. Because an operation of the cooling system 101a for the casing is substantially the same as that described in Embodiment 9, description is omitted.

[0119] In thus constructed cooling system 101a, it is possible to use an air conditioner, corresponding to the evaporator case 145 having the intake port 147 and a cold air exhaust port 133, the evaporator 113, and a common fan 131, positioned in the forced circulation refrigerating circuit 109 without modifications.

[0120] Meanwhile, it is possible to form a suction side space 166 by forming a separator between the equipment case 106 and the cooler case 143 and between an inside of the casing 103 and the cooler case 143. When the separator 165 is not formed, it is possible to locate a plurality of communication equipments 104 having built-in heat components 105 in a casing 103.

EMBODIMENT 11

[0121] In the cooling system 101 for the casing according to Embodiment 11, as illustrated in Figure 30, a control device 138 composed, for example, microcomputers and so on is located. In this embodiment, the control device 138 has a function of a compressor control means 140 to be described below. Further, the cooling system 101 includes a temperature detecting means 134 for detecting an outer air temperature, and a temperature detecting means 135 for detecting a temperature inside the casing, preferably a temperature of airs in the vicinity of the exhaust port 108 of the equipment case 108 and the heat air intake port 132 of the common case 129.

[0122] When the temperature detecting means 134 detects an outer air temperature, and the temperature detecting means 135 detects the temperature inside the casing, the compressor control means 140 operates a difference between the detected temperature inside the casing and outer air temperature. Succeedingly, the compressor control means 140 stops an operation of the compressor 110 in the forced circulation refrigerating circuit 109 based on the obtained temperature difference.

[0123] Specifically, a capability (kW) of the natural circulation refrigerating circuit 120, illustrated in Figure 32, and data about the temperature difference between the temperature inside the casing and the outer air temperature are previously set and memorized in a memory, and a requisite capability of the natural circulation refrigerating circuit 120 is obtained from the temperature difference operated from various detected temperatures. Then, when the requisite capability of the natural circulation refrigerating circuit 120 is lower than the set values of the data under the same conditions, the compressor control means 140 forcibly stops the compressor 110 in the forced circulation refrigerating circuit 109, whereby only the natural circulation refrigerating circuit 120 is continued to operate.

[0124] In other words, in this system according to Embodiment 11, when it is sufficient to cool the inside of the casing using only the capability of the natural circulation refrigerating circuit 120, the compressor 110 is not operated regardless of the air temperature in the position 305, whereby it is possible to avoid a needless running cost for the forced circulation refrigerating circuit 109.

[0125] Meanwhile, the capability of the natural circulation refrigerating circuit 120 is enhanced as the outer air temperature is low, whereby a load to the forced circulation refrigerating circuit 109 can be reduced. For example, when the natural circulation refrigerating circuit 120 having a characteristic as in Figure 32 is designed, a cooling capacity under a temperature difference $\Delta T=25°C$, for example the outdoor air temperature is 15°C and the temperature inside the casing is 40°C, is 40 kW. In other words, if the capability of the natural circulation refrigerating circuit 120 is designed based on conditions of midsummer, when the outdoor air temperature is extremely high, the running capacity on a side of the forced circulation refrigerating circuit 9 in winter seasons and intermediate seasons can be drastically reduced, whereby a running cost is reduced. It is also possible to provide ebullient cooling in the natural circulation refrigerating circuit 120 with a reduction of the capacity on the side of the forced circulation refrigerating circuit 109.

[0126] Needless to say that the structure of Embodiment 11 can be applied to not only the cooling system 101, illustrated in Figure 30, but also the cooling system 101a, illustrated in Figure 31.

[0127] Further, when a calorific load inside the casing scarcely changes through a year, it is possible to stop an operation of the compressor 110 based solely on the detected outer air temperature by the temperature detecting means 134. In such a case, because the temperature detecting means 135 for the casing is omitted, a structure for controlling is simplified and a cost becomes low.

EMBODIMENT 12

[0128] In the cooling system 1 for the casing according to Embodiment 12, as illustrated in Figure 30, a pressure detecting means 136 for detecting a low-pressure refrigerant pressure on a suction side of the compressor 110 and a temperature detecting means 137 for detecting a high pressure refrigerant pressure on an exhaust side of the compressor 110 are included. Further, the control device 138 has functions as a malfunction detecting means 141 for the forced circulation refrigerating circuit 109, detecting based on detected pressures respectively by the pressure detecting means 136, 137 and an operation hold means 142, which holds the common fan 131 in a running state when the malfunction detecting means 141 detects malfunctions of the forced circulation refrigerating circuit 109.

[0129] Accordingly, in the cooling system according to Embodiment 12, when a difference between the detected pressure, respectively from the pressure detecting means 136, 137, is larger than a predetermined value, the malfunction detecting means 141 detects malfunctions of the forced circulation refrigerating circuit 109. The malfunction detecting means 141 urgently stops the compressor 110. Simultaneously, the malfunction detecting means 141 continues to send an air by holding the common fan 131 in the running state. Since the natural circulation refrigerating circuit 120 is constantly operated even in such a case, an air is sent to the boiling pipe cooler 121 by the common fan 131. In other words, even though the forced circulation refrigerating circuit 109 is urgently stopped by malfunctions, an air cooled by the boiling type cooler 121 is blown into the casing 103, and a temperature inside a casing 103 is not boiled. However, it is possible to detect a refrigerant temperature on the exhaust side of the compressor instead of using the pressure detecting means 136, 137 for detecting malfunctions of the forced circulation refrigerating circuit 109. Such modification may be applied to the malfunction detecting means other than the pressure detecting means.

[0130] Embodiment 12 can be applied to not only the cooling system 101 but also the cooling system 101a in Figure 31.

[0131] Although in the above embodiments, examples that the common airflow paths 130, 130a are built in the casing 103, the present invention is not limited to such structures. For example, the common case 129 for the common airflow paths 130, 130a, the cooler case 143, the connecting duct 144, the evaporator case 145 may be attached to an outside of the casing for connecting the heat air intake port 132 and the cool air exhaust port 133 into an inside of the casing 103.

[0132] Further, although the common fan 131 is located between the boiling type cooler 121 and the evaporator 113, the present invention is not limited to this structure. The common fan 131 may be located on an upstream side of the airflow in the boiling type cooler 121 or on the downstream side of the airflow in the evaporator 113 in the common airflow paths 130, 130a.

[0133] The first advantage of the cooling control method for a communication station as described above is that the temperature of the suction air for cooling the communication equipments is rendered to be the set temperature based on the detected value by the temperature detecting means; the temperature of the suction air for the communication equipments is stabilized; and it is possible to accurately follow up the calorific value, i.e. load, caused by the operation of the communication equipments.

[0134] The second advantage of the cooling control method for the communication station as described above is that it is possible to control with a good capability of following up the load and to efficiently control cooling in response to the load of the air conditioner, whereby the energy can be saved during the operation.

[0135] The third advantage of the cooling control method for communication stations as described above is that frequent thermo-ons and thermo-offs are prevented to avoid the moisture condensation in the casing.

[0136] The fourth advantage of the cooling control method for communication stations as described above is that the electric power consumption can be detected using an ampere meter at a cost lower than that at an electric power meter.

[0137] The fifth advantage of the cooling control method for communication stations as described above is that the moisture condensation can be prevented because a blowing-out air temperature from the air conditioner is increased.

[0138] The sixth advantage of the cooling control method for communication stations as described above is that a lifetime of the air conditioner is not adversely affected.

[0139] The seventh advantage of the cooling control method for communication stations as described above is that it is possible to prevent troubles such that a usable temperature range for the air conditioner is outstripped.

[0140] The eighth advantage of the cooling control method for communication stations as described above is that preferable control target values can be set regardless of hours and changes of the seasons, and a stable operation is obtainable.

[0141] The ninth advantage of the cooling control method for communication stations as described above is that it is possible to deal with troubles of the cooling devices because the auxiliary cooling device and the main cooling device are independently controlled to operate.

[0142] The tenth advantage of the cooling control method for communication stations as described above is that it is possible to set the air temperature, introduced into the evaporator of the boiling type cooling device, high, enhance the cooling capability of the boiling type cooling device, and increase the cooling air temperature inside the main cooling device, whereby the moisture condensation in the casing can be prevented.

[0143] The eleventh advantage of the cooling control method for communication stations as described above is that the energy can be saved while maintaining the cooling effect inside the casing.

[0144] The twelfth advantage of the cooling control method for communication stations as described above is that the capability of the natural circulation refrigerating circuit can be expansively used by adopting a fan providing a high airflow path as the common fan.

[0145] The thirteenth advantage of the cooling control method for communication stations as described above is that bypassing and short cycles do not occur in the casing, a high cooling efficiency is realized, and therefore the energy can be saved.

[0146] The fourteenth advantage of the cooling control method for communication stations as described above is that troubles in the forced circulation refrigerating circuit, caused at time of directly sucking a heat air, can be prevented.

[0147] The fifteenth advantage of the cooling control method for communication stations as described above is that the air conditioner indoor unit for the forced circulation refrigerating circuit can be used without modification when the common airflow path is formed and the common fan is located in the airflow path on the evaporator side.

[0148] The sixteenth advantage of the cooling control method for communication stations as described above is that the cooler of the natural circulation refrigerating circuit on the use side can be used without modification when the common fan is located in the airflow path on the cooler side.

[0149] The seventeenth advantage of the cooling control method for communication stations as described above is that an unnecessary running cost for the forced circulation refrigerating circuit can be saved.

[0150] The eighteenth advantage of the cooling control method for communication stations as described above is that the troubles in the forced circulation refrigerating circuit can be rapidly solved without an abrupt increment of the temperature in the casing.

[0151] Obviously, numerous modifications and variations of the present invention are possible in light of the above teachings. It is therefore to be understood that within the scope of the appended claims, the invention may be practiced otherwise than as specifically described herein.

[0152] The entire disclosure of Japanese Patent Applications JP11-219661 filed on August 3, 1999, JP11-239406 filed on August 26, 1999 and JP12-014583 filed on January 24, 2000 including specification, claims, drawings and summary are incorporated herein by reference in its entirety.

Claims

1. A cooling system (101) for a communication station for cooling a casing (103) of the communication station, accommodating communication equipment including heat components, by a boiling-type cooler (121) in a natural circulation refrigerating circuit and an evaporator (113) in a forced circulation refrigerating circuit (109), activated by a compressor (110), the cooling system comprising:

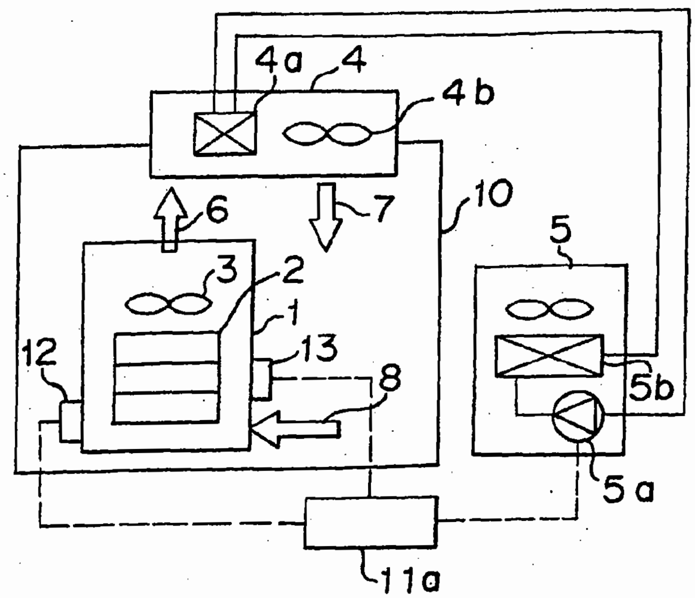
   a common airflow path (130) having a heated air intake port (132) for taking heated air into the casing (103) and a cooled air exhaust port (133) for blowing cooled air into the casing (103);
   a common fan (131) for sending air to the boiling-type cooler (121) and the evaporator (113);
   the boiling-type cooler (121), the evaporator (113), and the common fan (131) being built in the common airflow path (130);
   temperature detecting means (134) for detecting at least an outer air temperature; and
   compressor control means (140) for stopping the operation of the compressor (110) in the forced circulation refrigerating circuit (109) based on a detected temperature received from the temperature detecting means (134).

2. A cooling system according to claim 1, wherein the common airflow path (130a) comprises an air path (148) on a cooler side, inside which the boiling-type cooler (121) is installed, an air path (150) on an evaporator side, in which the evaporator (113) is installed, and a connection air path (149) for connecting the air path (148) on the cooler side and the air path (150) on the evaporator side.

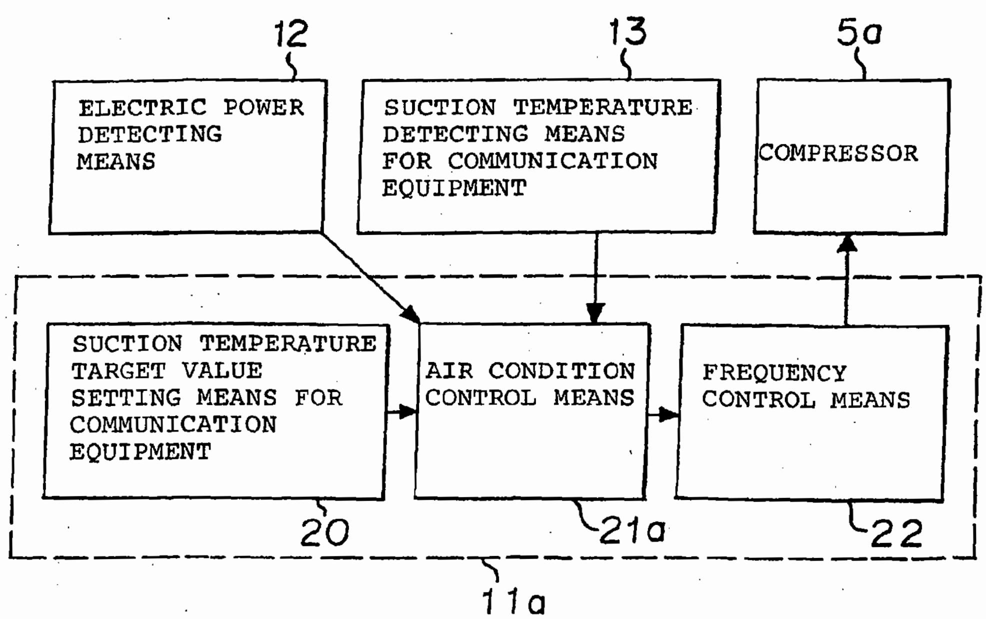
3. A cooling system according to claims 1 or 2, further comprising:

   malfunction detecting means (141) for detecting malfunctions of the forced circulation refrigerating circuit (109); and
   driving means (142) for keeping the common fan (131) in a running state when the malfunctions in the forced circulation refrigerating circuit (109) are detected by the malfunction detecting means (141).

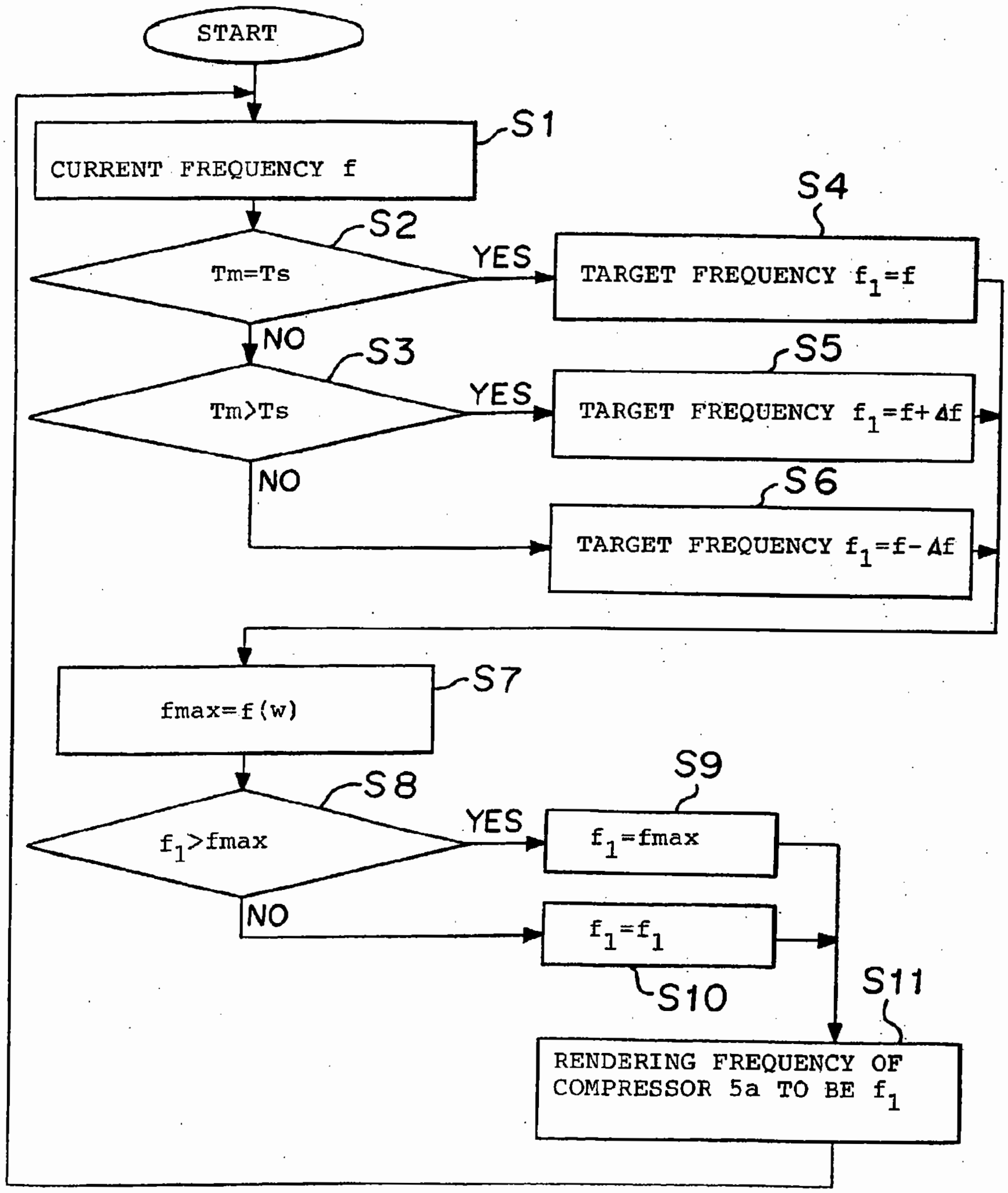
F I G. 1


4a
4
4b
6
7
10
5
3
2
1
12
13
8
5b
5a
11a

F I G. 2

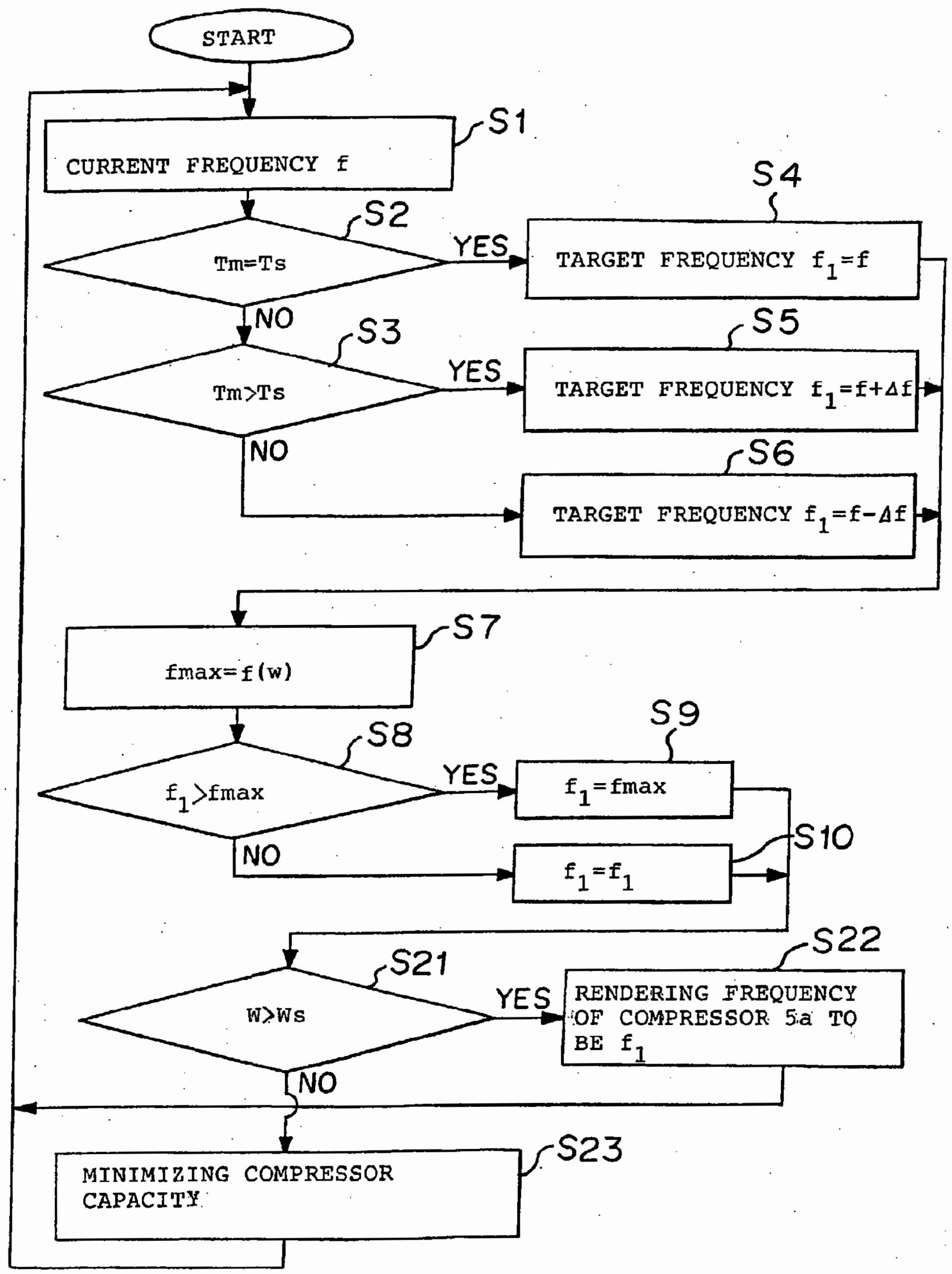

12
13
5a
ELECTRIC POWER DETECTING MEANS
SUCTION TEMPERATURE DETECTING MEANS FOR COMMUNICATION EQUIPMENT
COMPRESSOR
SUCTION TEMPERATURE TARGET VALUE SETTING MEANS FOR COMMUNICATION EQUIPMENT
AIR CONDITION CONTROL MEANS
FREQUENCY CONTROL MEANS
20
21a
22
11a

F I G. 3



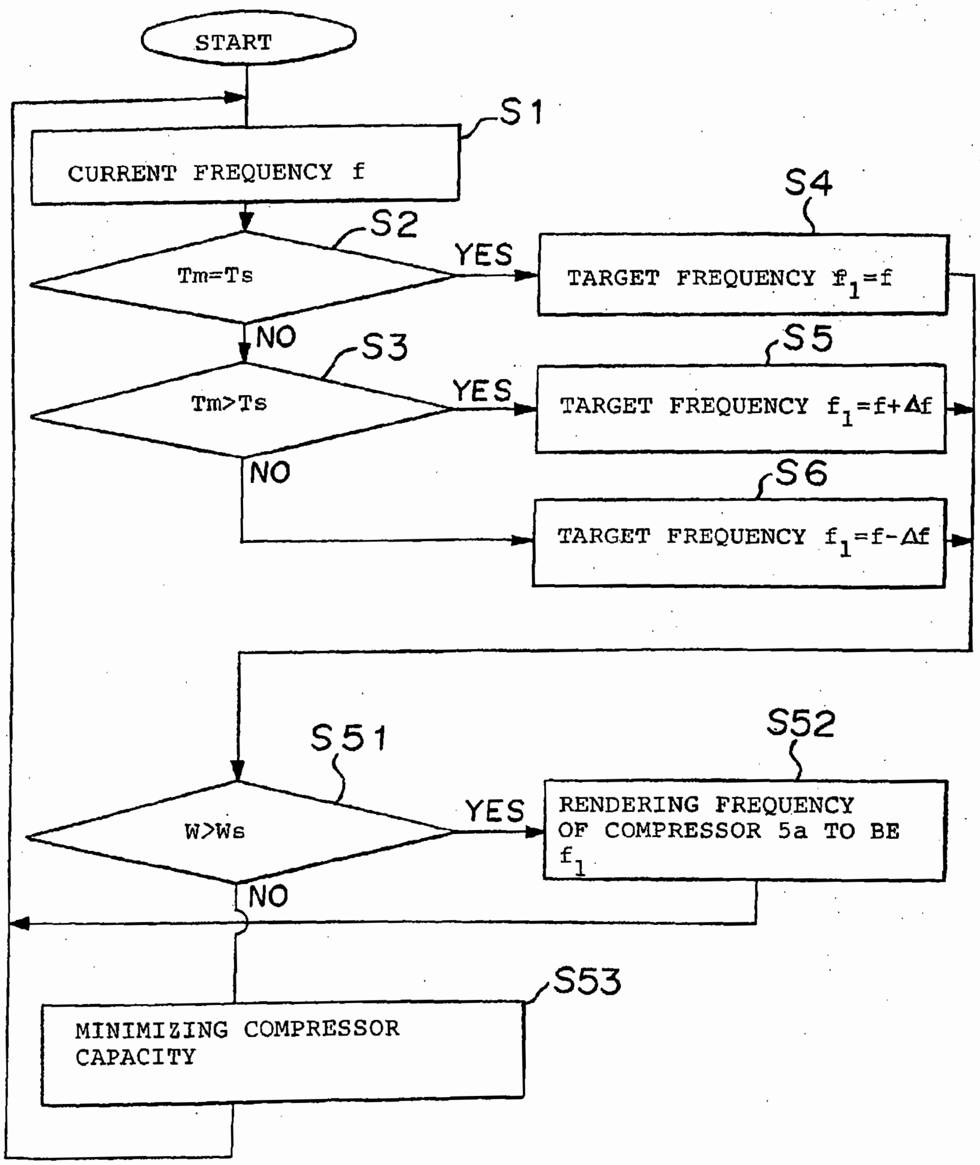
Tm: TEMPERATURE OF SUCTION AIR 8 FOR COMMUNICATION EQUIPMENT
Ts: SET TEMPERATURE OF SUCTION AIR 8 FOR COMMUNICATION EQUIPMENT
Δf: PREVIOUSLY SET VARIATION OF FREQUENCY
w: OUTPUT FROM ELECTRIC POWER DETECTING MEANS 12
$f_{(w)}$: MAXIMUM FREQUENCY OPERATING FUNCTION HAVING VARIABLE w

FIG. 4



Tm: TEMPERATURE OF SUCTION AIR 8 FOR COMMUNICATION EQUIPMENT
Ts: SET TEMPERATURE OF SUCTION AIR 8 FOR COMMUNICATION EQUIPMENT
Δf: PREVIOUSLY SET VARIATION OF FREQUENCY
w: OUTPUT FROM ELECTRIC POWER DETECTING MEANS 12
f(w): MAXIMUM FREQUENCY OPERATING FUNCTION HAVING VARIABLE w
Ws: PREVIOUSLY SET VALUE OF ELECTRIC POWER CONSUMPTION

F I G. 5



Tm: TEMPERATURE OF SUCTION AIR 8 FOR COMMUNICATION EQUIPMENT
Ts: SET TEMPERATURE OF SUCTION AIR 8 FOR COMMUNICATION EQUIPMENT
Δf: PREVIOUSLY SET VARIATION OF FREQUENCY
W: OUTPUT FROM ELECTRIC POWER DETECTING MEANS 12

F I G. 6

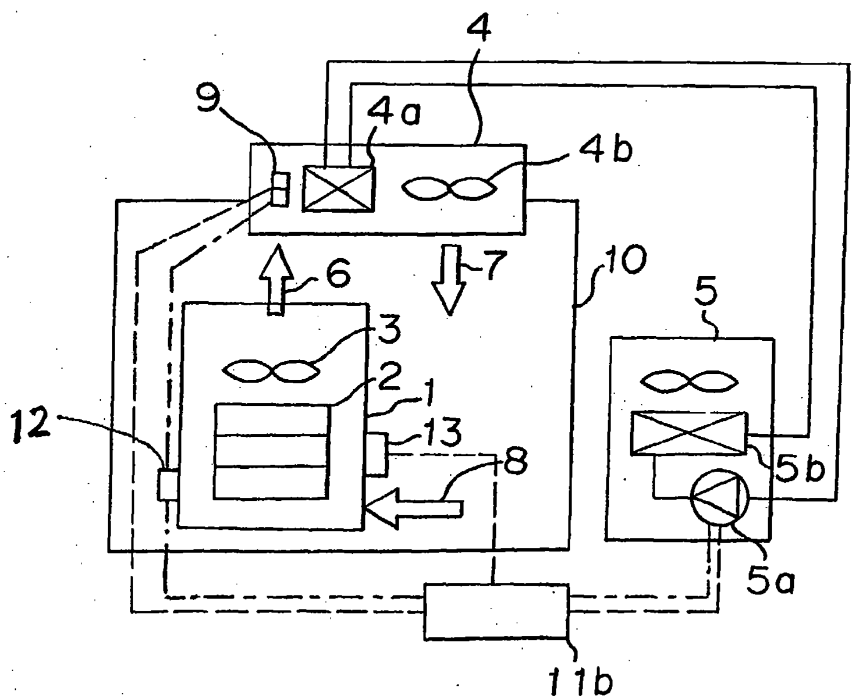


FIG. 7

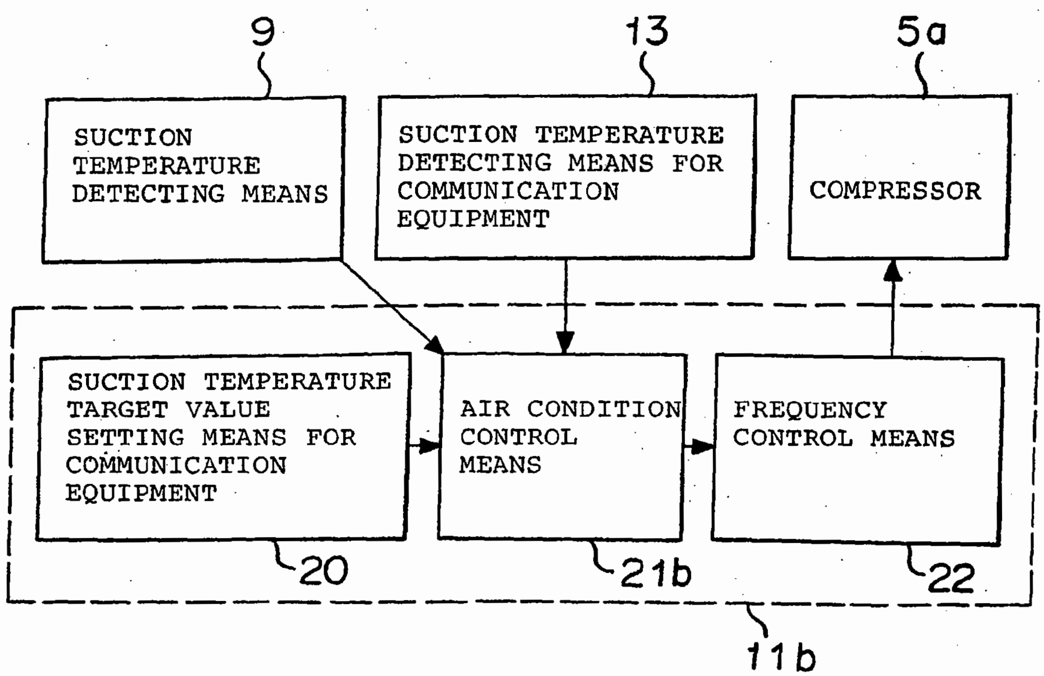
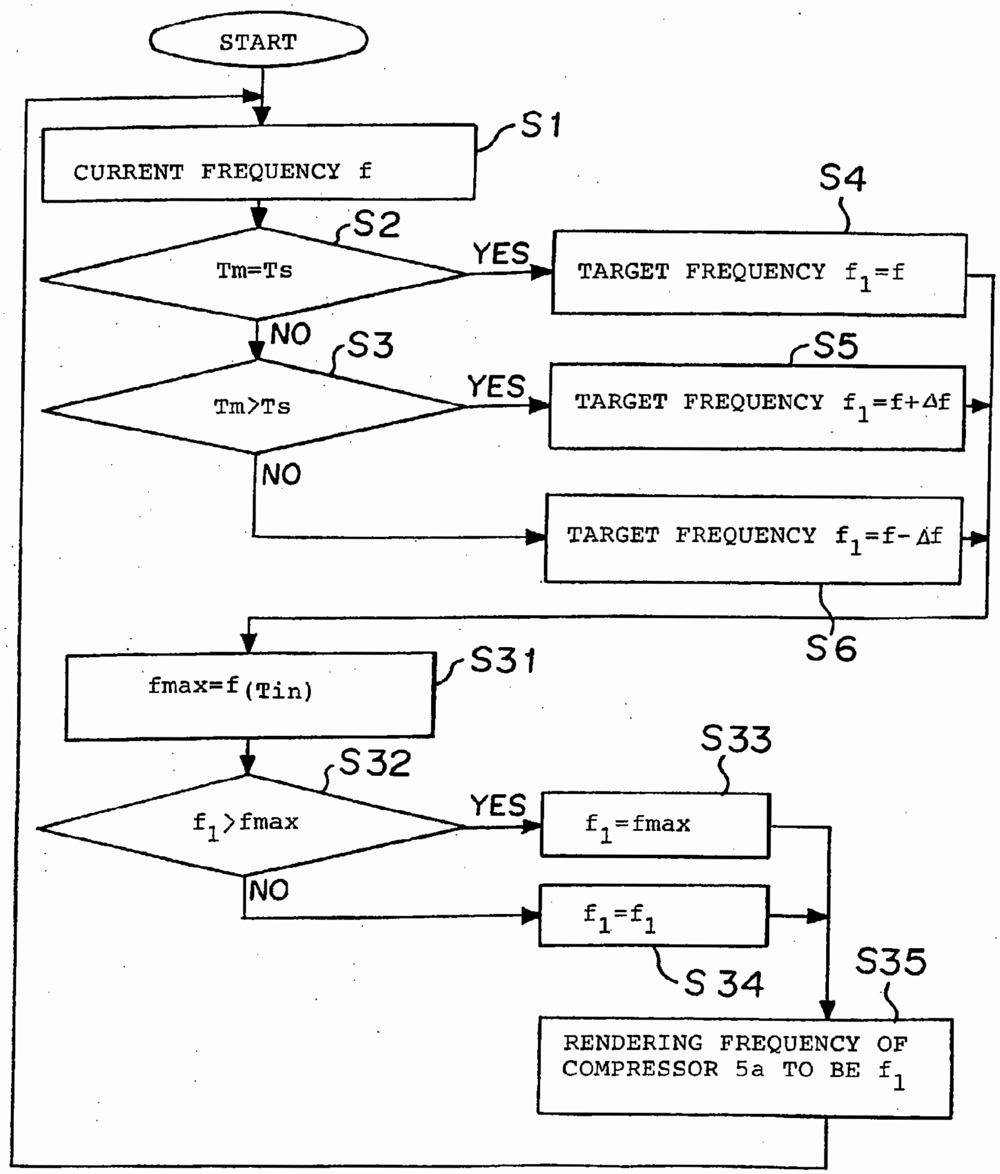
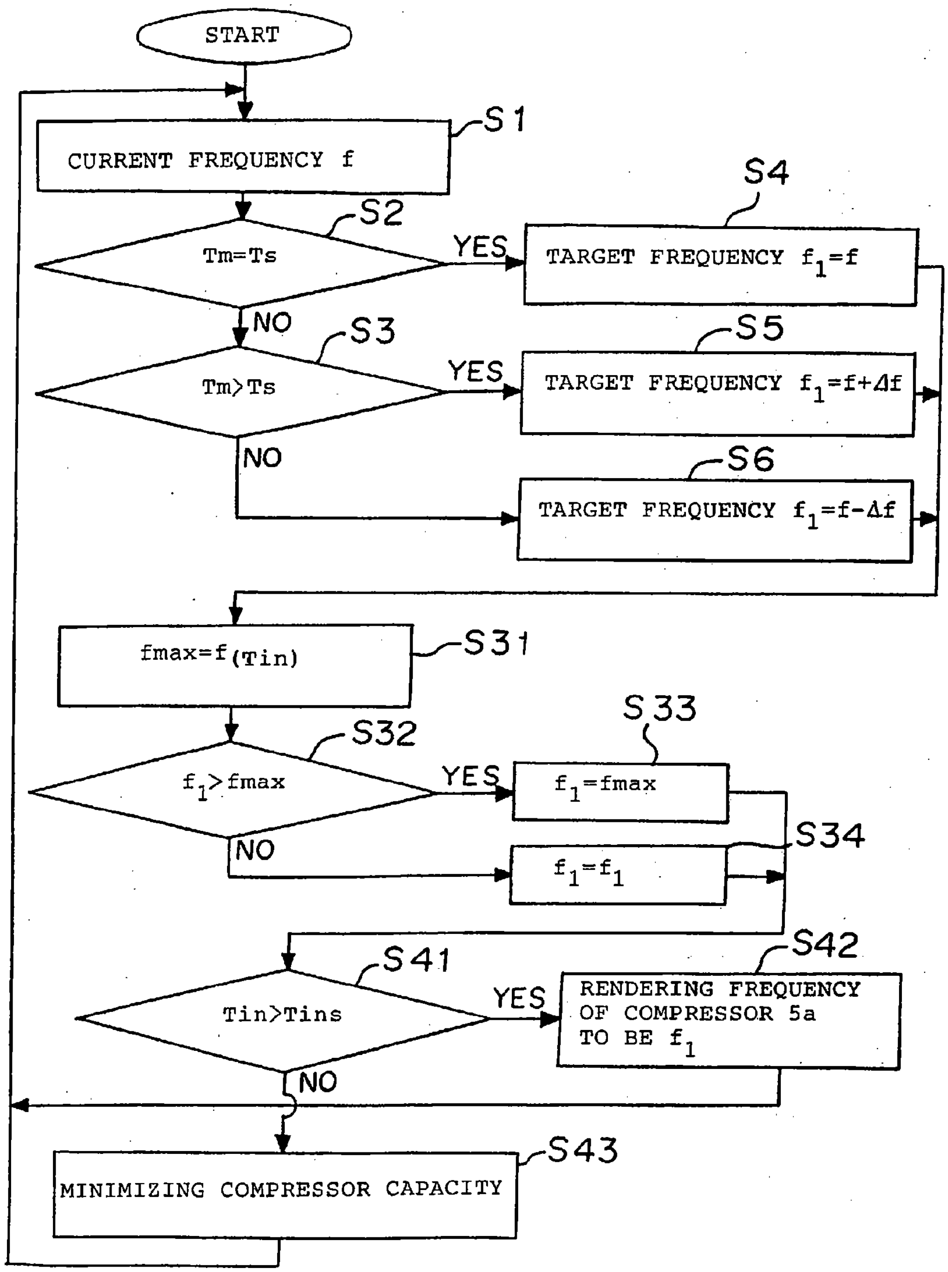

9
13
5a
SUCTION TEMPERATURE DETECTING MEANS
SUCTION TEMPERATURE DETECTING MEANS FOR COMMUNICATION EQUIPMENT
COMPRESSOR
SUCTION TEMPERATURE TARGET VALUE SETTING MEANS FOR COMMUNICATION EQUIPMENT
AIR CONDITION CONTROL MEANS
FREQUENCY CONTROL MEANS
20
21b
22
11b

FIG. 8



Tm: TEMPERATURE OF SUCTION AIR 8 FOR COMMUNICATION EQUIPMENT
Ts: SET TEMPERATURE OF SUCTION AIR 8 FOR COMMUNICATION EQUIPMENT
Δf: PREVIOUSLY SET VARIATION OF FREQUENCY
Tin: SUCTION TEMPERATURE
f(Tin): MAXIMUM FREQUENCY OPERATING FUNCTION HAVING VARIABLE Tin

F I G. 9



Tm: TEMPERATURE OF SUCTION AIR 8 FOR COMMUNICATION EQUIPMENT
Ts: SET TEMPERATURE OF SUCTION AIR 8 FOR COMMUNICATION EQUIPMENT
Δf: PREVIOUSLY SET VARIATION OF FREQUENCY
f(Tin): MAXIMUM FREQUENCY OPERATING FUNCTION HAVING VARIABLE Tin
Tin: SUCTION TEMPERATURE
Tins: SET VALUE OF SUCTION TEMPERATURE

FIG. 10

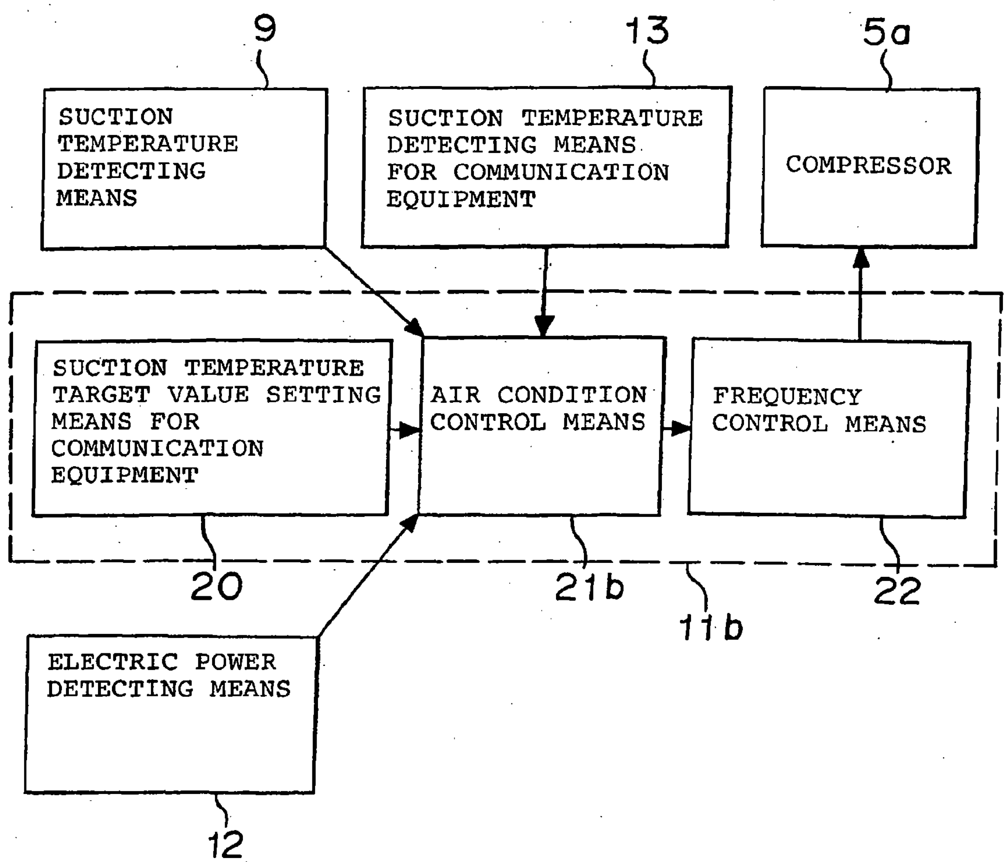
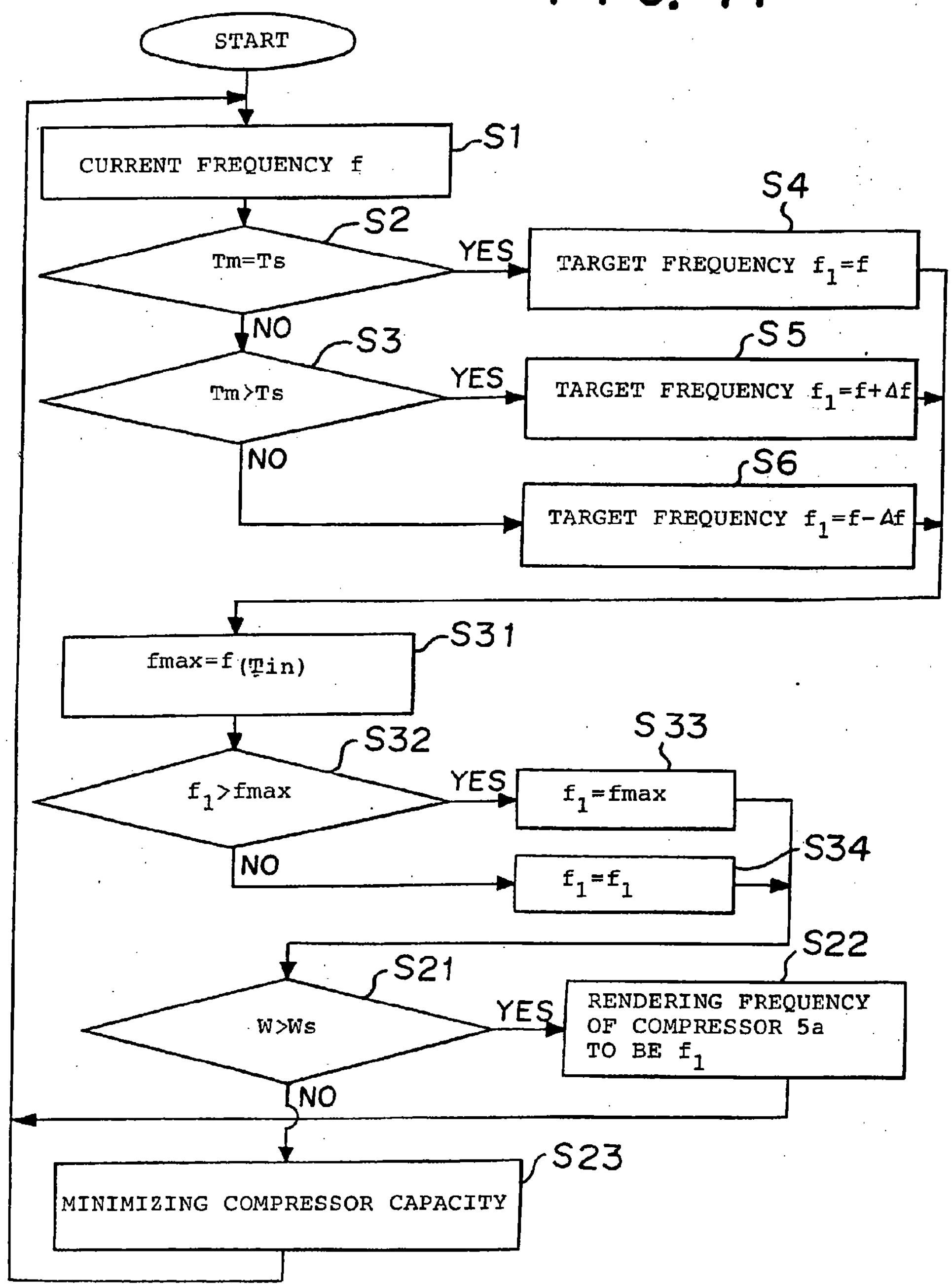
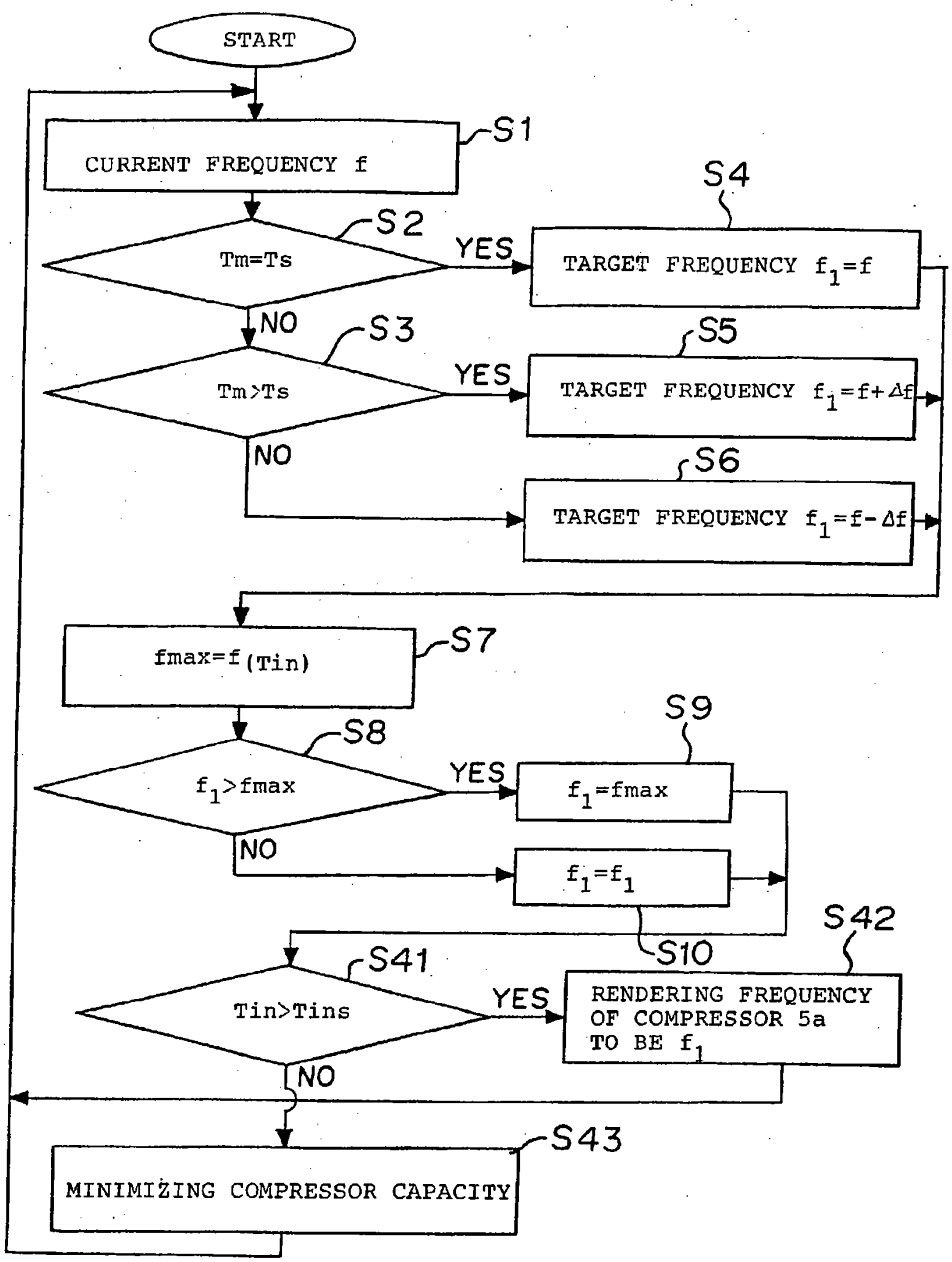

9
SUCTION TEMPERATURE DETECTING MEANS
13
SUCTION TEMPERATURE DETECTING MEANS FOR COMMUNICATION EQUIPMENT
5a
COMPRESSOR
SUCTION TEMPERATURE TARGET VALUE SETTING MEANS FOR COMMUNICATION EQUIPMENT
AIR CONDITION CONTROL MEANS
FREQUENCY CONTROL MEANS
20
21b
11b
22
ELECTRIC POWER DETECTING MEANS
12

FIG. 11



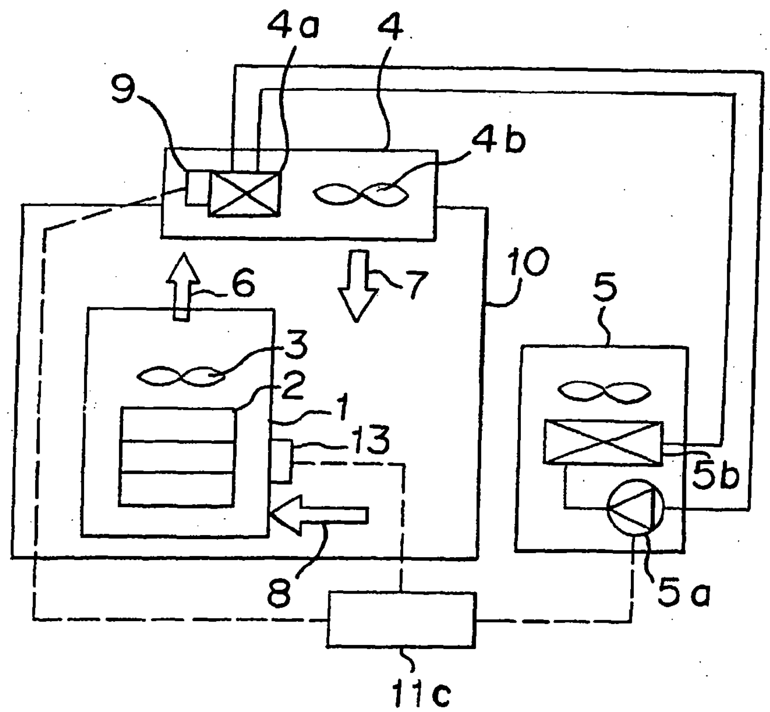
Tm: TEMPERATURE OF SUCTION AIR 8 FOR COMMUNICATION EQUIPMENT
Ts: SET TEMPERATURE OF SUCTION AIR 8 FOR COMMUNICATION EQUIPMENT
Δf: PREVIOUSLY SET VARIATION OF FREQUENCY
W: OUTPUT FROM ELECTRIC POWER DETECTING MEANS 12
Ws: PREVIOUSLY SET VALUE OF ELECTRIC POWER CONSUMPTION

FIG. 12



Tm: TEMPERATURE OF SUCTION AIR 8 FOR COMMUNICATION EQUIPMENT
Ts: SET TEMPERATURE OF SUCTION AIR 8 FOR COMMUNICATION EQUIPMENT
Δf: PREVIOUSLY SET VARIATION OF FREQUENCY
w: OUTPUT FROM ELECTRIC POWER DETECTING MEANS 12
f(w): MAXIMUM FREQUENCY OPERATING FUNCTION HAVING VARIABLE w
Tin: SUCTION TEMPERATURE
Tins: SET VALUE OF SUCTION TEMPERATURE

F I G. 13



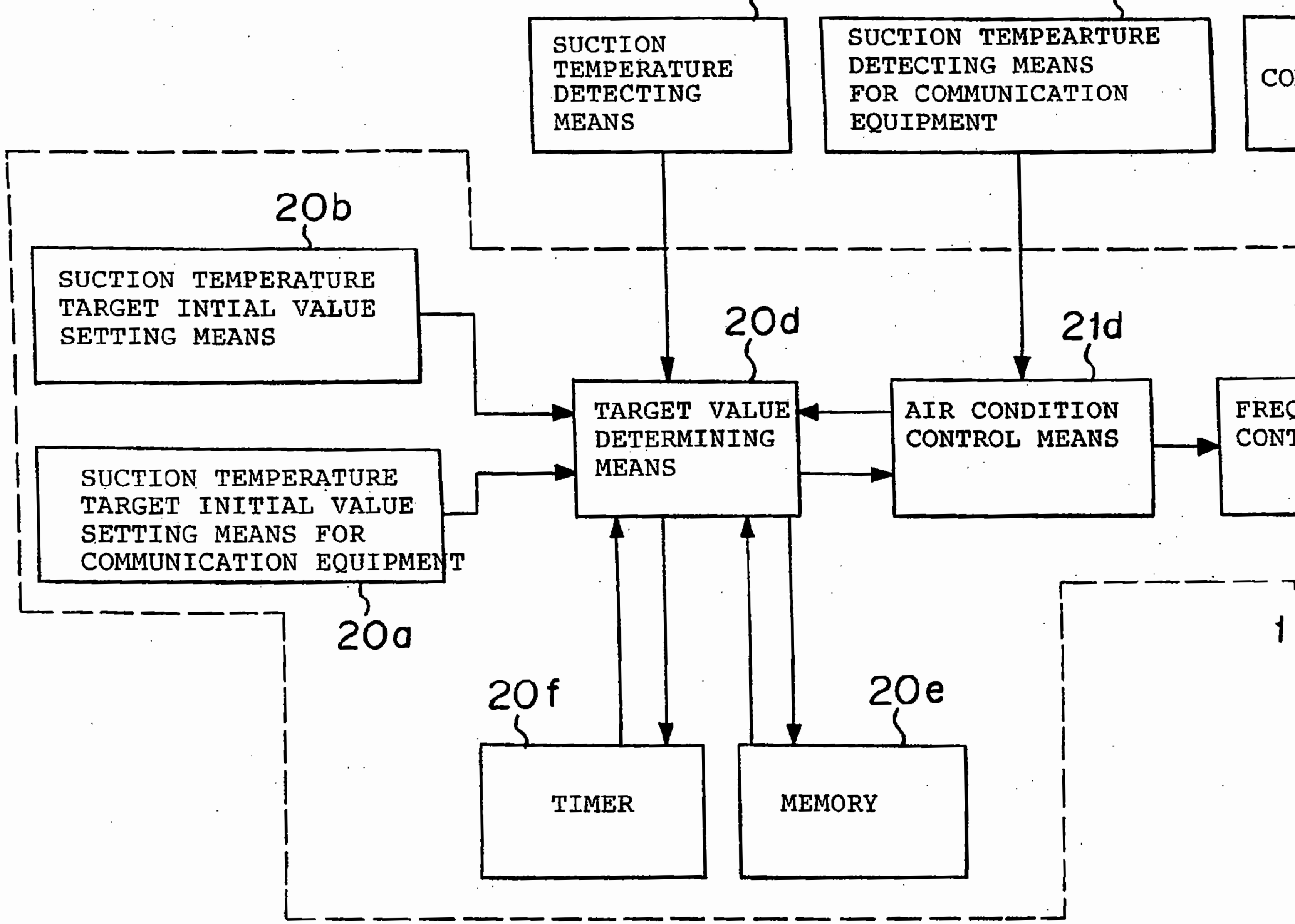
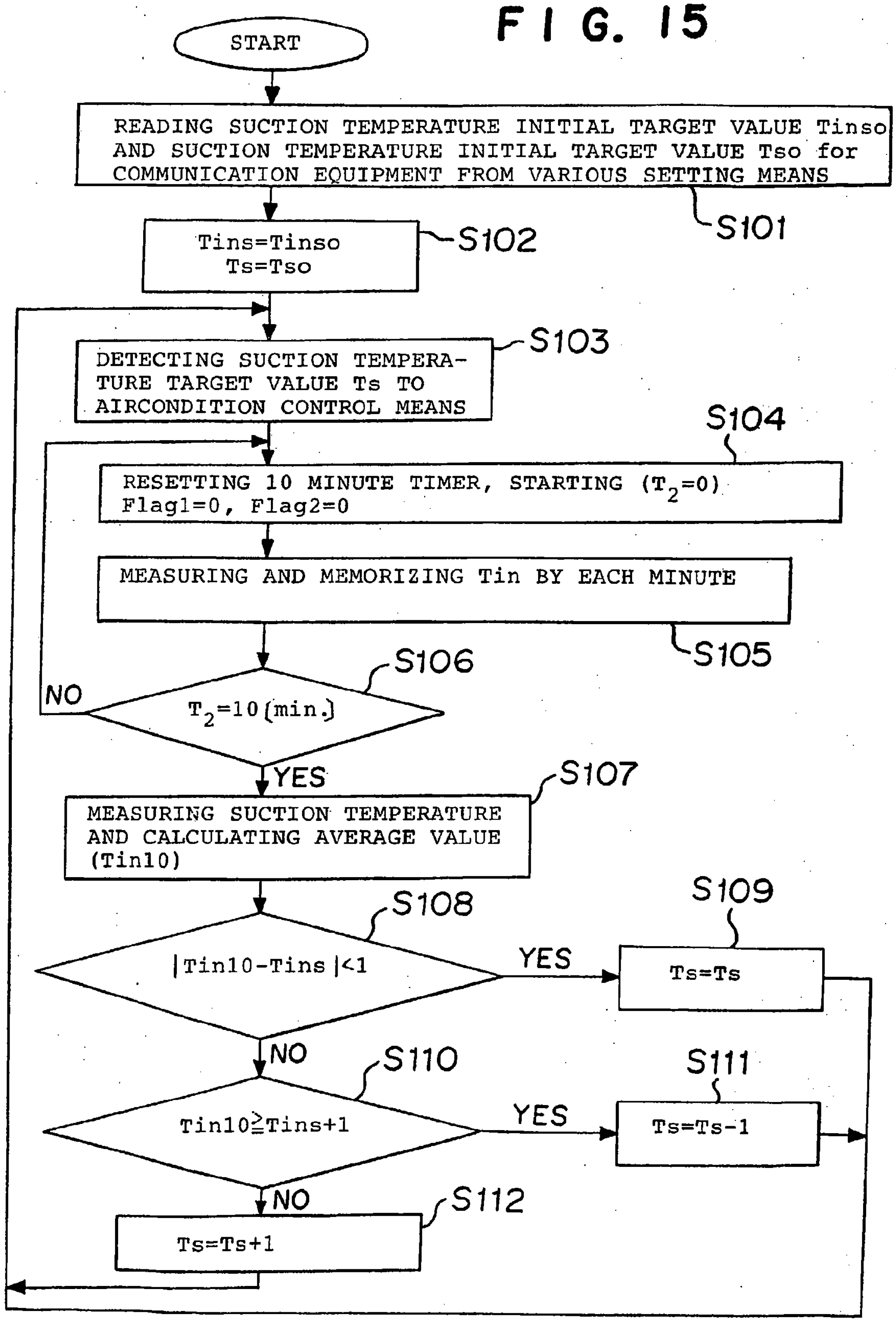
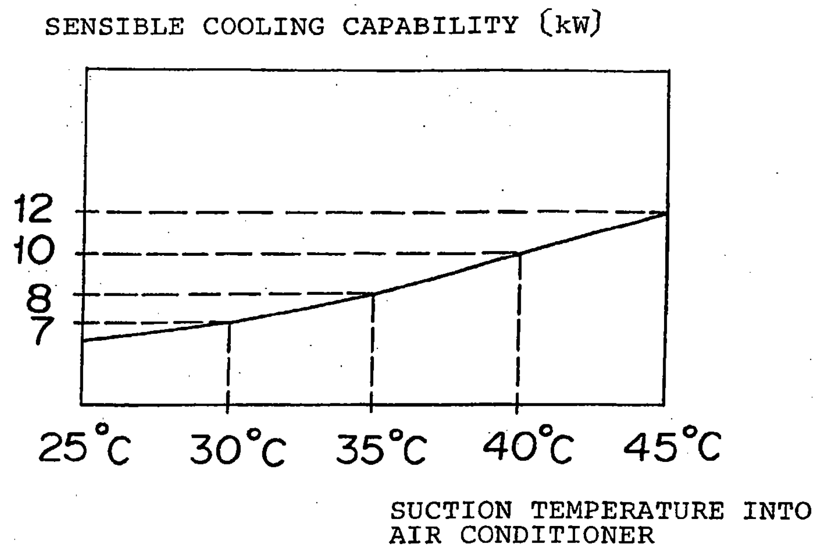

SUCTION TEMPERATURE DETECTING MEANS
SUCTION TEMPEARTURE DETECTING MEANS FOR COMMUNICATION EQUIPMENT
CO
20b
SUCTION TEMPERATURE TARGET INTIAL VALUE SETTING MEANS
20d
21d
TARGET VALUE DETERMINING MEANS
AIR CONDITION CONTROL MEANS
FREQ
CONT
SUCTION TEMPERATURE TARGET INITIAL VALUE SETTING MEANS FOR COMMUNICATION EQUIPMENT
20a
1
20f
20e
TIMER
MEMORY

FIG. 15


START
READING SUCTION TEMPERATURE INITIAL TARGET VALUE Tinso AND SUCTION TEMPERATURE INITIAL TARGET VALUE Tso for COMMUNICATION EQUIPMENT FROM VARIOUS SETTING MEANS
S101
Tins=Tinso
Ts=Tso
S102
DETECTING SUCTION TEMPERATURE TARGET VALUE Ts TO AIRCONDITION CONTROL MEANS
S103
RESETTING 10 MINUTE TIMER, STARTING (T_2=0)
Flag1=0, Flag2=0
S104
MEASURING AND MEMORIZING Tin BY EACH MINUTE
S105
T_2=10 (min.)
S106
NO
YES
MEASURING SUCTION TEMPERATURE AND CALCULATING AVERAGE VALUE (Tin10)
S107
|Tin10-Tins|<1
S108
YES
Ts=Ts
S109
NO
Tin10≧Tins+1
S110
YES
Ts=Ts-1
S111
NO
Ts=Ts+1
S112

F I G. 16


SENSIBLE COOLING CAPABILITY (kW)
12
10
8
7
25°C
30°C
35°C
40°C
45°C
SUCTION TEMPERATURE INTO AIR CONDITIONER

F I G. 17

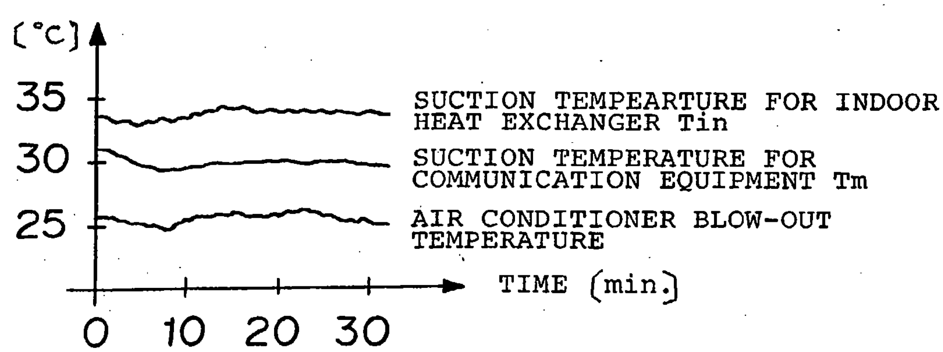

(°C)
35
30
25
SUCTION TEMPEARTURE FOR INDOOR HEAT EXCHANGER Tin
SUCTION TEMPERATURE FOR COMMUNICATION EQUIPMENT Tm
AIR CONDITIONER BLOW-OUT TEMPERATURE
TIME (min.)
0
10
20
30

FIG. 18

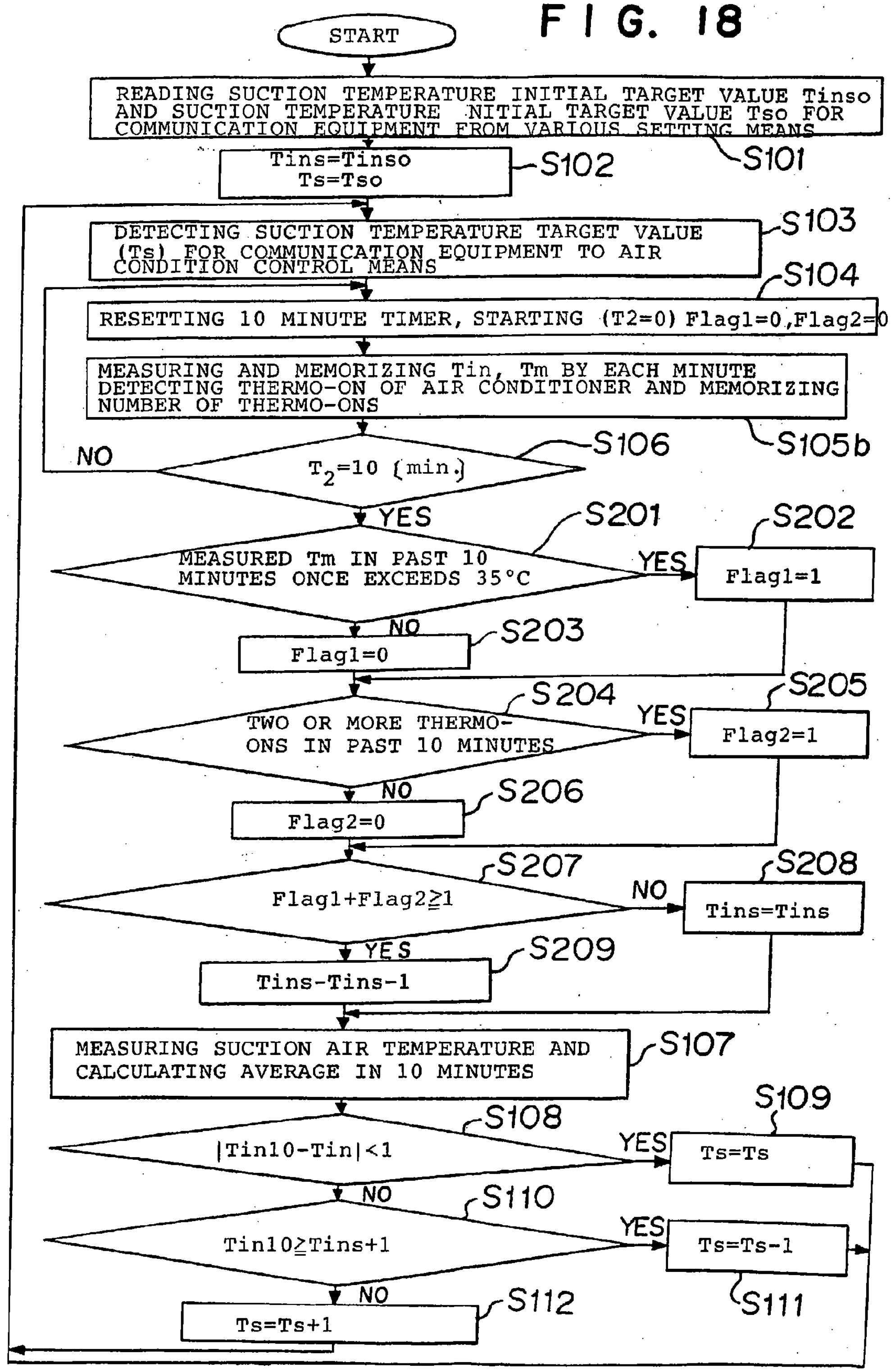
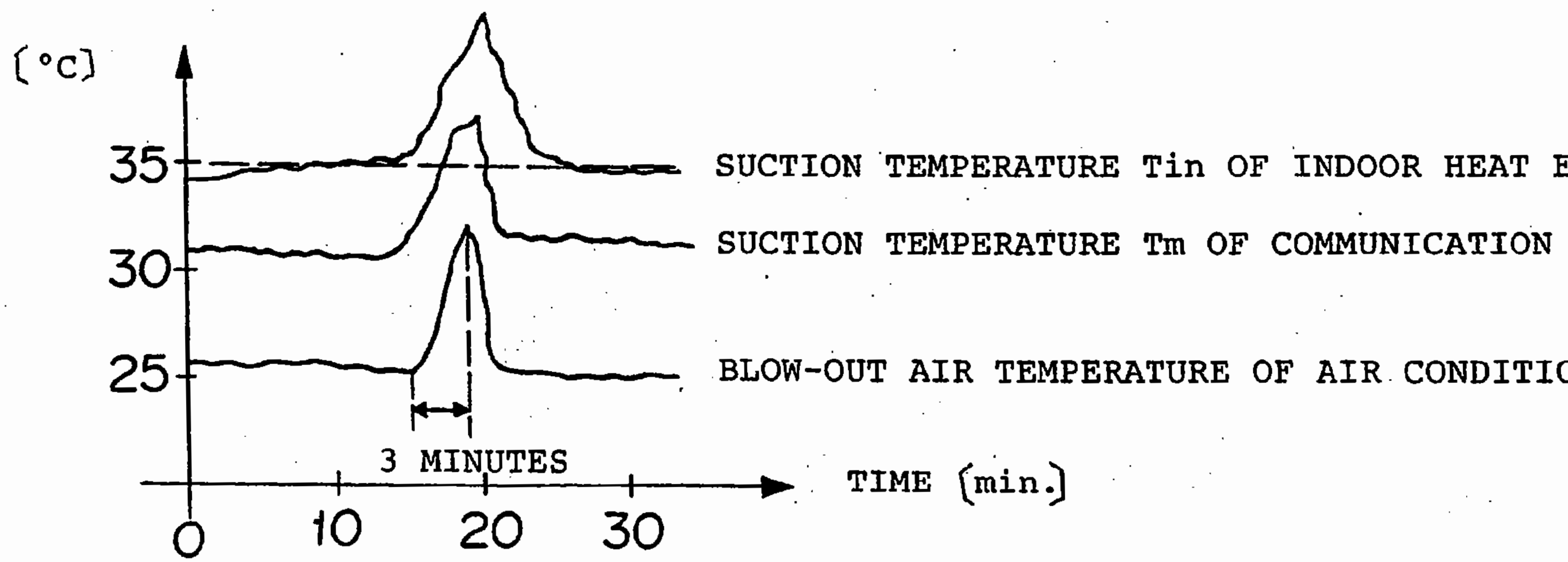

START
READING SUCTION TEMPERATURE INITIAL TARGET VALUE Tinso AND SUCTION TEMPERATURE INITIAL TARGET VALUE Tso FOR COMMUNICATION EQUIPMENT FROM VARIOUS SETTING MEANS
S101
Tins=Tinso
Ts=Tso
S102
DETECTING SUCTION TEMPERATURE TARGET VALUE (Ts) FOR COMMUNICATION EQUIPMENT TO AIR CONDITION CONTROL MEANS
S103
RESETTING 10 MINUTE TIMER, STARTING (T2=0) Flag1=0,Flag2=0
S104
MEASURING AND MEMORIZING Tin, Tm BY EACH MINUTE DETECTING THERMO-ON OF AIR CONDITIONER AND MEMORIZING NUMBER OF THERMO-ONS
S105b
T2=10 (min.)
S106
NO
YES
MEASURED Tm IN PAST 10 MINUTES ONCE EXCEEDS 35°C
S201
YES
Flag1=1
S202
NO
Flag1=0
S203
TWO OR MORE THERMO-ONS IN PAST 10 MINUTES
S204
YES
Flag2=1
S205
NO
Flag2=0
S206
Flag1+Flag2≧1
S207
NO
Tins=Tins
S208
YES
Tins-Tins-1
S209
MEASURING SUCTION AIR TEMPERATURE AND CALCULATING AVERAGE IN 10 MINUTES
S107
|Tin10-Tin|<1
S108
YES
Ts=Ts
S109
NO
Tin10≧Tins+1
S110
YES
Ts=Ts-1
S111
NO
Ts=Ts+1
S112

FIG. 19


[°C]
35
30
25
SUCTION TEMPERATURE Tin OF INDOOR HEAT EX
SUCTION TEMPERATURE Tm OF COMMUNICATION I
BLOW-OUT AIR TEMPERATURE OF AIR CONDITIO
3 MINUTES
0
10
20
30
TIME [min.]

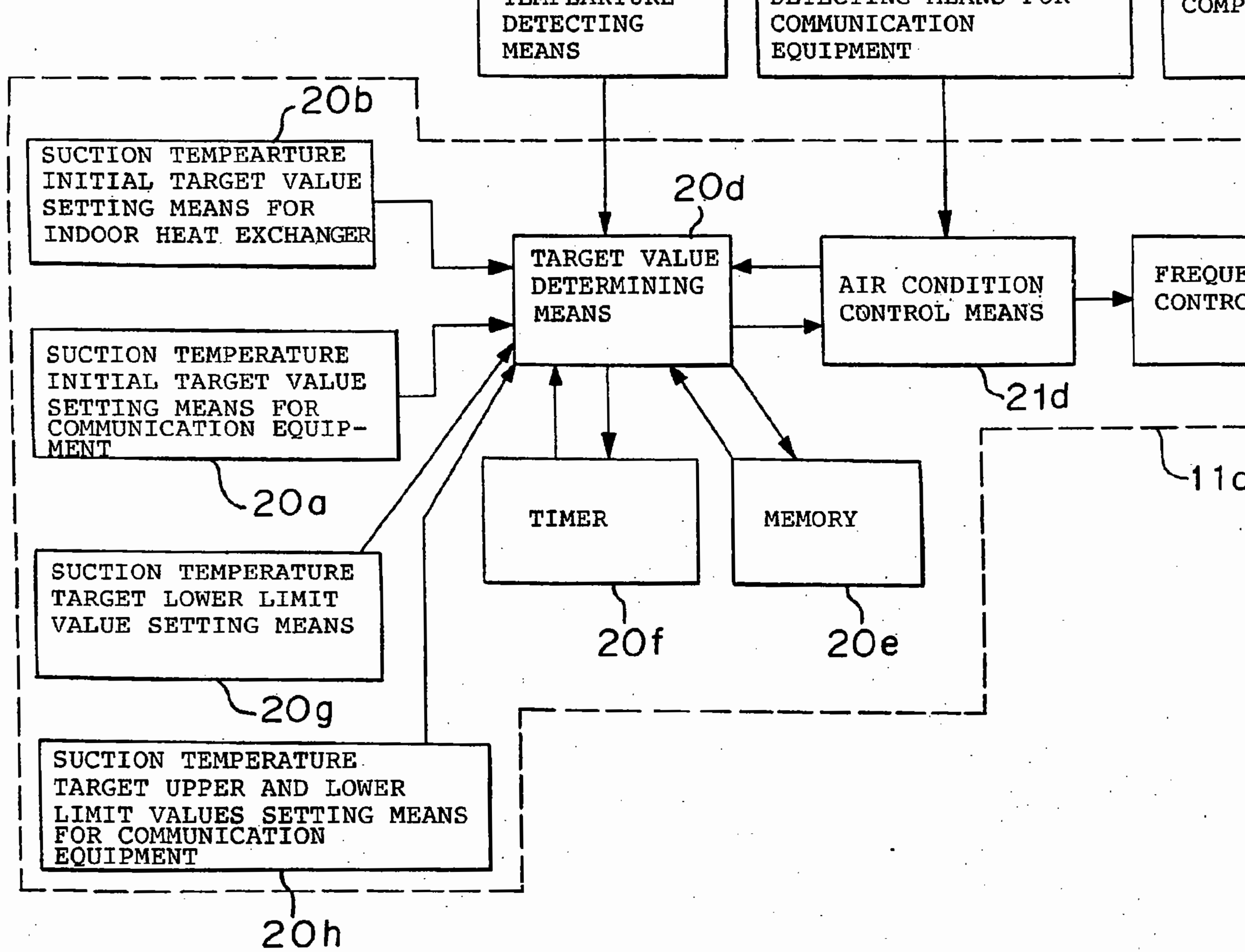
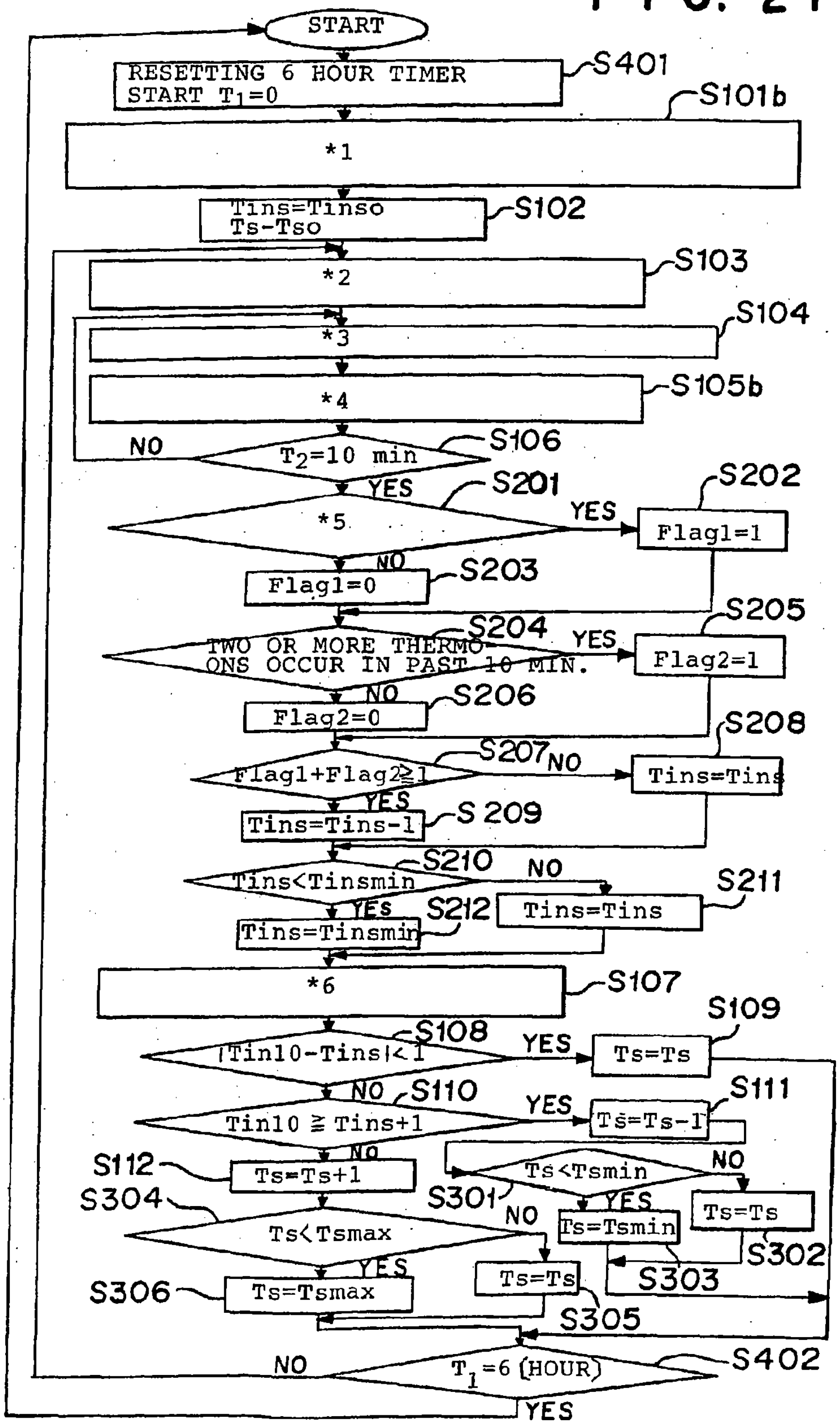

TEMPERATURE DETECTING MEANS
DETECTING MEANS FOR COMMUNICATION EQUIPMENT
COMP
20b
SUCTION TEMPEARTURE INITIAL TARGET VALUE SETTING MEANS FOR INDOOR HEAT EXCHANGER
20d
TARGET VALUE DETERMINING MEANS
AIR CONDITION CONTROL MEANS
FREQUE CONTRO
SUCTION TEMPERATURE INITIAL TARGET VALUE SETTING MEANS FOR COMMUNICATION EQUIP-MENT
21d
20a
TIMER
MEMORY
11c
SUCTION TEMPERATURE TARGET LOWER LIMIT VALUE SETTING MEANS
20f
20e
20g
SUCTION TEMPERATURE TARGET UPPER AND LOWER LIMIT VALUES SETTING MEANS FOR COMMUNICATION EQUIPMENT
20h

FIG. 21


START
RESETTING 6 HOUR TIMER
START T1=0
S401
S101b
*1
Tins=Tinso
Ts-Tso
S102
*2
S103
*3
S104
*4
S105b
T2=10 min
S106
NO
YES
*5
S201
YES
Flag1=1
S202
NO
Flag1=0
S203
TWO OR MORE THERMO-ONS OCCUR IN PAST 10 MIN.
S204
YES
Flag2=1
S205
NO
Flag2=0
S206
Flag1+Flag2≧1
S207
NO
Tins=Tins
S208
YES
Tins=Tins-1
S 209
Tins<Tinsmin
S210
NO
Tins=Tins
S211
YES
Tins=Tinsmin
S212
*6
S107
|Tin10-Tins|<1
S108
YES
Ts=Ts
S109
NO
Tin10 ≧ Tins+1
S110
YES
Ts=Ts-1
S111
NO
Ts=Ts+1
S112
Ts<Tsmin
NO
S301
YES
Ts=Tsmin
Ts=Ts
S302
S303
S304
Ts<Tsmax
NO
YES
Ts=Tsmax
S306
Ts=Ts
S305
T1=6 (HOUR)
S402
NO
YES

F I G. 21 (CONTINUED)

*1 SUCTION TEMPERATURE INITIAL TARGET VALUE Tinso
READING SUCTION TEMPERATURE INITIAL TARGET VALUE Tso FOR COMMUNICATION EQUIPMENT FROM VARIOUS SETTING MEANS
LOWER LIMIT VALUE OF SUCTION TEMPERATURE TARGET VALUE: Tinsmin
LOWER LIMIT VALUE OF SUCTION TEMPERATURE TARGET VALUE FOR COMMUNICATION EQUIPMENT: Tsmin
UPPER LIMIT VALUE: Tsmax

*2 DIRECTING SUCTION TEMPERATURE TARGET VALUE Ts FOR COMMUNICATION EQUIPMENT TO AIR CONDITION CONTROL MEANS

*3 RESETTING 10 MINUTE TIMER, START (T2=0) Flag1=0 Flag2=0

*4 MEASURING AND MEMORIZING Tin, Tm BY EACH MINUTE
DETECTING THERMO-ON OF AIR CONDITIONER AND MEMORIZING NUMBER OF THERMO-ONS

*5 MEASURED Tm ONCE EXCEEDS 36°C IN PAST 10 MINUTES

*6 MEASURING SUCTION TEMPERATURE AND CALCULATE AVERAGE VALUE In 10 MINUTES (Tin10)

FIG. 22

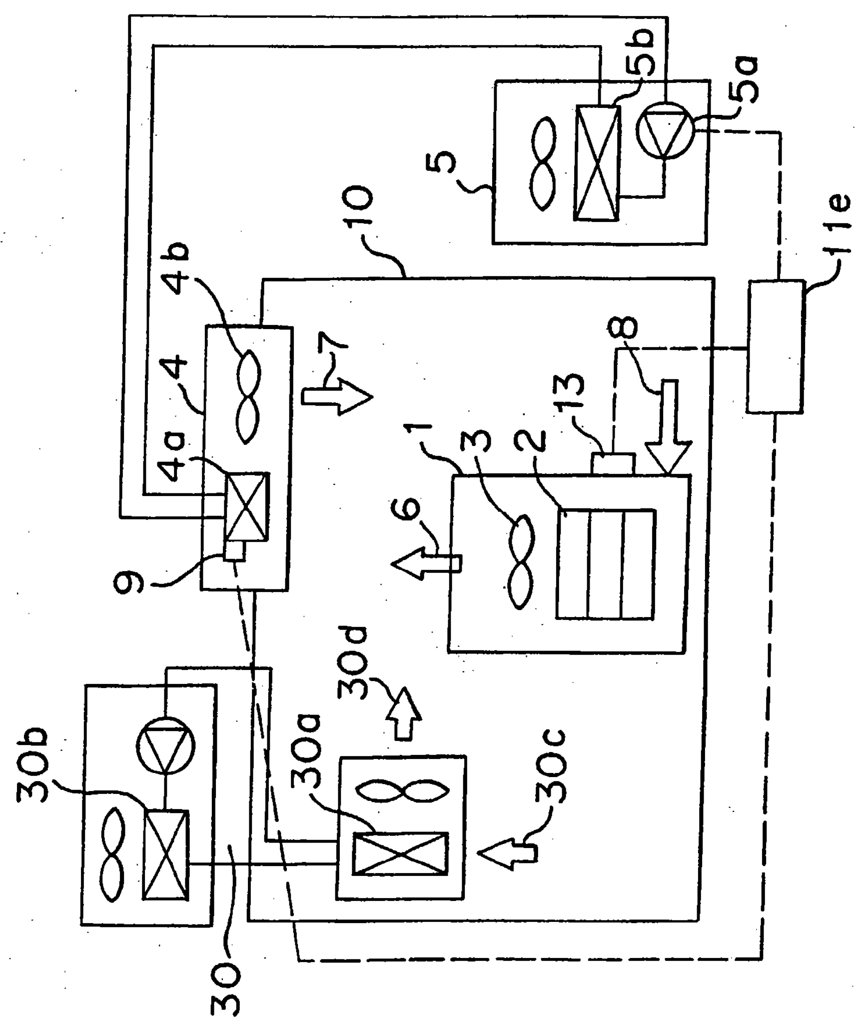


FIG. 23

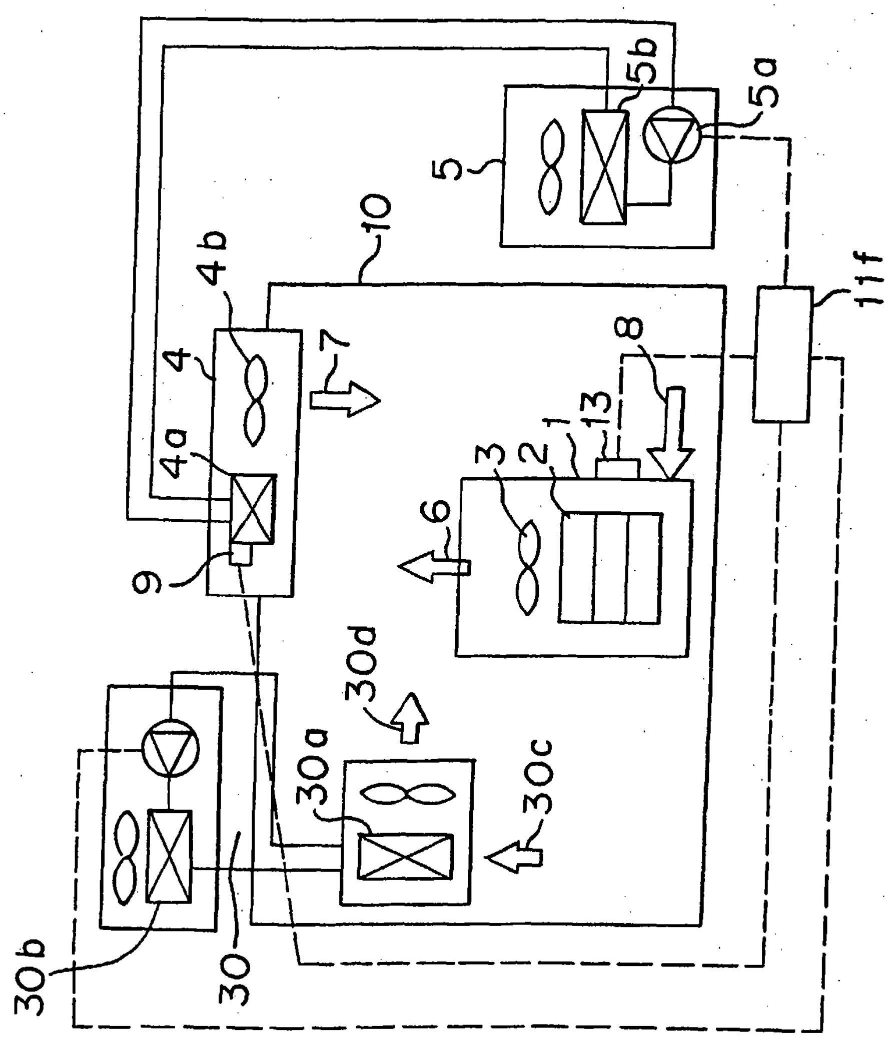


FIG. 24

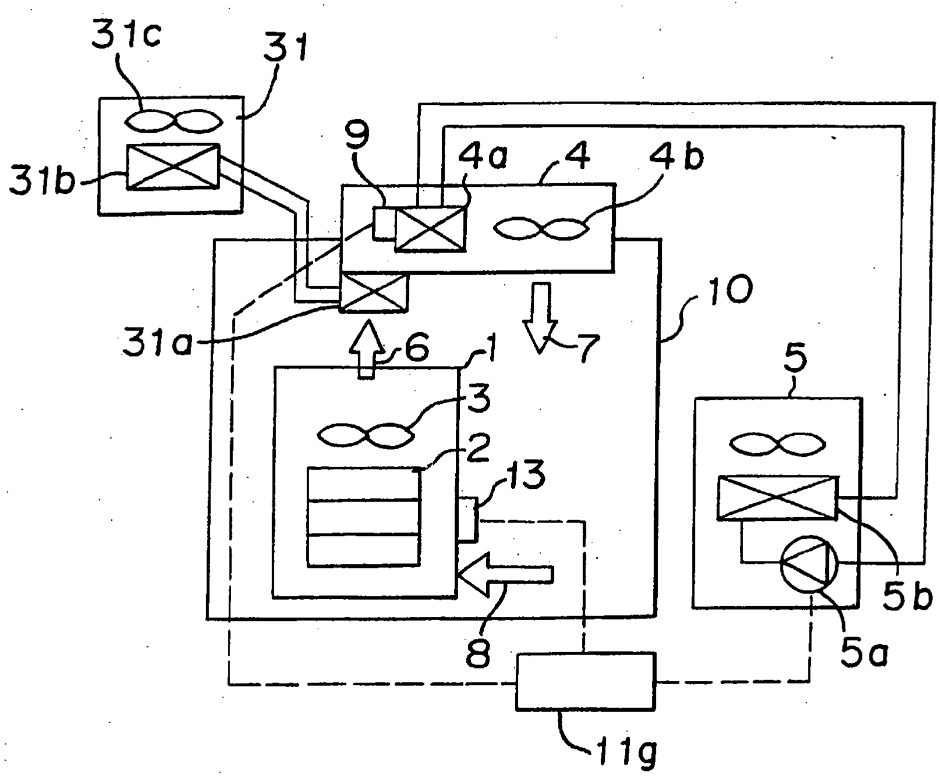


FIG. 25

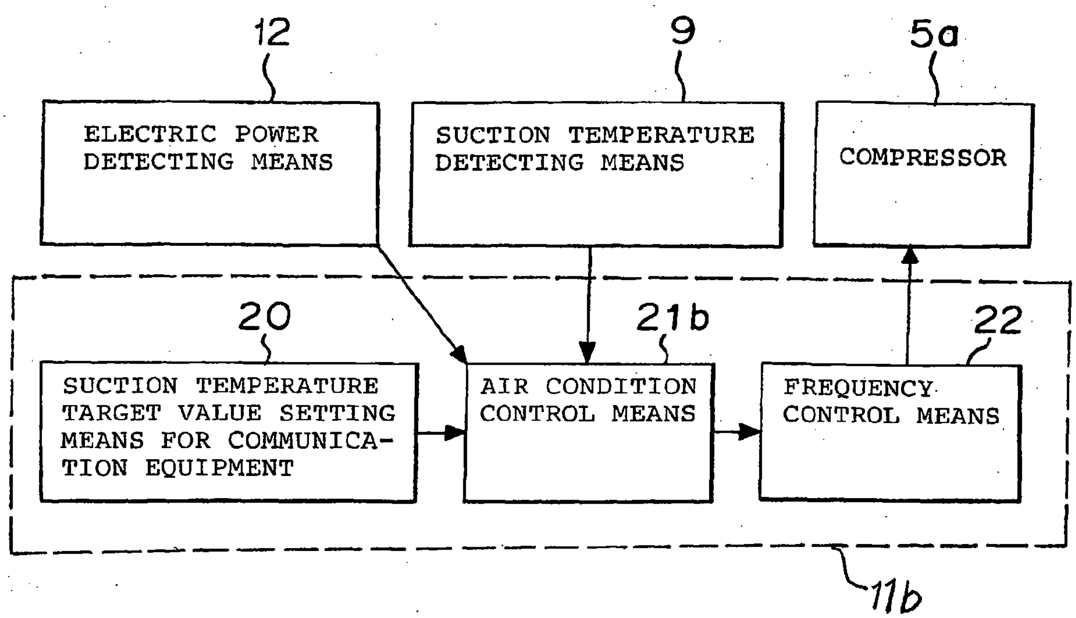
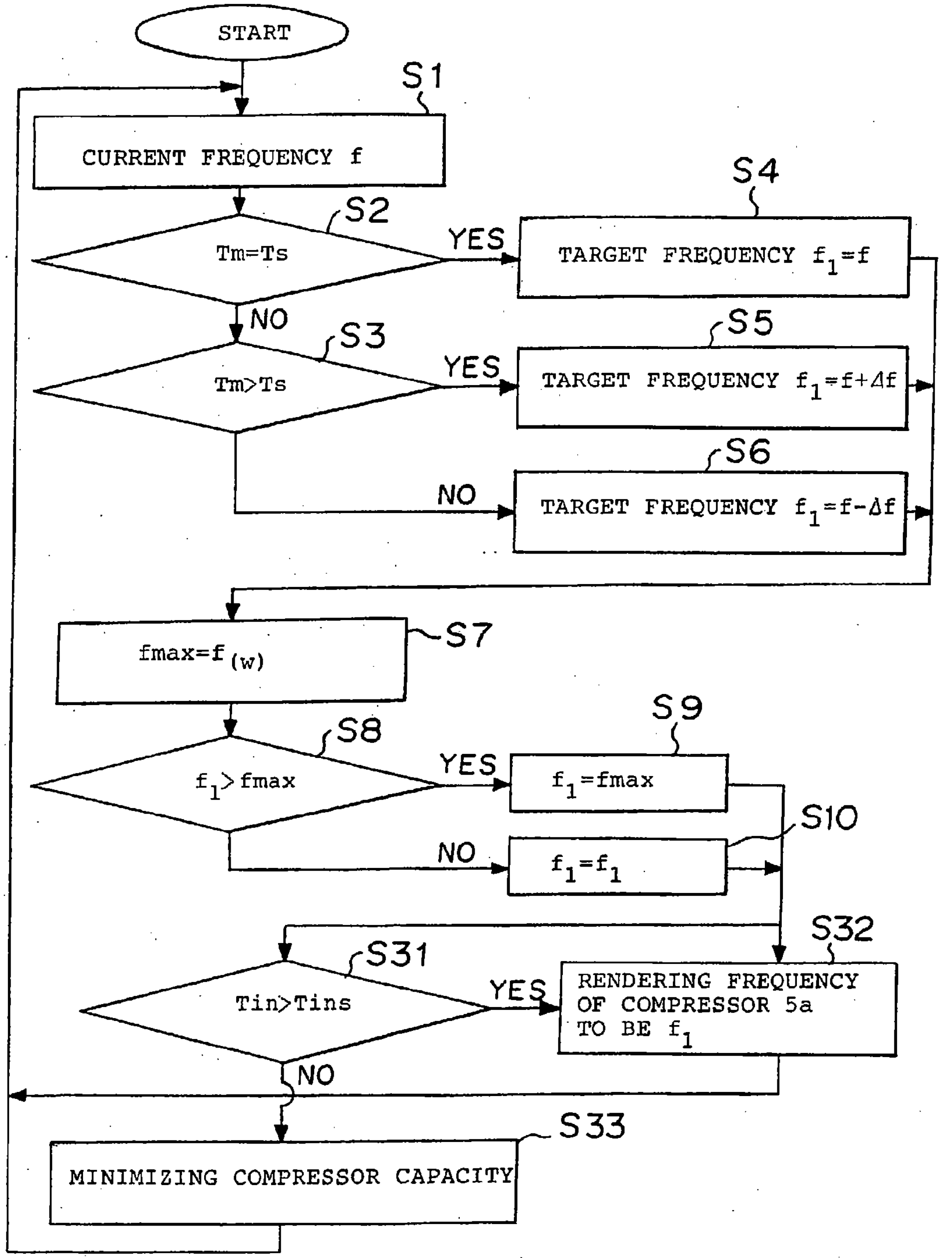
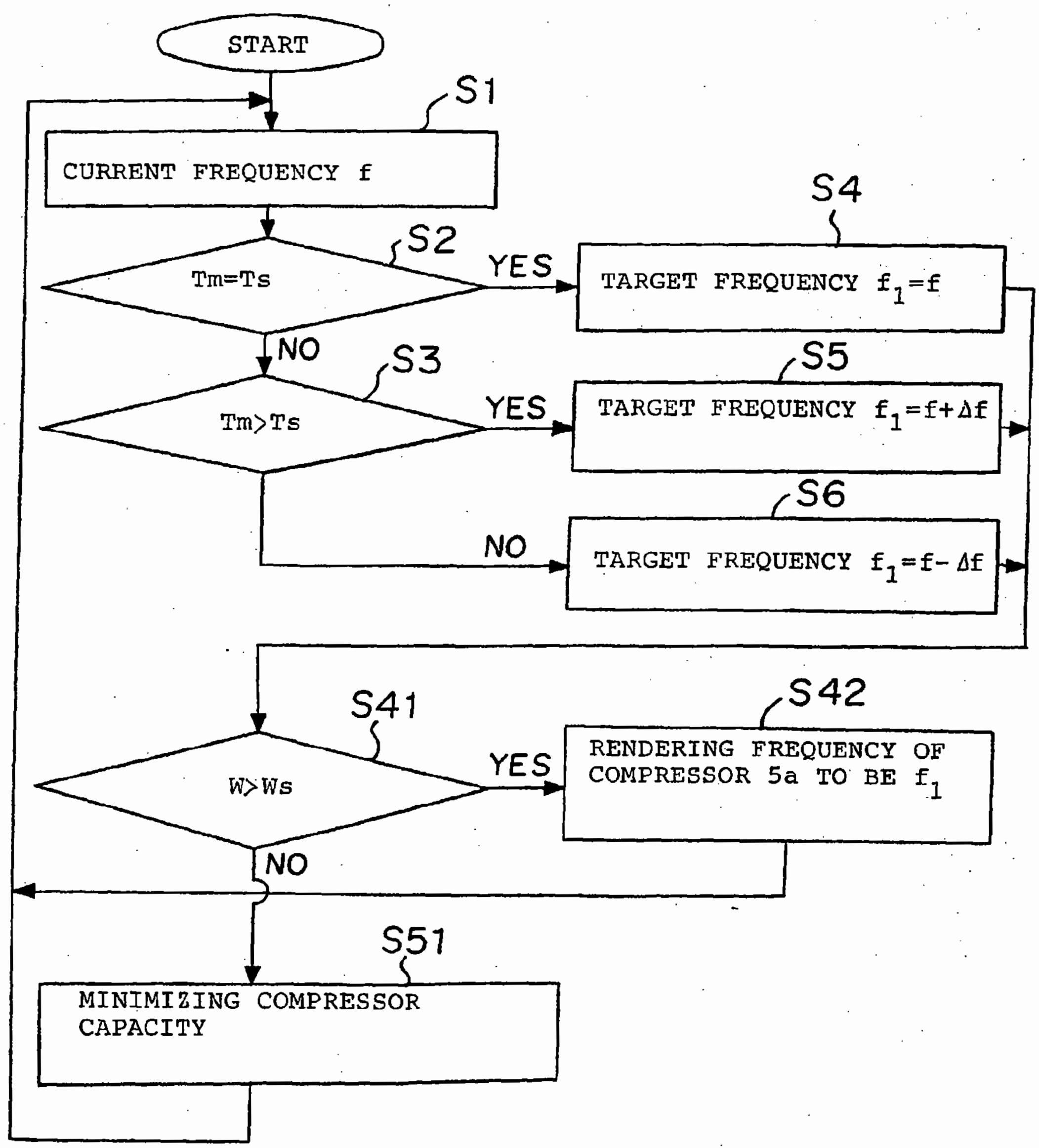

12
ELECTRIC POWER DETECTING MEANS
9
SUCTION TEMPERATURE DETECTING MEANS
5a
COMPRESSOR
20
SUCTION TEMPERATURE TARGET VALUE SETTING MEANS FOR COMMUNICATION EQUIPMENT
21b
AIR CONDITION CONTROL MEANS
22
FREQUENCY CONTROL MEANS
11b

FIG. 26



Tm: TEMPERATURE OF SUCTION AIR 8 FOR COMMUNICATION EQUIPMENT
Ts: SET TEMPERATURE OF SUCTION AIR 8 FOR COMMUNICATION EQUIPMENT
Δf: PREVIOUSLY SET VARIATION OF FREQUENCY
w: OUTPUT FROM ELECTRIC POWER DETECTING MEANS 12
f(w): MAXIMUM FREQUENCY OPERATING FUNCTION HAVING VARIABLE w
Tin: SUCTION TEMPERATURE
Tins: SET VALUE OF SUCTION TEMPERATURE

F I G. 27



Tm: TEMPERATURE OF SUCTION AIR 8 FOR COMMUNICATION EQUIPMENT
Ts: SET TEMPERATURE OF SUCTION AIR 8 FOR COMMUNICATION EQUIPMENT
Δf: PREVIOUSLY SET VARIATION OF FREQUENCY
w: OUTPUT FROM ELECTRIC POWER DETECTING MEANS 12

FIG. 28

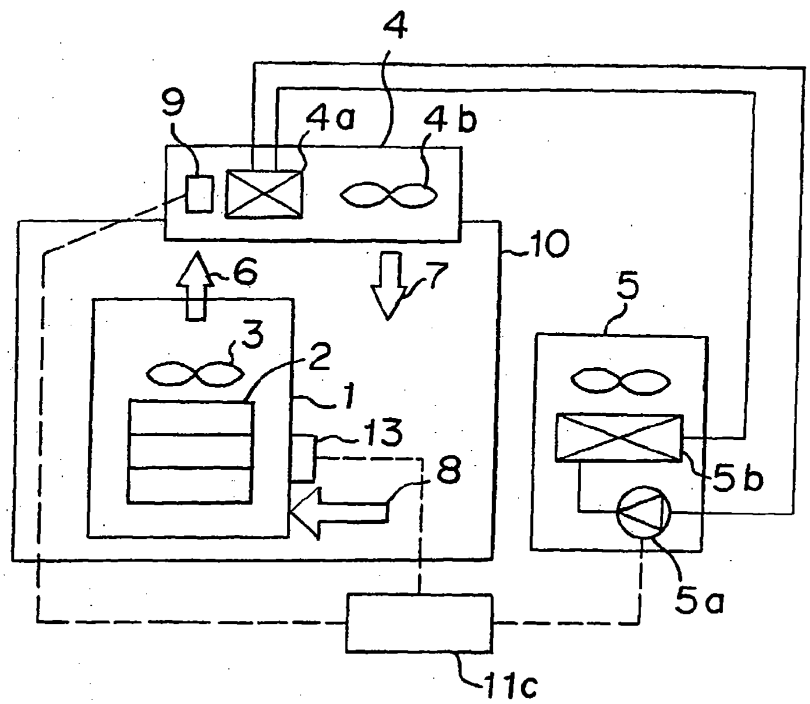
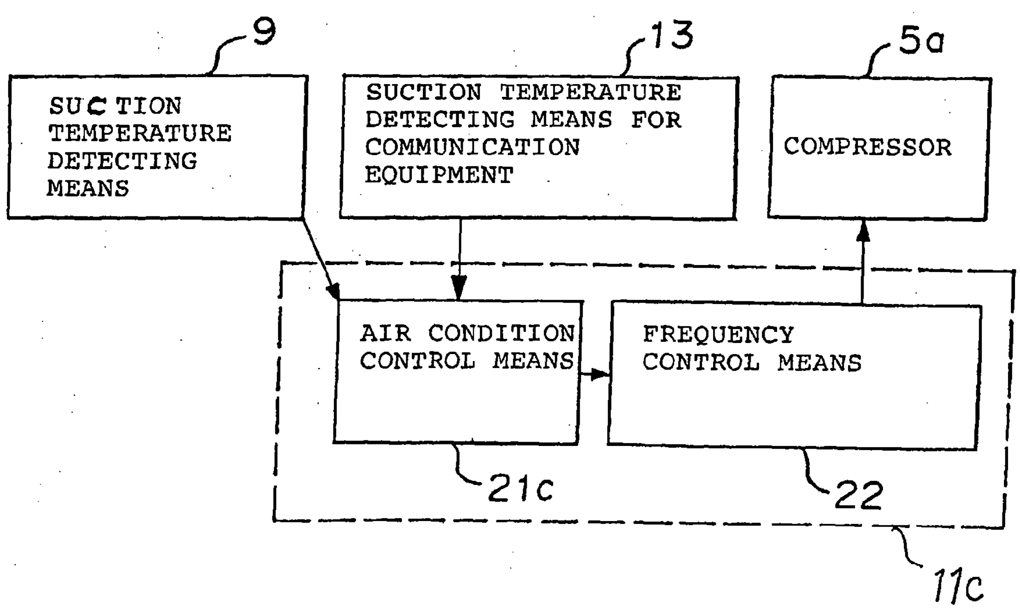
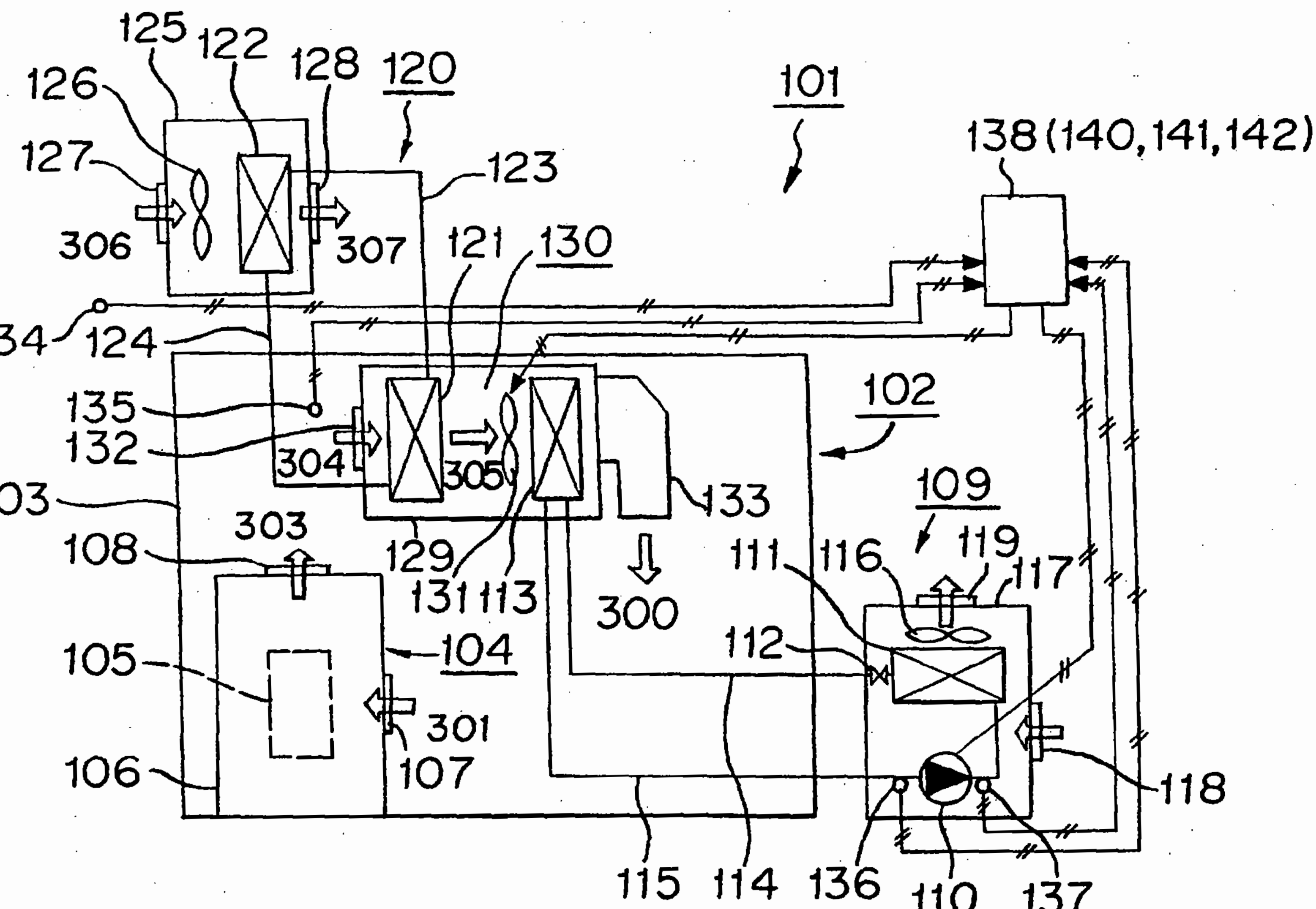

4
9
4a
4b
10
6
7
5
3
2
1
13
8
5b
5a
11c

FIG. 29


9
13
5a
SUCTION TEMPERATURE DETECTING MEANS
SUCTION TEMPERATURE DETECTING MEANS FOR COMMUNICATION EQUIPMENT
COMPRESSOR
AIR CONDITION CONTROL MEANS
FREQUENCY CONTROL MEANS
21c
22
11c

F I G. 30


101
102
103
104
105
106
107
108
109
110
111
112
113
114
115
116
117
118
119
120
121
122
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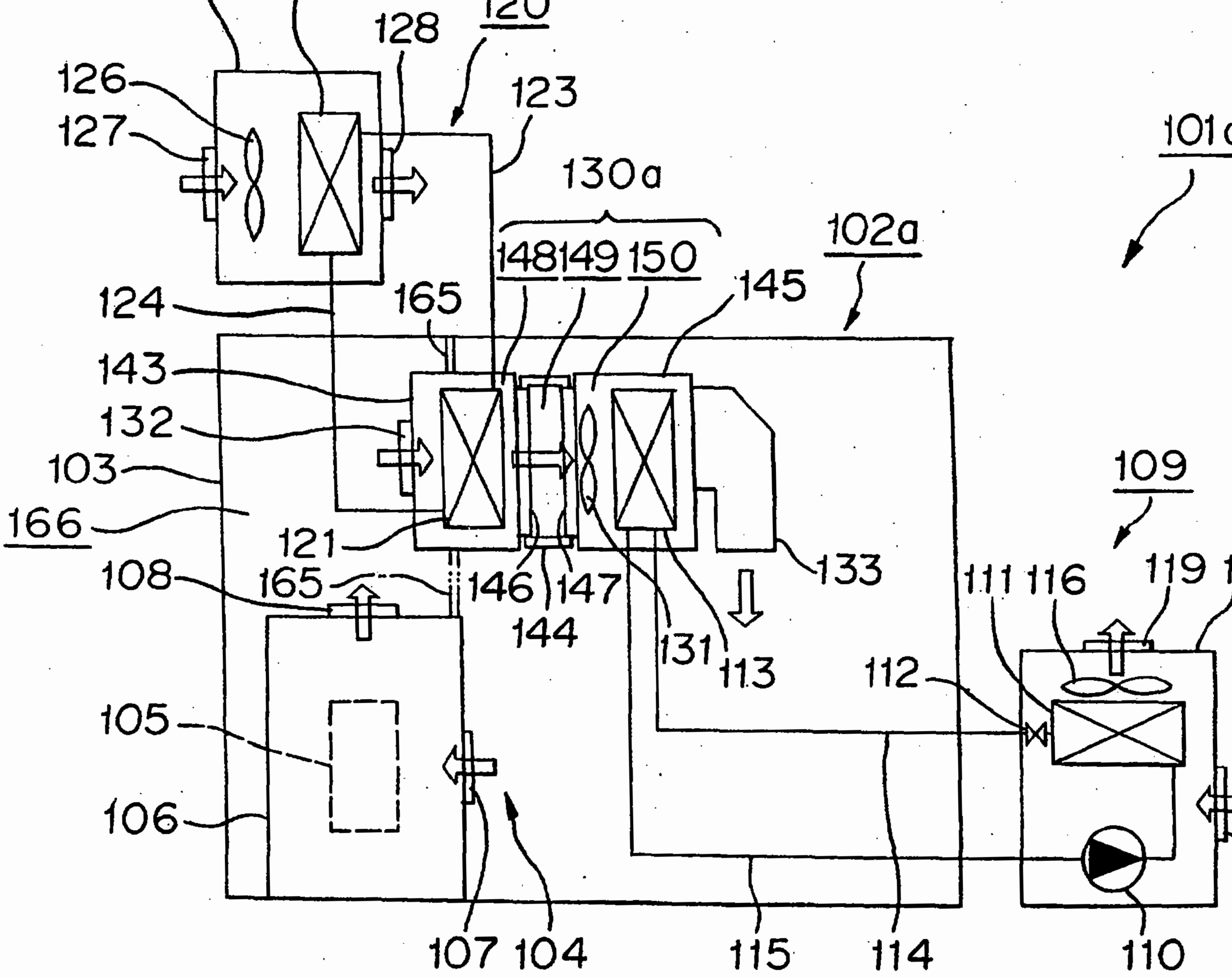
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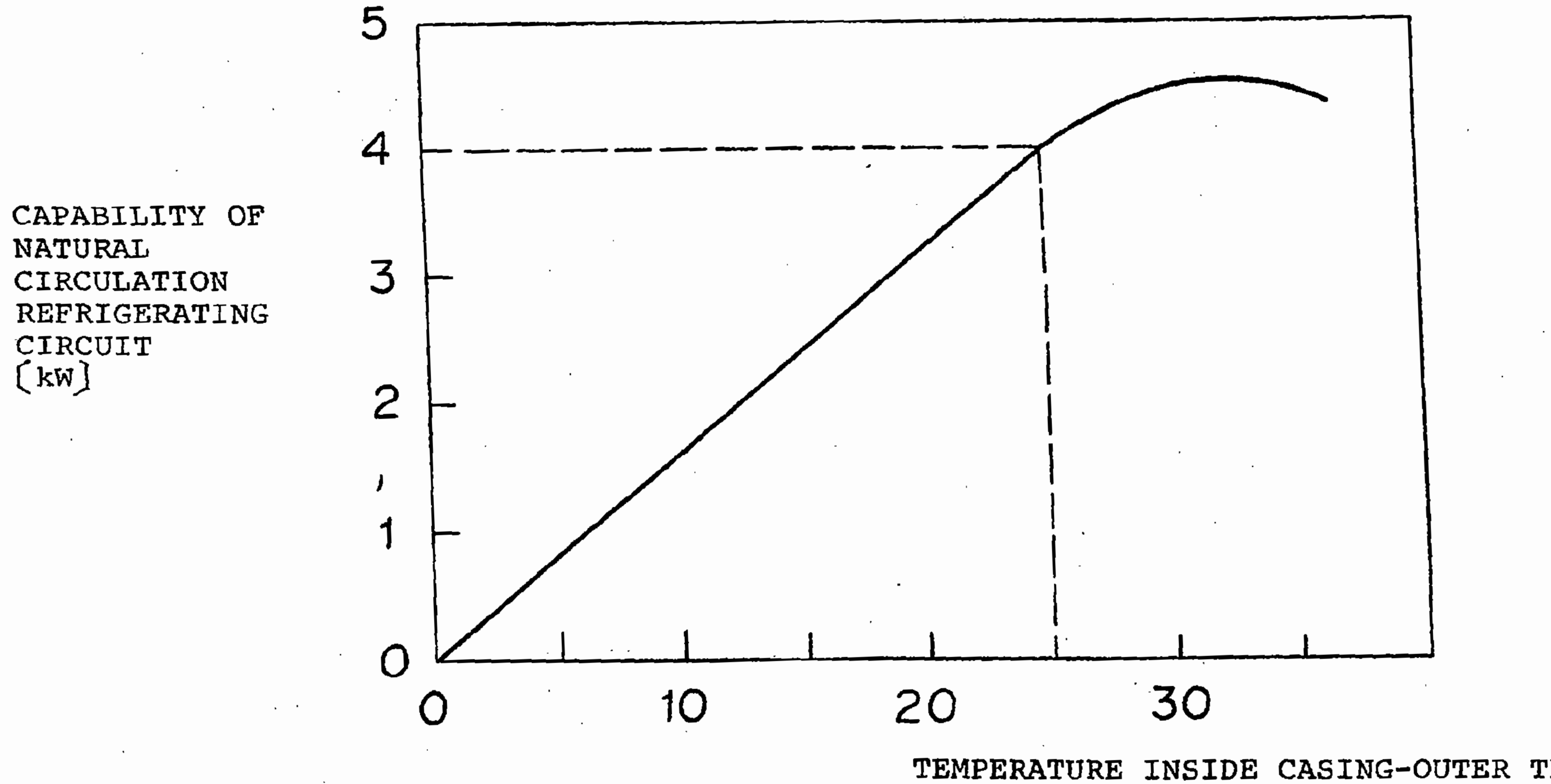
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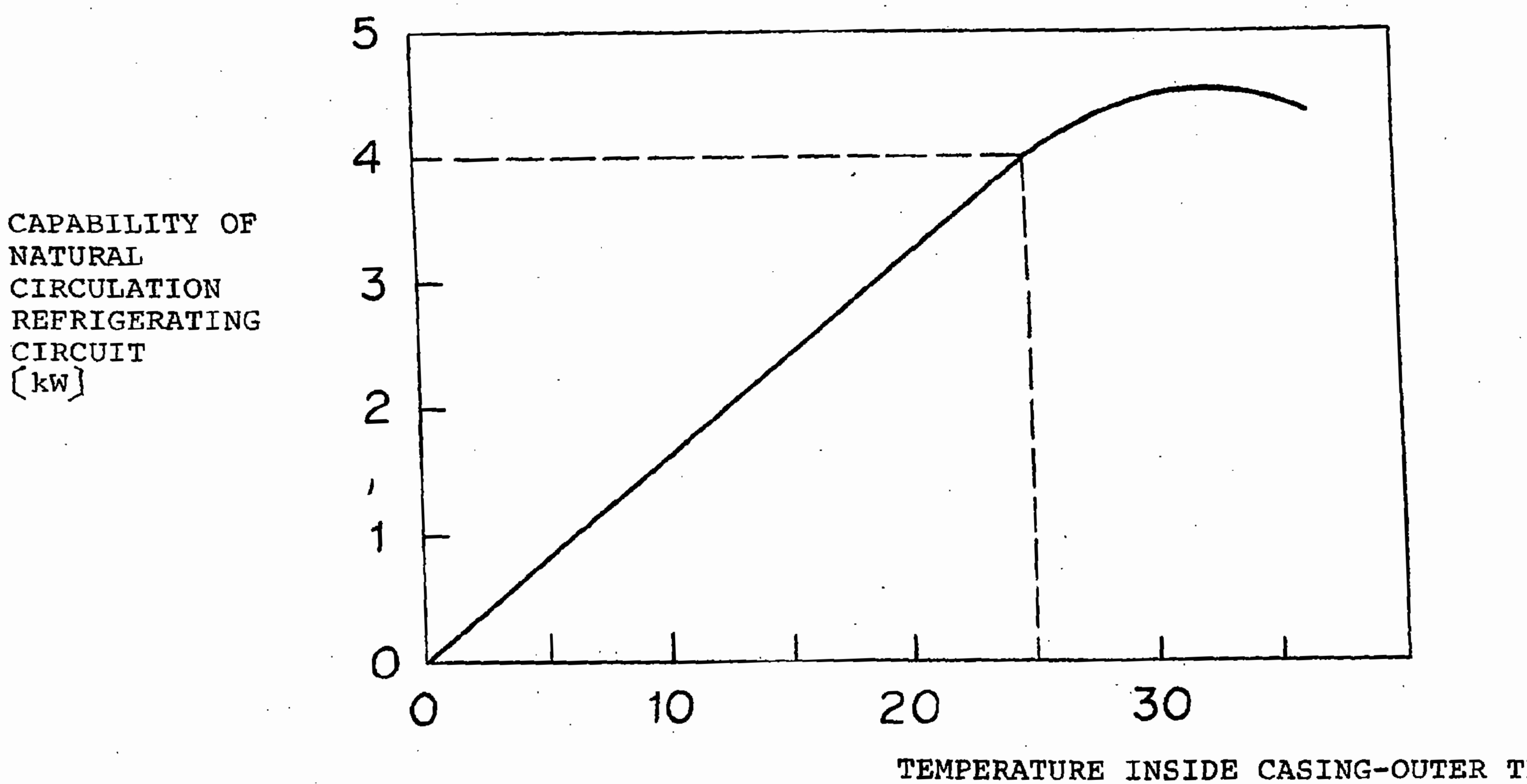
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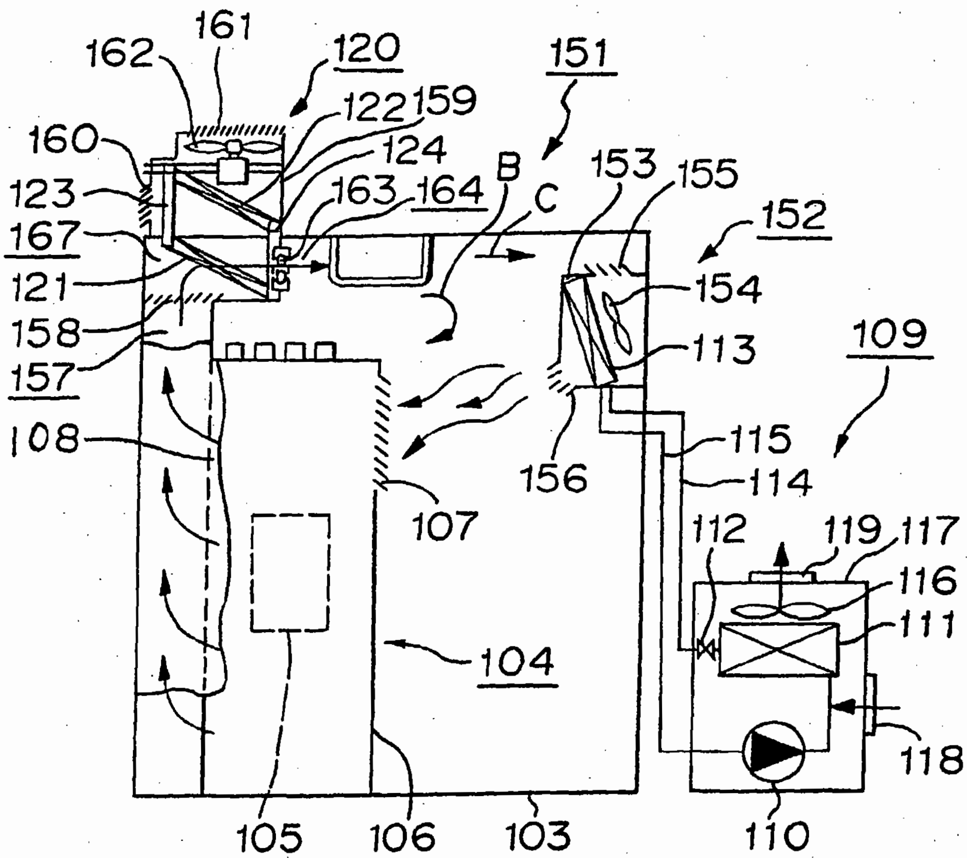
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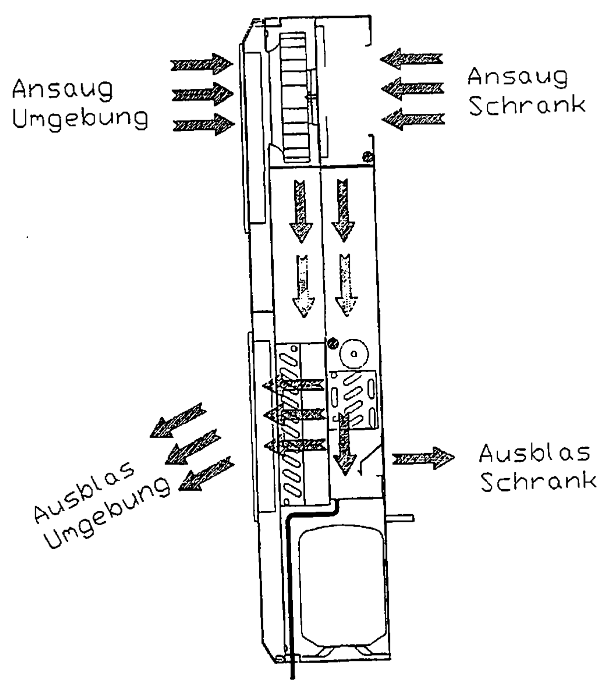
(54) Vorrichtung zum Kühlen eines Schalt- oder Steuerschrankes

(57) Vorrichtung zum Kühlen eines Schalt- oder Steuerschrankes;
1.1 mit einem Luft-Luft-Kühlsystem ("passives Kühlsystem"), umfassend einen ersten Lüfter zum Ansaugen eines Luftstromes aus dem Schrank sowie einen Wärmetauscher;
1.2 mit einem thermodynamischen Kühlsystem ("aktives Kühlsystem"), umfassend einen Verdichter, einen Verflüssiger, einen Sammler/Trockner, ein Expansionsventil, einen Verdampfer sowie einen zweiten Lüfter zum Ansaugen eines zweiten Luftstromes aus der Umgebung;
der Wärmetauscher ist den beiden Lüftern nachgeschaltet.



DE 299 08 370 U 1

Vorrichtung zum Kühlen eines Schalt- oder Steuerschrankes

Die Erfindung betrifft eine Vorrichtung zum Kühlen eines Schalt- oder Steuerschrankes. Solche Steuerschränke enthalten elektrische oder elektronische Geräte oder Baugruppen, die beim Betrieb Wärme erzeugen. Zur Vermeidung von Schäden muß dafür gesorgt werden, daß der Innenraum des Schrankes eine gewisse Höchsttemperatur nicht überschreitet. Darüberhinaus ist es häufig erforderlich, den Innenraum auf einem gleichmäßigen Temperaturniveau zu halten.

Das Problem wird dann gravierend, wenn die Umgebungstemperatur hoch ist. Dies kann saisonal bedingt sein. Doch kann die Umgebungstemperatur auch ständig hoch sein, etwa in Industrieanlagen, in denen beispielsweise durch Kraft- oder Arbeitsmaschinen ständig hohe Temperaturen herrschen.

Zum Kühlen von Schaltschränken der genannten Art werden u.a. sogenannte passive Klimageräte eingesetzt - sogenannte Luft-Luft-Klimageräte. Dabei wird die erwärmte Luft aus dem Innenraum des Schrankes mittels eines Lüfters abgesaugt, durch einen Wärmetauscher hindurchgeführt, im allgemeinen einen Luft-Luft-Wärmetauscher hindurchgefördert, und schließlich wieder auf einen niedrigeren Temperaturwert gekühlt in den Schrank eingeführt. Solche Vorrichtungen arbeiten so lange zufriedenstellend, als keine extremen Umgebungstemperaturen auftreten, und die Anforderungen an eine gleichmäßige Temperatur im Innenraum des Schrankes nicht allzu hoch sind. Die Schaltschranktemperatur hat bei dieser Art der Kühlung je nach Wärmeanfall eine Temperatur, die bis zu 20 - 30 °C über der Raumtemperatur liegt. Vorrichtungen dieser Art sind einfach im Aufbau, kostengünstig in der Anschaffung und verursachen außer gelegentlichen Wartungsarbeiten keine Betriebskosten.

Für höhere Anforderungen sind Klimageräte notwendig, die man als "aktiv" bezeichnen könnte. Diese arbeiten mit einem flüssigen Kältemittel nach dem bekannten linksdrehenden Carnot-Prozeß. Dabei wird gasförmiges Kältemittel von einem Verdichter angesaugt und komprimiert. Sodann wird es in einem Verflüssiger verflüssigt. Es folgt ein Trockner/Sammler. Sodann wird das flüssige Kältemittel in einem Expansionsventil entspannt. Schließlich wird die nunmehr sehr niedrige Temperatur des Kältemittels in einem Verdampfer dazu genutzt, Wärme der Umgebung zu entziehen, im vorliegenden Falle dem betreffenden Schaltschrank.

Mit solchen Kühlvorrichtungen lassen sich praktisch beliebig niedrige Temperaturen erzielen. Auch läßt sich ein bestimmtes Temperaturniveau durch Einsatz von Reglern über längere Zeiträume hinweg einstellen. Die Anschaffungskosten sind höher als bei einer passiven Kühlvorrichtung, lassen sich jedoch häufig aufgrund der gehobenen Anforderungen nicht vermeiden. Außerdem benötigen Geräte dieser Art zu ihrem Betreiben etwa 10 mal soviel elektrische Energie, wie Luft-Luft-Klimageräte, was sich in Betriebskosten niederschlägt, die ganz erheblich sein können.

Ein Beispiel für eine aktive Kühlvorrichtung ist in DE 44 13 128 beschrieben. Ein Beispiel für eine passive Kühlvorrichtung ist in DE 196 41 552 C1 beschrieben.

Der Erfindung liegt die Aufgabe zugrunde, eine Kühlvorrichtung anzugeben, die zum Kühlen eines Schalt- oder Steuerschrankes geeignet ist, die hohen Anforderungen entspricht, insbesondere bei wechselnden Umgebungstemperaturen, die aber verhältnismäßig niedrige Betriebskosten hat.

Diese Aufgabe wird durch die Merkmale der unabhängigen Ansprüche gelöst.

Der Erfinder hat demgemäß in geschickter Weise die Vorzüge der beiden Systeme miteinander kombiniert. Dabei wird der passive Teil des Systems mit seinen Kostenvorteilen ohne Unterbrechung oder fast ohne Unterbrechung benutzt, während der aktive Teil des Systems bei Bedarf zugeschaltet wird. Der Vorteil liegt unter anderem in der Ausnutzung des in jedem Falle notwendigen Wärmetauschers durch die beiden Teile des Gesamtsystems, aber auch durch die geschickte räumliche Zuordnung der einzelnen Aggregate der beiden Teile des Gesamtsystems. In jenen Phasen, in welchen beide Systeme arbeiten, wird die Arbeit des Verflüssigers bei relativ niedrigen Umgebungstemperaturen begünstigt durch den vorgeschalteten Luft-Luft-Wärmetauscher, der ständig betrieben wird.

Die Erfindung ist anhand der Zeichnungen erläutert. Darin ist im einzelnen folgendes dargestellt:

Fig. 1 ist eine schematische Seitenansicht eines erfindungsgemäßen Kombigerätes.

Fig. 2 zeigt denselben Gegenstand, wie Fig. 1, in etwas verkleinertem Maßstab.

Fig. 3 ist eine Draufsicht auf das Kombigerät gemäß Fig. 2 von der Schaltschrankseite her.

Fig. 4 ist eine Draufsicht auf das Gerät gemäß Fig. 2 von der Umgebungsseite her.

Fig. 5 ist eine weitere schematische Schnittansicht des Kombigerätes gemäß Fig. 2, wobei die Schaltschrankseite und die Umgebungsseite gegenüber der Darstellung gemäß Fig. 2 vertauscht sind.

Das Kombinationsgerät ist wie folgt aufgebaut:

Das Luft-Luft-Klimagerät LLK ist durch die Montage einer Verdampferwanne 1a im Gehäuse 1 hermetisch in zwei Kammern geteilt. Eine erste Kammer (Innenkreislauf Schrank) beinhaltet einen Innenlüfter 9, einen Luft-Luft-Wärmetauscher 10, zwei Temperaturfühler 14, 15, ein Expansionsventil 13 sowie die Verschraubung mit einem Kondenswasser-Ablaufschlauch 6.

Eine zweite Kammer (Außenkreislauf) beinhaltet einen Außenlüfter 8, einen Luft-Luft-Wärmetauscher 10, einen Verflüssiger 11, einen Verdichter 7, einen Kondensatablauf 6 und eine Regelplatine 16. Der Außenkreislauf wird durch folgende Anbauteile zur Umgebung begrenzt: einen seitlichen Deckel 1b, einen Deckel mit umlaufender Dichtung 2 und einer Haube 3 mit integrierten Luftleitgittern 4, 5.

Die Bauteile Verdichter, Pressostat, Verflüssiger, Sammler, Trockner, Expansionsventil, Verdampfer sind Komponenten des (aktiven) Kältekreislaufes.

Der Luft-Luft-Wärmetauscher besteht aus einem Verbund mit Aluminium-Profilteilen.

Wie man insbesondere aus Fig. 1 erkennt, gibt es die folgenden Luftkreisläufe:

Innenkreislauf

Die warme Luft im Schrank wird mittels Innenlüfter 9 abgesaugt und durch den Luft-Luft-Wärmetauscher 10 und den Verdampfer 12 gedrückt. Sie verläßt das LLK durch den Ausblasschlitz wieder in den Schrank.

Außenkreislauf

Die kühlere Umgebungsluft wird mittels Außenlüfter 8 durch das Ansauggitter 4 (eventuell mit Filter) angesaugt und durch den Luft-Luft-Wärmetauscher 10 und den Verflüssiger 11 gedrückt. Sie verläßt das LLK durch das Ausblasgitter 5 und wird dadurch schräg nach unten abgelenkt.

Der passive Teil des Kombigerätes, somit das Luft-Luft-Klimagerät, arbeitet wie folgt:

Durch Aneinanderreihen von Aluminium-Profilteilen ist ein Wärmetauscher-Paket 10 mit zwei voneinander luftdicht getrennten Kammern mit Aluminium-Lamellen gebildet.

Diese zwei getrennten Kammern werden einerseits mit Luft aus der Umgebung und andererseits mit Luft aus dem Schrank mit Hilfe von Lüftern durchströmt.

Durch den Temperaturunterschied der beiden Luftvolumenströme wird ein Wärmeaustausch bewirkt. Dies wird genutzt, um die Luft aus dem Schrank mit höherer Temperatur abzukühlen. Gleichzeitig wird die Umgebungsluft erwärmt und wieder an die Umgebung abgegeben.

Die Nutzkühlleistung dieser Art der Kühlung hängt sowohl von der Form und Anzahl der Lamellen und dem Luftvolumenstrom als auch von der bestehenden Temperaturdifferenz zwischen der Umgebungsluft und der Schrankluft ab.

In der Praxis bedeutet dies, bei der Annahme einer konstanten Verlustleistung durch elektrische Bauteile im Schrank, eine im Verhältnis zum Anstieg der Umgebungstemperaturen ansteigende Schranktemperatur. Dies ist bezüglich der Lebensdauer teurer Elektronik im Schaltschrank unerwünscht bzw. bei zu hoher Temperatur nicht tragbar. Zu diesem Zeitpunkt muß der aktive Teil des Kombigerätes zugeschaltet werden.

Der aktive Teil des Gerätes arbeitet wie folgt:

Das Kältemittel R134a zirkuliert im luftdicht geschlossenen Kreislauf durch die o.a. Bauteile. Dabei durchläuft es verschiedene Aggregatzustände von flüssig über Gemisch flüssig-gasförmig und gasförmig, nimmt dabei Wärme auf und gibt sie wieder an definierter Stelle ab.

Im Einzelnen:
Im Sammler/Trockner sammelt sich das Kältemittel im flüssigen Zustand und kann darin seine Volumenänderung ausgleichen. Das integrierte Trocknungssystem ist sehr wichtig, da sich im Kreislauf keine Feuchtigkeit absetzen darf, was zum einen leistungsmindernd und zum anderen zu Beeinträchtigungen der Funktion führen kann.

Der Druck und dadurch die spezifische Temperatur sind in diesem Teil der Anlage relativ hoch (10 bis 17 bar).

Das flüssige Kältemittel wird im Expansionsventil 13 gedrosselt und durch den Druckabfall auf ca. 4 bar (entspricht einer Temperatur von ca. 7 bis 10° C) in einen Mischzustand gasförmig-flüssig gebracht. Diese relativ niedrige Temperatur wird im Verdampfer 12 genutzt, um der Umgebung (bzw. dem Schaltschrank) Wärmeenergie zu entziehen. Das Expansionsventil 13 ist mit einem Temperatur-Fernfühler am Ausgang des Verdampfers verbunden und läßt nur soviel Kältemittel in den Verdampfer einspritzen, daß es zu 100% verdampft. Eine sogenannte Überhitzung von 2 bis 4° Kelvin ist zwar leistungsmindernd aber notwendig, um am Ausgang des Verdampfers 12 dampfförmiges Kältemittel zu garantieren. Der Druck in diesem Teil des Kreislaufes ist annähernd dem Druck hinter dem Expansionsventil 13.

Das gasförmige Kältemittel wird vom Verdichter 7 angesaugt und auf den sogenannten Verflüssigungsdruck von 10 bis 17 bar komprimiert. Dieses Heißgas wird im Verflüssiger 11 bei konstantem Druck und durch Wärmeabgabe an die Umgebung wieder in den flüssigen Zustand versetzt und gelangt dann wieder in den Trockner/Sammler. Damit ist der Kreislauf geschlossen.

Um den Wärmeaustausch am Verdampfer 12 bzw. Verflüssiger 11 zu unterstützen, werden Lüfter 8, 9 eingesetzt.

Dieser Kältemittelkreislauf kann durch den linksläufigen Carnot-Prozeß annähernd beschrieben werden.

Bei dem beschriebenen Kombigerät werden beide Arten der Wärmeübertragung zur Schaltschrankkühlung genutzt. Dabei übernehmen sowohl für die Luft-Luft- als auch für den Kältemittelbetrieb zwei Lüftern 8, 9 den Transport der Luft zum Wärmeaustausch.

Dabei ist das Wärmetauscherpaket 10 für die Luft-Luft-Übertragung konstruktiv so angeordnet, daß die beiden Luftkreisläufe immer zuerst diesen Wärmetauscher durchströmen.

Sind optimale Voraussetzungen (d.h. niedere Umgebungstemperaturen) vorhanden, reicht die Nutzkühlleistung der Luft-Luft-Wärmeübertragung aus, die produzierte Wärme aus dem Schaltschrank abzuführen.

Steigende Umgebungstemperaturen vermindern die Leistung des Luft-Luft-Wärmetauschers und führen dazu, daß die Nutzkühlleistung allein durch diese

Komponente nicht mehr ausreicht, eine konstante Temperatur im Schaltschrank zu gewährleisten. Dadurch wird der Kältemittelkreislauf aktiviert und erhöht die geforderte Nutzkühlleistung. Der Kältemittelkreis bleibt solange aktiv, bis eine bestimmte Solltemperatur im Schaltschrank wieder erreicht wird.

Sind die Umgebungstemperaturen so hoch, daß die Leistung der Luft-Luft-Wärmeübertragung gegen Null geht, übernimmt der Kältemittelkreislauf die gesamte geforderte Nutzkühlleistung.

Übersteigen die Umgebungstemperaturen die Solltemperatur im Schaltschrank, muß der Kältemittelkreislauf zusätzlich die entgegenwirkende "Negativleistung" des Luft-Luft-Wärmetauschers kompensieren. Diese Kompensation kann bis zu 10% der Nominalleistung des Kühlgerätes erreichen.

Die Vorteile des erfindungsgemäßen Kombigerätes sind die folgenden:

Durch den Luft-Luft-Wärmetauscher 10 ist immer gewährleistet, daß sich die Luft vor Eintritt in den Verflüssiger 11 erwärmt und dadurch diesem Phänomen entgegenwirkt. D.h. durch diese Kombination ist das Kombigerät bei niederen Umgebungstemperaturen herkömmlichen Kältemittel-Kühlgeräten immer überlegen. Die Luft, die den Verflüssiger 11 anströmt, wird durch das Luft-Luft-Wärmetauscherpaket 10 angekühlt. Dies hat zur Folge, daß der zulässige Kondensationsdruck (begrenzt durch die Bauteile) in diesem Bereich immer niederer liegt, als bei herkömmlichen Kältemittel-Kühlgeräten. Die Nutzkühlleistung im oberen Temperaturbereich der Umgebungsluft nimmt zwar vergleichsweise ab, der Betrieb bei höheren Umgebungstemperaturen ist aber gewährleistet.

Durch die Anwendung des Kombigerätes läßt sich gegenüber einem vergleichbaren, konventionellen Kältemittel-Kühlgerät eine Energieeinsparung von über 50% erzielen, wenn sich der Schaltschrank in einer Halle befindet. Befindet er sich im Freien, so sind sogar Energieeinsparungen von bis zu 90% erreichbar. Ein weiterer Vorteil besteht darin, daß die Aggregate, die Bestandteile des aktiven Teiles des Kombigerätes sind, eine höhere Lebensdauer haben, da sie nur etwa während eines Zehntels der Laufzeit eines herkömmlichen Gerätes betrieben werden müssen.

Einsatz bei niedrigen Umgebungstemperaturen

Beim Einsatz von Kompressor-Kühlgeräten ist darauf zu achten, daß die Umgebungstemperatur nicht zu niedrig ist, da sonst der Kältekreislauf (thermodynamischer Prozeß) zusammenbricht. Der Verflüssiger von Kälteanlagen wird konstruktiv bedingt immer der Umgebungstemperatur ausgesetzt. Fällt die Umgebungstemperatur unter 0°C, erhöht sich die Leistung des Verflüssigers, was dazu führt, daß sich der größte Teil des Kältemittels im Verflüssiger sammelt und dadurch dem Kühlkreislauf entzogen wird.

Um bei niedrigen Umgebungstemperaturen <0°C eine Nutzkühlleistung zu gewinnen, muß die Verflüssigertemperatur durch spezielle Technik auf einer bestimmten Mindesttemperatur gehalten werden, gleichzeitig ist aber darauf zu achten, daß eine Vereisung der Lamellen bei Verdampfungstemperaturen von kleiner -3°C die Leistung erheblich mindert.

Diese Tatsachen begrenzen den Einsatz von herkömmlichen Kältemittel-Kühlgeräten im Bereich niederer Umgebungstemperaturen.

Das Luft-Luft-Kompressor-Kühlgerät hat dabei einen erheblichen Vorteil. Durch das Vorschalten des Luft-Luft-Wärmetauschers wird die Umgebungsluft erwärmt (bei maximaler Leistung ca 10° K) und erst dann dem Verflüssiger zugeführt. Dadurch kann dieses Gerät vergleichbar mit herkömmlichen Geräten bei Umgebungstemperaturen, die um diese Temperaturdifferenz (ca. 10° K) niedriger sind, eingesetzt werden.

Einsatz bei hohen Umgebungstemperaturen

Beim Einsatz von hohen Umgebungstemperaturen ist der Einsatzbereich durch den maximal zulässigen Kondensationsdruck von 17 bis 21 bar beschränkt, was beim Kältemittel R134a einer Temperatur von 65 bis 70 °C entspricht.

Dadurch ist bei herkömmlichen Kältemittel-Kühlgeräten eine maximale Einsatztemperatur von ca. 55°C vorgegeben.

Beim erfindungsgemäßen Kombigerät wird durch den vorgeschalteten Luft-Luft-Wärmetauscher die Luft vor dem Eintritt in den Kondensator um ca. 5° K gekühlt, was theoretisch den Einsatzbereich ebenfalls um 5° K erhöht.

Notlaufeigenschaften

Ein nicht zu unterschätzender Vorteil der konstruktiven Lösung beim erfindungsgemäßen Kombi-Klimagerät ist die Tatsache, daß bei Außenaufstellungen (z.B. Mobilfunkstationen) eine Notlauf-Nutzkühlleistung beim Ausfall des Verdichters durch den Luft-Luft-Wärmetauscher gegeben ist.

Des weiteren können die zur "Notkühlung" relevanten Lüfter so ausgelegt werden, daß sie mit einer Gleichspannung von 24V betrieben werden. Dadurch kann die Spannung bei Ausfall des Netzes durch einfache Batterien zur Verfügung stehen.

Bei herkömmlichen Kältemittel-Kühlgeräten ist es gar nicht oder allenfalls kurzzeitig möglich, den Verdichter für den Notlauf mit der erforderlichen Energie bei Netzausfall zu versorgen.

Regelverhalten und Steuerung

Hauptbestandteil der Regelung und Steuerung ist die Regelplatine 16 mit den zwei Temperaturfühlern 14, 15.

Der Innenlüfter ist nach der Verbindung des LLK mit der Netzspannung immer in Betrieb.

Der Temperaturfühler 14 fühlt die Temperatur nach dem Ansaugen aus dem Schrank. Erreicht die Temperatur am Fühler 14 die Solltemperatureinstellung (z.B. 35°C), so wird der Außenlüfter 8 eingeschaltet. Falls die Umgebungstemperatur kleiner als die Solltemperatureinstellung ist, wird dadurch die Luft des Innenkreislaufes durch den Luft-Luft-Wärmetauscher 10 abgekühlt und verläßt das LLK in den Schrank. Die Luft im Außenkreislauf wird erwärmt und an die Umgebung abgegeben. Ist durch den Betrieb die Nutzkühlleistung größer als die Verlustleistung im Schrank, kühlt sich die Luft im Schrank ab und der Fühler 14 gibt der Regelplatine 16 das Signal zum Ausschalten des Außenlüfters 8. Ist die erforderliche Nutzkühlleistung größer als durch den Luft-Luft-Wärmetauscher 10 bereitgestellt werden kann, erwärmt sich die Luft im Schrank, übersteigt die Solltemperatureinstellung und veranlaßt beim Erreichen der Solltemperatur + 5° K am Fühler 15 am Ausgang des Luft-Luft-Wärmetauschers 10 die Regelplatine 16, den Verdichter 7 einzuschalten. Durch den Betrieb des Verdichters 7 wird der Kältekreislauf gestartet und erhöht zusätzlich die zur Verfügung stehende Nutzkühlleistung. Der Kältekreislauf wird solange aufrechterhalten, bis der Fühler 15 die Solltemperatur + 5° K erreicht hat. Um ein unzulässig häufiges Takten des

Verdichters 7 zu vermeiden, ist die Regelplatine 16 mit einer Anlaufverzögerung zur Steuerung des Verdichters 7 ausgestattet.

Des weiteren ist das Gerät mit einer Leistungsregelung durch Heißgasbeimischung in den Verdampfer ausgestattet (nicht dargestellt).

Die Vorteile des erfindungsgemäßen Kombigerätes lassen sich wie folgt zusammenfassen:

Vielseitiger und umfassender Einsatzbereich;
konstante Nutzkühlleistung durch logische Verknüpfung der einzelnen Komponenten zur Wärmeübertragung und durch Leistungsregelung über Heißgas-Bypassbeimischung;
Notlaufeigenschaften durch Luft-Luft-Wärmetauscher;
beträchtliche Energieeinsparung;
geringe Geräuschemission bei Solobetrieb des Luft-Luft-Wärmetauschers.

Ansprüche

1. Vorrichtung zum Kühlen eines Schalt- oder Steuerschrankes;

1.1 mit einem Luft-Luft-Kühlsystem ("passives Kühlsystem"), umfassend einen ersten Lüfter zum Ansaugen eines Luftstromes aus dem Schrank sowie einen Wärmetauscher;

1.2 mit einem thermodynamischen Kühlsystem ("aktives Kühlsystem"), umfassend einen Verdichter, einen Verflüssiger, einen Sammler/Trockner, ein Expansionsventil, einen Verdampfer sowie einen zweiten Lüfter zum Ansaugen eines zweiten Luftstromes aus der Umgebung;

der Wärmetauscher ist den beiden Lüftern nachgeschaltet.

2. Vorrichtung nach Anspruch 1, dadurch gekennzeichnet, daß das passive Kühlsystem ein Luft-Luft-Kühlsystem ist.

3. Vorrichtung nach Anspruch 1 oder 2, gekennzeichnet durch die folgenden Merkmale:

3.1 es sind zwei Kammern vorgesehen, die aneinander grenzen;

3.2 die erste Kammer enthält den ersten Lüfter (Innenlüfter), einen Teil des Wärmetauschers, einen Temperatursensor, das Expansionsventil;

3.3 die zweite Kammer enthält den zweiten Lüfter (Außenlüfter), einen Teil des Wärmetauschers, einen Verflüssiger, einen Verdichter.

4. Vorrichtung nach Anspruch 3 dadurch gekennzeichnet, daß die erste Kammer baulich an den Schrank angeschlossen ist, während die zweite Kammer der freien Umgebung zugewandt ist.

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Fig.1

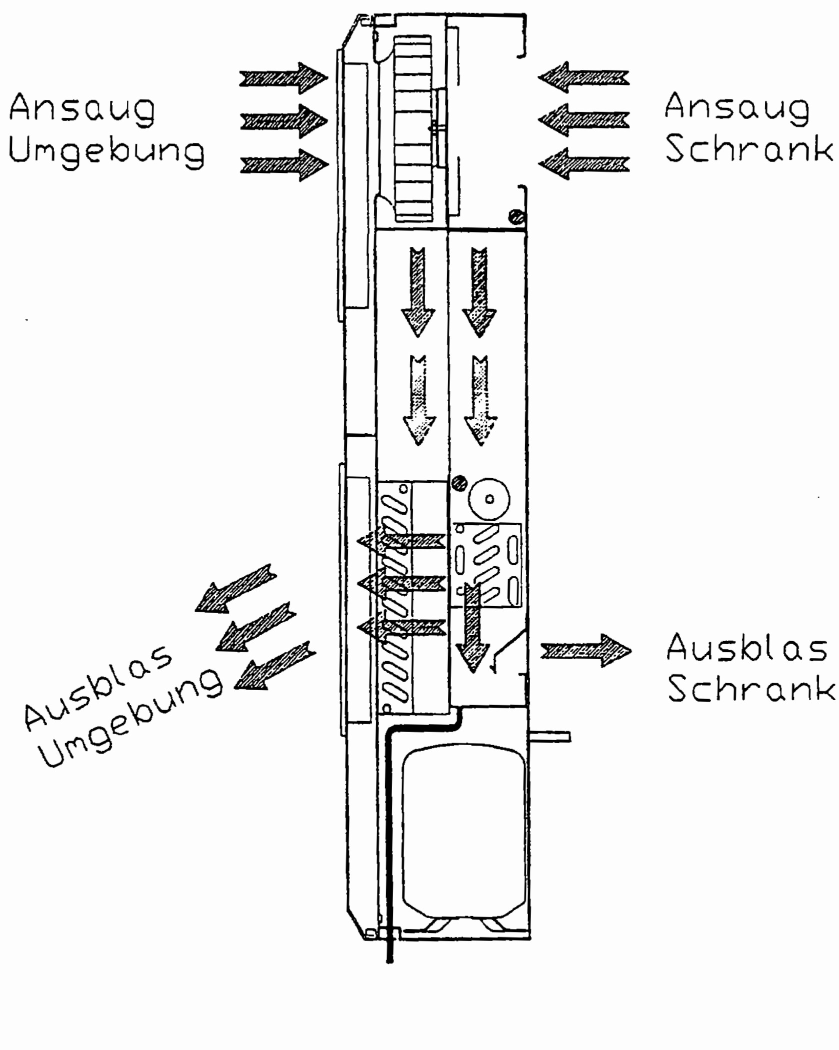
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Fig.3

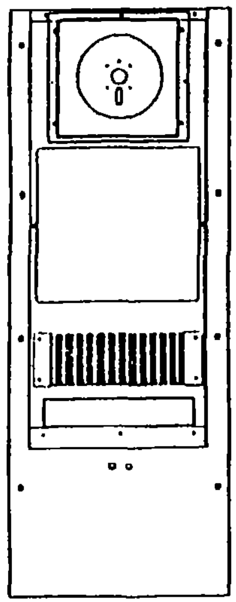


Fig.2

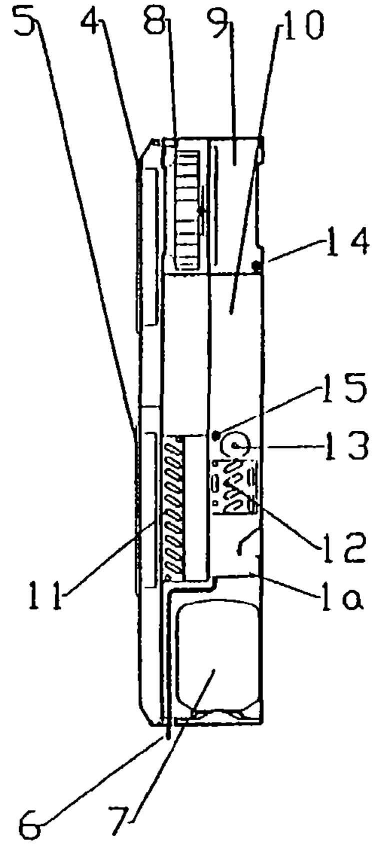


Fig.4

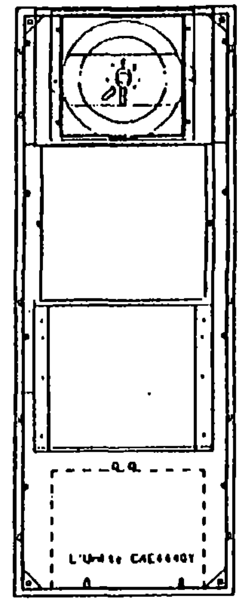
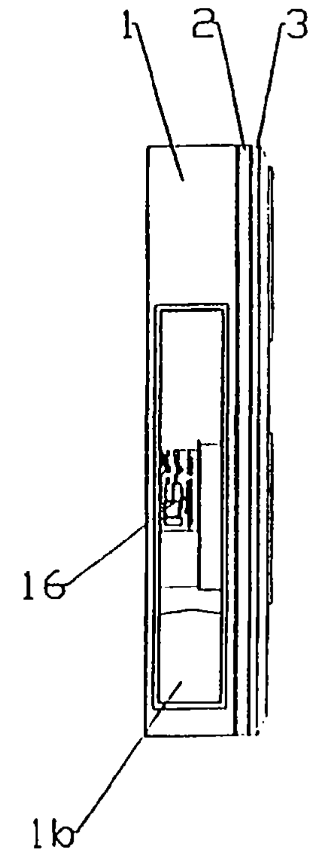


Fig.5



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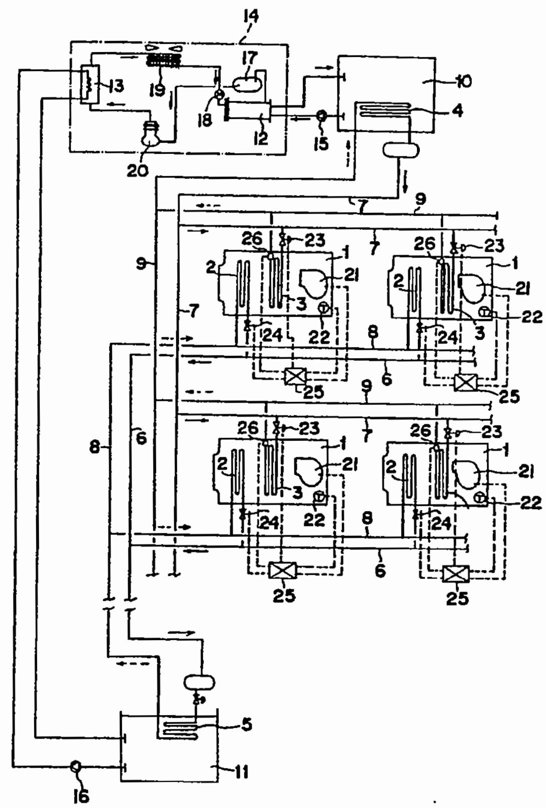
EP 0 281 762 A2

(54) Air conditioning system for buildings.

(57) The disclosure relates to an air conditioning system for buildings which includes a plurality of air conditioners on the different floors, a cooling heat pipe of a gravity type for connecting a vaporizers in the air conditioners with a cold thermal source installed on a top of the building and a warming heat pipe of a gravity type for connecting a condenser of the air conditioners with a hot thermal source installed on a basement of the building. The heat pipe are, respectively, provided with a flow control valves, at inlet to the vaporizer and the outlet to condenser. Each air conditioner is provided with thermistor for detecting the temperature of the returning air and a liquid level detection switch for detecting the liquid level of the thermal medium in the vaporizer. The flow control valves are respectively controlled based on the detection signals from the thermistor and the liquid level detection switch.

Fig. 1



AIR CONDITIONING SYSTEM FOR BUILDINGS

BACKGROUND OF THE INVENTION

1. Field of the invention

The present invention generally relates to an air conditioning system for buildings and more particularly, to an air conditioning system of a type in which the thermal conveyance is conducted through the heat pipes of a gravity type.

2. Description of the prior art

Generally, in an air conditioning system for a building water has been in use as a thermal medium for thermal conveyance between the thermal source equipment and the individual air conditioners. Such use of water, however, has involved a problem relating to troubles of water leakage once an air conditioner gives way to such a disorder. Recently, therefore, for air conditioning system for buildings, introduction of a volatile substance, such as freon, as a thermal medium to be passed directly between the thermal source equipment and the respective heat exchangers of the air conditioners is coming into consideration so as to reduce troubles of liquid leakage by virtue of the volatility of the substance.

The inventors has provided such a type of air conditioning system which utilizes the heat pipe of a gravity type (refer to European Pat. Application/87 115: 909.1). This air conditioning system includes a first thermal storage tank as a cold thermal reservoir which is installed at a high place, such as the roof, in an air-conditioned building; a second thermal storage tank as a hot thermal reservoir which is installed at a low place, such as the basement, in the same building; a plurality of air conditioners each of which is installed in an air-conditioned room or at a place close to it at a height such as the floors, between the first thermal storage tank as a cold thermal reservoir and the second thermal storage tank as a hot thermal source; and heat pipe of a gravity which connect the air conditioner with the first heat storage tank and with the second heat storage thank. The heat pipe of a gravity type is essentially designed to allow thermal medium to circulate therethrough under a natural pressure for circulation generated by phase change of the thermal medium.

In the air conditioning system as described above the flow of the circulating thermal medium is adjusted to control the warming and cooling conditions in accordance with the heat load in the respective air-conditioned rooms. It is to be noted, however, there are some problems in controlling the flow of the thermal medium. For example, in the air conditioners installed on the different floors the water head of the thermal medium in liquid phase is different between the respective air conditioners, resulting in that the equal heat exchange condition can not be obtained in the respective air conditioners. More specifically, since the air conditioner arranged at a lower floor receives a higher water head, the flow control by the throttle valve or the like is very difficult. When the throttle valve is overly throttled, the thermal medium becomes in short supply to the air conditioner. On the contrary to the above, if the heat pipe for vapor at this air conditioner is filled by the liquid of thermal medium, the smooth flow of the vaporized thermal medium is hampered so that the heat exchange at the air conditioner can not be effected efficiently, resulting in a poor cooling. In addition, in the state in which higher water head is caused, the temperature of the thermal medium in vapor phase becomes higher so that the sufficient cooling can not be obtained.

Moreover, when the heat load at the different air-conditioned rooms are different from each other, the warming and cooling operations at their rooms must be separately adjusted in accordance with their specific heat load and accordingly, when, for instance, the air conditioning system is operated in the cooling mode, in the air conditioner being subject to larger heat load, the amount of exchanged heat and accordingly the amount of vapor of the thermal medium become larger so that the resistance in the heat pipe for vapor becomes larger, that is, since the line or the air conditioners requiring larger flow of thermal medium is subject to larger resistance, a great quantity of thermal medium is, disadvantageously, supplied to the other line or air conditioners which is not subject to large heat load and accordingly is not need large quantity of thermal medium.

SUMMARY OF THE INVENTION

Accordingly, an essential object of the present invention is to provide an air-conditioning system for buildings including a plurality of air conditioners installed, respectively, on the different floors and the heat pipes of a gravity type for effecting the thermal conveyance, which can control the respective air conditioners on the different floors with high accuracy to condition the temperature of the air in the respective airconditioned locations as desired.

In accomplishing this and other objects, there is provided an air conditioning system for buildings which comprises:

a cold thermal source which is installed at a high place in the building,

a hot thermal source which is installed at a low place in the building,

air conditioners which are installed at various heights between said cold and hot thermal sources and each of which has, respectively, a condenser as a heat exchanger for warming air in air-conditioned location, a vaporizer as a heat exchanger for cooling the air in the air-conditioned location and a fan for blowing air into the air-conditioned location through said condenser and vaporizer,

a cooling heat pipe of a gravity type by which the vaporizers of the air conditioners are connected with the cold thermal source, and in which thermal medium is filled so as to circulate therein,

a warming heat pipe of a gravity type by which the condensers of the air conditioners are connected with the hot thermal source and in which thermal medium is filled so as to circulate therein,

temperature sensors for detecting temperature of the air in the air-conditioned locations,

flow control valves which are respectively incorporated in the heat pipes at inlets of the vaporizers and at outlets of the condensers,

liquid level detection means for detecting a predetermined level of the thermal medium in liquid phase in the vaporizer, and

controllers which control, respectively, the flow control valves based on detection signals supplied from the temperature sensors and the liquid level detection means so that the air in the air-conditioned locations meet a set point. provided

With the above air conditioning system, the warming operation is effected by the first natural circulation line including the hot thermal source, air conditioners and the warming heat pipe, while the cooling operation is effected by the second natural circulation line including the cold thermal source, air conditioners and the cooling heat pipe. The flow of the thermal medium being supplied into the condensers and vaporizers is controlled by the corresponding flow control valves. The valves are controlled so as to accomplish the desired temperature of the returning air returning to the respective air conditioners. For example, when the temperature of the returning air is higher than the set point, the flow control valve corresponding to the vaporizer is opened or its opening is increased, while the other flow control valve corresponding to the condenser is closed. On the contrary to the above, when the temperature of the returning air is lower than the set point, the flow control valve corresponding to the condenser is opened or its opening is increased, while the other flow control valve corresponding to the vaporizer is closed. As described above, since the flow control valves , accordingly the flow of the thermal medium to be supplied, are controlled in accordance with the heat load in the respective air conditioners, is prevented such problem in the prior art that the required quantity of thermal medium can not be sufficiently supplied into the corresponding air conditioners, while the thermal medium exceeding the desired quantity is supplied into the corresponding air conditioners. It is to be noted that the flow control valve can be controlled in ON-OFF control or in continuous or step control. Furthermore, the flow control valve corresponding the vaporize is also controlled such that the liquid level of the thermal medium does not exceed the predetermined value. Namely, when the valve is opened and the thermal medium in liquid phase passes through the valve into the vaporizer and reaches the predetermined level, the liquid level detection means detects its condition and generate a detection signal for closing the valve. Accordingly, the supply of the thermal medium to the vaporizer is controlled based on the difference of temperature between the returning air and the set point and the detection signal from the liquid level detection means.

BRIEF DESCRIPTION OF THE DRAWINGS

This and other objects and feature of the present invention will become apparent from the following description taken in conjunction with the preferred embodiment thereof with reference to the accompanying drawings, in which:

Fig. 1 is a schematic flowchart of an air conditioning system for a building according to an embodiment of the present invention,

Fig. 2 is a diagram showing the operations of the flow control valves and the fan with respect to the temperature of the returning air returning to the air conditioners,

Fig. 3 is a diagram showing an operation of the first vaporizer and first condenser with respect to the temperature of the returning air,

Fig. 4 is an enlarged view showing an essential part of the air conditioner,

Fig. 5 is an enlarged view showing a modification of the air conditioner as shown in Fig. 4

Fig. 6 is a schematic flowchart showing a second embodiment of the present invention,

Fig. 7 is a section showing an embodiment of the flow control valve employed in the air control system as shown in Fig. 1,and

Fig. 8 is a section taken along a line VIII-VIII in Fig. 7.

DETAILED DESCRIPTION OF THE INVENTION

Before the description of the present invention proceeds, it is to be noted that like parts are designated by like reference numerals and symbols throughout the several views of the accompanying drawings.

Referring now to the drawings, there is shown in Fig. 1 an air conditioning system for a building in a schematic flowchart according to the present invention. The installations in this system are positioned according to a specification with respect to height. The air conditioners 1 are installed at the air-conditioned rooms on the different floors of the building. The air conditioner 1 includes a first condenser 2 as a heat exchanger for warming the air in the air-conditioned room and a first vaporizer 3 as a heat exchanger for cooling the air in the air-conditioned room. A second vaporizer 5, forming a thermal medium circulation line for warming together with the first condenser 2, is installed on a position lower than any of the locations where the air conditioners 1 are installed. A second condenser 4, forming a thermal medium circulation line for cooling together with the first vaporizer 3, is installed on a position higher than any of the locations where the air conditioners 1 are installed. In each thermal medium circulation line, the vaporizer and condenser are connected to each other through the corresponding heat pipes 6, 8 and 7, 9. The second condenser 4 is incorporated in a cold thermal source, wherefore an ice type thermal reservoir 10 is used, which is installed at a high place, such as a roof, whereas the second vaporizer 5 is incorporated in a hot thermal source, wherefore a hot water type thermal reservoir 11 is used, which is installed at a lower place, such as a basement of a building. The thermal source device to provide the thermal reservoirs 10 and 11 with cold energy or hot energy is a heat pump chiller 14 with an ice-making device 12 and hot water making device 13. A slurry pump 15 is provided between the ice type thermal reservoir 10 and the ice making device 12 so that ice made by the ice-making device 12 is forced into the ice type thermal reservoir 10 by the slurry pump 15. A hot water heat recovery pump 16 is provided between the hot water type thermal reservoir 11 and the hot water making device 13 to forces hot water into the thermal reservoir 11. In the drawings, 17 denotes an accumulator, 18 an expansion valve, 19 an air-heat exchanger and 20 a compressor.

The air conditioner 1, essentially, includes, other than the above described first condenser 2 and first vaporizer 3, a fan 21 and a thermistor 22 for detecting the temperature of the returning air in the air-conditioned room which is to be sucked by the fan 21. Flow control valves 23 and 24 are incorporated in the inlet of the first vaporizer 3 and the outlet of the first condenser 2. A liquid level detector 26 is mounted on a upper position of the first vaporizer 3. The detection signals from the thermistor 22 and the liquid level detector 26 are supplied into a controller 25. The controller 25 carries out the calculation based on the above signals to obtain the suitable opening of the flow control valves 23 and 24 and the suitable revolution of the fan 21, the command signals corresponding to which are generated and output into the flow control valves 23 and 24 and the fan 21.

Fig. 2 diagrammatically shows a control of the opening of the valves 24 and 23 corresponding to the first condenser 2 and the first vaporizer 3 and revolution of the fan 21 with respect to the temperature of the returning air in the air-conditioned room. In Fig. 2 the operation of the valve 24 corresponding to the first condenser 2 is illustrated on the uppermost stage, the operation of the valve 23 corresponding to the first vaporizer 3 is illustrated on the middle stage, and the operation of the fan 21 is illustrated on the lowermost stage. In this embodiment, the desired value of the temperature in the air-conditioned room is set at a set point SP. When the temperature of the returning air has lowered under the set point SP, the warming operation is started, while when the temperature of the returning air has risen above the set point SP, the cooling operation is started.

In this embodiment, the flow control valves 23 and 24 respectively consist of the ON-OFF type valves. When the temperature of the returning air is above the set point SP, the valve 24 corresponding to the first condenser 2 is closed and the valve 23 corresponding to the first vaporizer 3 is opened for cooling operation. On the contrary, the temperature of the returning air is below the set point SP, the valve 24 is opened and valve 23 is closed for the warming operation. That is, the cooling and warming operations are mutually switched around the set point SP. In this switching operation, the difference between the temperature, on which the respective valve 23 or 24 are opened, and the set point SP is larger than the difference between the temperature, on which the respective valve 23 or 24 are closed, and the set point SP. Namely, when the operation is switched from the cooling to the warming, as shown by the solid line in Fig. 2, the valve 23 is closed at a point slightly below the set point SP and subsequently the valve 24 is opened at a point further below the set point SP. On the contrary, when the operation is switched from the warming to the cooling, as shown by the two-dot chain line in Fig. 2, the valve 24 is closed at a point slightly above the set point SP and subsequently the valve 23 is opened at a point further above the set point SP. It is to be noted, here, that even if the temperature of the returning air is not sufficiently lowered to stop the cooling operation, in the case where the thermal medium is sufficiently filled in the first vaporizer 3, the liquid level detector 26 generates a detection signal to close the valve 23.

The fan 21 is controlled so as to rotate at three stages of speed, i.e. high, intermediate and low speeds. The more remote from the set point SP, the temperature of the returning air is, in both cooling and warming operations, the faster the fan 21 rotates. The solid line illustrates the switching operation from cooling to warming, while the two-dot chain line illustrates the switching operation from warming to cooling. The switching points where the revolution of the fan is switched from higher speed to lower speed, is closer to the set point SP than the switching points where the revolution of the fan is switched from lower speed to higher speed is. In addition, the fan 21 is so controlled that when the valve 24 corresponding to the first condenser 2 and the valve 23 corresponding to the first vaporizer 3 are respectively opened, the revolution of the fan 21 is, substantially the same time, switched from low speed to intermediate speed.

Alternatively, as shown in Fig. 3, it is possible that the revolution of the fan 21 is maintained constant and only the opening of the valves 23 and 24 are controlled so as to proportionally increase in accordance with the divergence of the temperature of the returning air with respect to the set point SP. It is to be noted that in Fig. 3, the solid line illustrates the operation of the valve 23, while the broken line illustrates the operation of the valve 24.

Fig. 4 is an enlarged view of the air conditioner 1 showing the configuration of the first vaporizer 3 and the liquid level detection means in detail, the first condenser, fan and thermistor being omitted. The first vaporizer 3 comprises a lower header 3b which is connected with the heat pipe 7 for thermal medium in liquid phase, the flow control valve 23 being incorporated in the heat pipe at a position adjacent to the lower header 3b, a coil 3c connected with the lower header 3b and constituting the essential portion of heat exchanger, and a upper header 3a which is connected with both the coil 3c and the heat pipe 9 for thermal medium in vapor phase. The liquid level detection means comprises a bypass 7a, which is connected with both the heat pipes 7 and 9, and a liquid level detection switch 26 which is mounted on the bypass 7a at a predetermined height to detect the liquid level of thermal medium in the bypass 7a and accordingly in the coil 3c. When the switch 26 detects the status in which the thermal medium in liquid phase reaches the predetermined level, the switch 26 generates a detection signal and supply it to the controller 25 to close the valve 23.

The flow control valve 22 consist of, as one embodiment, an electromagnetic valve which is of a type of the switch 26 is mounted on the bypass 7a at such height that it detects such a condition in which the thermal medium in liquid phase reaches substantially over the uppermost fin 3d. It is to be noted, here, that, in order to obtain the effective performance of cooling, it is importance that the coil 3c, should be filled by the thermal medium in liquid phase up to the level corresponding to the uppermost fin. Furthermore, it is, also, importance that the inlets 7b of the heat pipe 9 for vapor, which are opened to the interior of the upper header 3a, should not be occupied by the thermal medium in liquid phase, in other words, at least the upper portion of the header 7b should be occupied by the vapor. If the inlets 7b are occupied by the thermal medium in liquid phase, the vapor, which has been vaporized in the coil 3a and the upper header 3a, can not flow quickly into the heat pipe 9.

The valve 23 is closed when the air conditioner 1 is not operated. The valve 23 is also closed when the temperature detected by the thermistor 22 is below the set point SP even if the liquid level of the thermal medium is below the predetermined level which is should be detected by the switch 26.

Alternatively, the first vaporizer 3 may be mounted in the housing of the air conditioner 1 along a inclined plane, as shown in Fig. 5.

In the embodiment as shown in Fig. 1, each air conditioner 1 is provided with a liquid level detection means. On the contrary to the above, as shown in Fig. 6, wherein the first condenser 2 is omitted, a group of air conditioners on the same floor of a building can be controlled by a single liquid level detection means if all of the air conditioners can be operated in the same mode. In this embodiment, a common bypass 7a' are connected with both the heat pipes 7 and 9. An additional ON-OFF type flow control valve 23a for supplying the thermal medium in liquid to every air conditioners is incorporated in the suitable portion of the heat pipe 7. The signals generated in the liquid level detection switch 26 and the respective thermistors 22 are supplied into the controller, not shown in Fig. 6, by which the valves 22 and 23 are controlled. In this embodiment, the liquid level of the thermal medium in the respective vaporizers 3 are represented by the liquid level of the thermal medium in the bypass 7a'. Accordingly, the liquid level of the thermal medium in the respective vaporizers 3 are basically controlled by the signal from the switch 26.

In the embodiments as described above, the liquid level detection switch 26 is employed to detect the liquid level of the thermal medium in the first vaporizer 3. On the contrary to the above, it is possible to detect the liquid level of the thermal medium in the vaporizer 3 by such system that comprises a pair of temperature sensors (not shown) respectively incorporated in the inlet and outlet of the vaporizer 3 for thermal medium. In detail, if the thermal medium in liquid phase does not reach the predetermined level, i.e. the outlet of the vaporizer 3, the temperature of the vapor, which is overheated and passes through the outlet, is higher than that of the thermal medium in liquid phase which is detected by the lower sensor mounted on the inlet of the vaporizer 3. On the contrary to the above, when the thermal medium in liquid phase reaches the predetermined level of the inlet in the vaporizer 3 so that the upper sensor detects the temperature of the thermal medium in liquid phase which is the same as that of thermal medium to be detected by the lower sensor. Accordingly, by comparing, with each other the both temperatures detected by the upper and lower sensors, liquid level detection can be carried out.

With the system as described above, since the opening of the valves 23 and 24 are, respectively, controlled based on the differences between the actual temperature of the returning air and the set point, that is, since the flow of the thermal medium is controlled in such a manner that the resistance to the flow of the thermal medium varies depending to the opening of the valves 23 and 24 instead of depending to the amount of heat exchange, i.e. amount of vapor, necessary flow of thermal medium can be supplied to the air conditioners which requires the supply of the thermal medium. Furthermore, even if a plurality of air conditioners are respectively installed at different height, i.e. on different floors of a building, the liquid level of thermal medium in the first vaporizer 3 is controlled by the liquid level detection means and the flow control valve 23 so as to be maintained constant, the water head of the thermal medium in the respective air conditioners is maintained constant. Therefore, in the respective air conditioners on the different floors of the building the thermal medium in liquid phase in the corresponding vaporizers can be vaporized at substantially the same temperature under substantially the same pressure, resulting in the same cooling performance.

In addition, with the above system, it is possible to effect the dehumidifying operation in such a manner that the supercooling dehumidification is carried out by the first vaporizer 3 and the reheat is carried out by the first condenser 2. In this case, both the valves 23 and 24 are opened.

Alternatively, in Fig. 1, when the water head of the thermal medium at the lower air conditioners is extremely high, it is preferable to incorporate a pressure reducing valve in each branch heat pipe for liquid at the upstream of the valve 23.

Figs. 7 and 8 show an example of the flow control valves 23 which can be preferably adapted to the above air conditioning system. This electromagnetic valve 23d is incorporated in the heat pipe 7 for thermal medium in liquid phase. The valve 23d has a housing 23f which extend downwardly from the horizontal heat pipe 7. A valve 23j is accommodated in the housing 23f so as to be movable upwardly and downwardly. The valve comprises a piston member or support member 23g and a flexible bag 23i which is secured on the top of the support member 23g and in which magnetic liquid 23h is filled. A semicircular electromagnet 23e is secured on the outer and upper periphery of the heat pipe 7 at a position above the valve 23j.

According to the above valve 23j, when the electromagnet 23e is excited, the magnetic liquid 23h is attracted by the excited electromagnet 23i so that the entire valve 23j moves upwardly from the housing 23f to close the heat pipe 7. The bag 23i, which is flexible, is brought into a tight contact with the inner peripheral surface, i.e. the sealing surface, of the heat pipe 7 so that the heat pipe 7 is sealed tight. As shown in Figs. 7 and 8, since the support member 23g remains in the housing 23f when the valve 23j closes the heat pipe 7, the entire valve 23j is prevented from being moved away from the predetermined position by the flow pressure of the thermal medium. Even if some foreign matter, such as dust, is sandwiched between the bag 23i of the valve 23j and the sealing surface of heat pipe 7, the bag 23i can be easily deformed to fit the outline of the foreign matter so that tight sealing can be effected. When the excitation of the electromagnet 23e is ceased, the valve 23i is released from the electromagnet 23e so that the valve 23i drops into the housing 23f under its gravity, resulting in the opening of the heat pipe 7. When the entire valve 23j is accommodated in the housing 23f, the opened passage of the heat pipe can have the same diameter as that of the heat pipe 7 per se. Accordingly, it is advantageous that the pressure loss at the valve is smaller than that of the normal electromagnetic valve.

Claims

1. An air conditioning system for buildings which comprises:

a cold thermal source which is installed at a high place in said building,

a hot thermal source which is installed at a low place in said building,

air conditioners which are installed at various heights between said cold and hot thermal sources and each of which has, respectively, a condenser as a heat exchanger for warming air in air-conditioned location, a vaporizer as a heat exchanger for cooling the air in the air-conditioned location and a fan for blowing air into the air-conditioned location through said condenser and vaporizer,

a cooling heat pipe of a gravity type by which the vaporizers of said air conditioners is connected with said cold thermal source, and in which thermal medium is filled so as to circulate therein,

a warming heat pipe of a gravity type by which the condensers of said air conditioners is connected with said hot thermal source and in which thermal medium is filled so as to circulate therein,

temperature sensors for detecting temperature of the air in the air-conditioned locations,

flow control valves which are respectively incorporated in the heat pipes at inlets of said vaporizers and at outlets of said condensers,

liquid level detection means for detecting a predetermined level of the thermal medium in liquid phase in the vaporizer, and

controllers which control, respectively, the flow control valves base on detection signals supplied from the temperature sensors and the liquid level detection means so that the air in the air-conditioned locations meet a set point.

2. An air conditioning system as claimed in claim 1, wherein revolution of the fan is controlled by said controller based on the detection signal from said temperature sensor.

3. An air conditioning system as claimed in claim 1, wherein said liquid level detection means comprises;

a bypass which is connected with the cooling heat pipe for thermal medium in liquid phase and with the cooling heat pipe for thermal medium in vapor phase in parallel with said vaporizer, and

a liquid level detection switch for detecting level of thermal medium in liquid phase in said bypass.

4. An air conditioning system as claimed in claim 3, wherein said liquid level detection switch is mounted on said bypass so as to detect such a condition that the thermal medium in liquid phase comes substantially over uppermost fin of said vaporizer.

5. An air conditioning system as claimed in claim 1, wherein said flow control valve comprises:

a housing downwardly extending from said heat pipe for thermal medium in liquid phase which extends horizontally,

an electromagnet surrounding a periphery of said heat pipe at a position above said housing, and

a valve which is accommodated in said housing so as to be movable upwardly and downwardly in said housing and which comprises a support member and a flexible bag secured on a top of said support member and accommodating magnetic liquid therein,

whereby, when said electromagnet is excited, said valve is attracted by said electromagnet so that said bag is brought into a tight contact with an inner periphery of said heat pipe to close said heat pipe.

Fig. 1

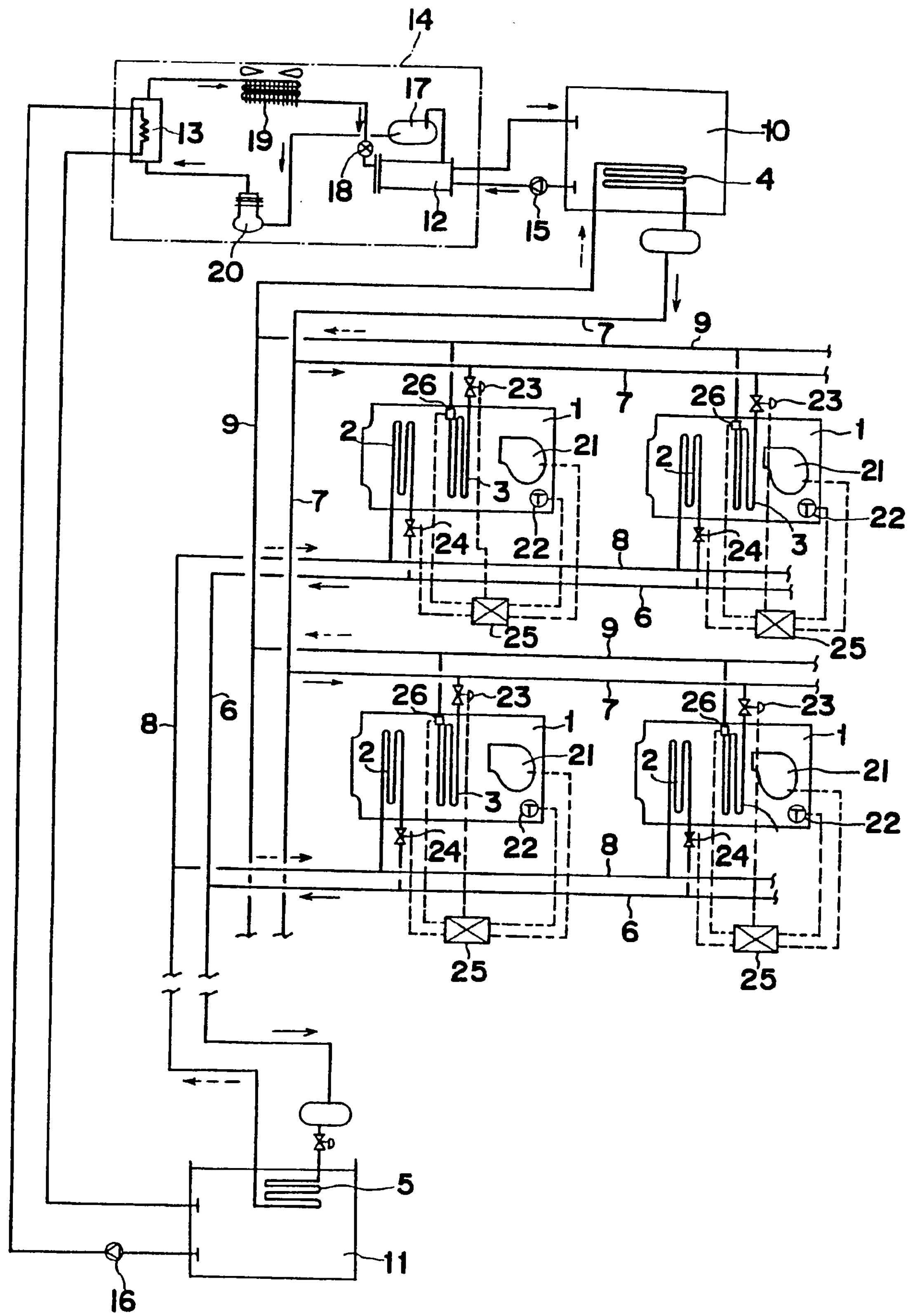

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Fig. 2

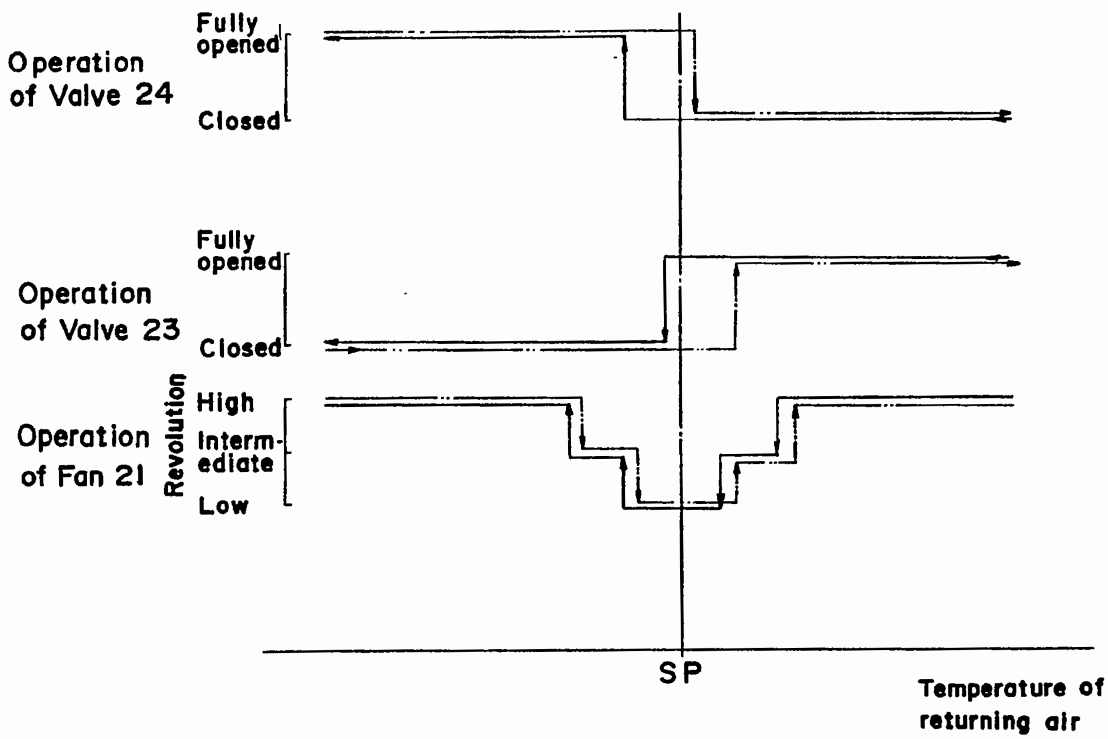

Operation of Valve 24
Fully opened
Closed
Operation of Valve 23
Fully opened
Closed
Operation of Fan 21
Revolution
High
Interm-ediate
Low
SP
Temperature of returning air

Fig. 3

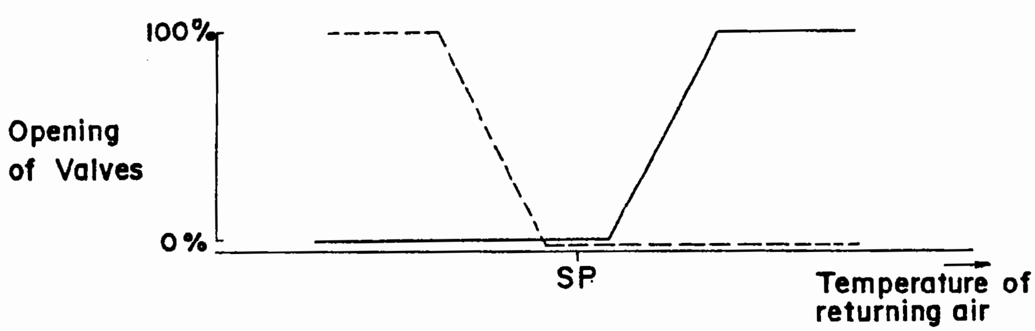

100%
Opening of Valves
0%
SP
Temperature of returning air

Fig. 4

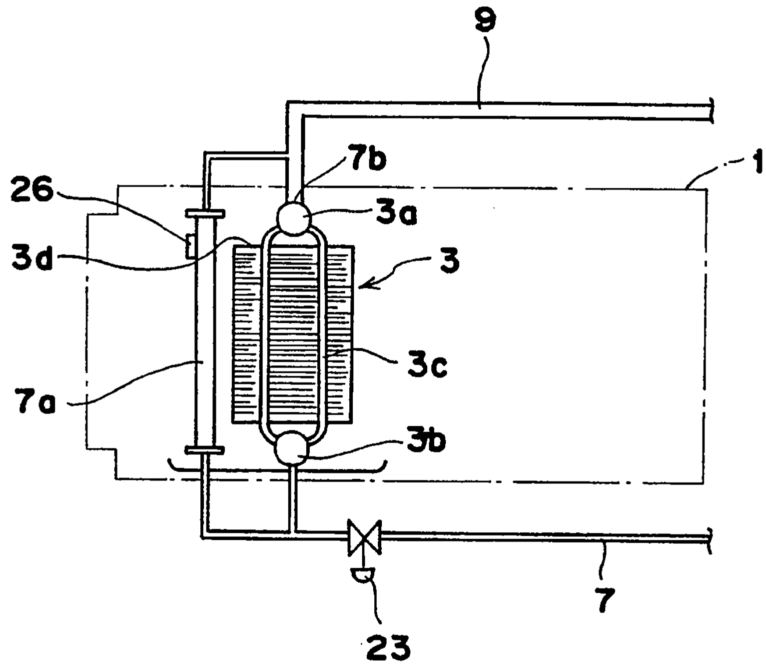


Fig. 5

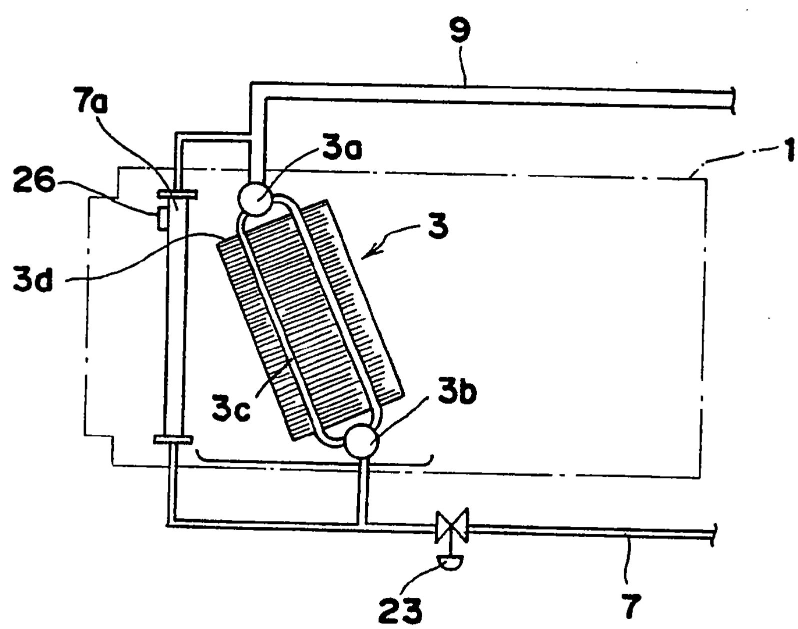


Fig. 6

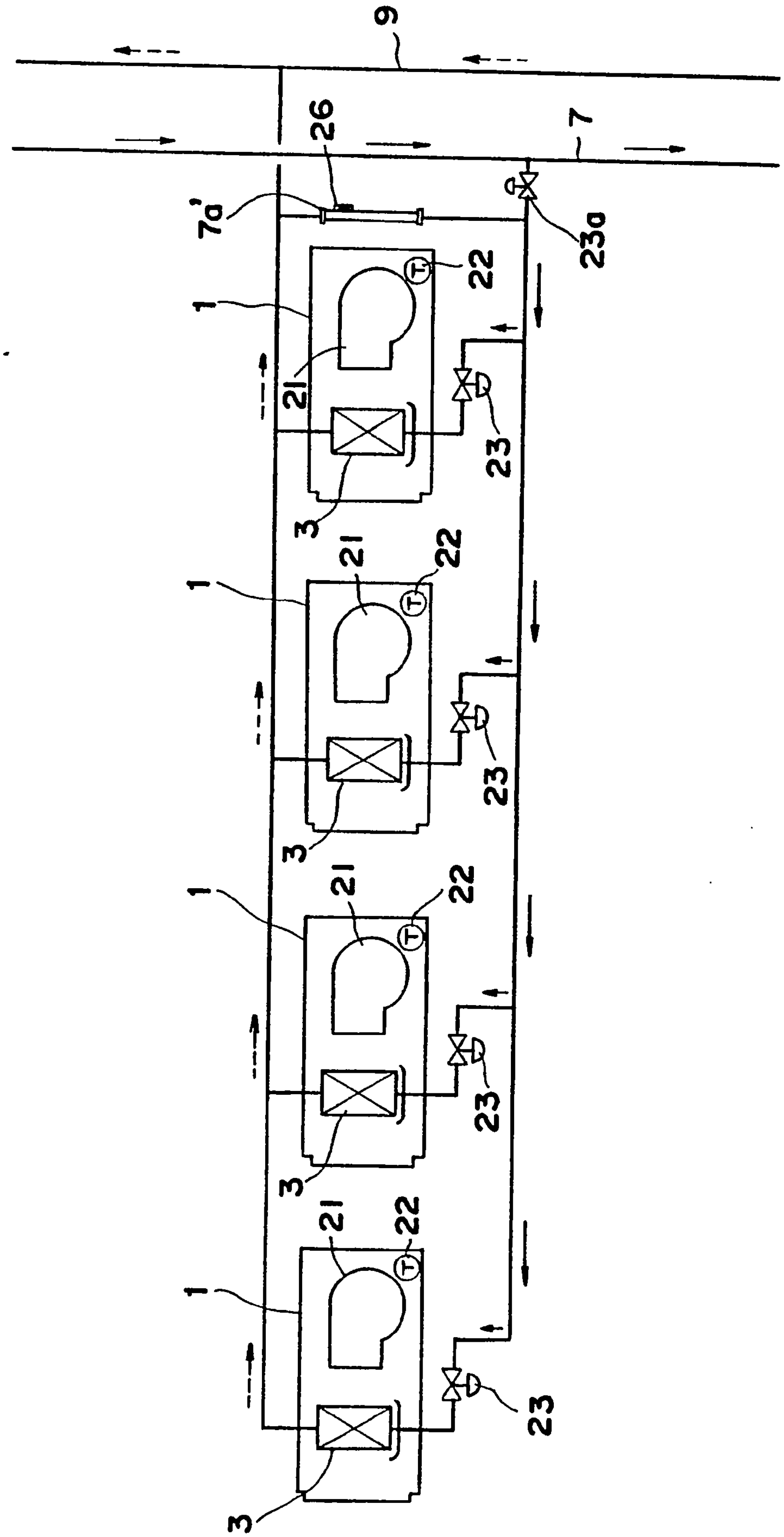


Fig. 7

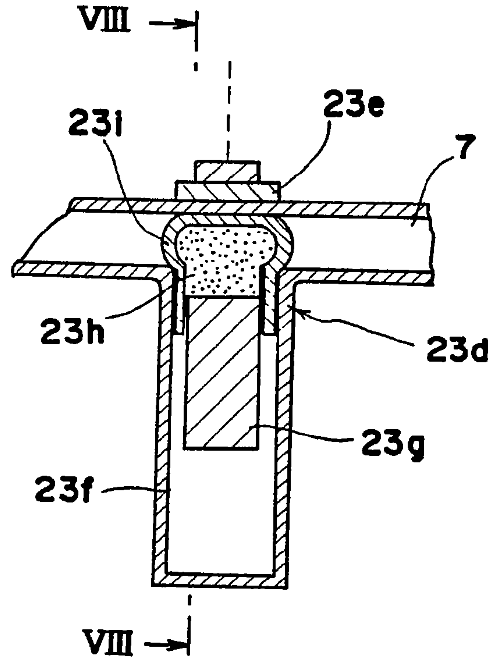
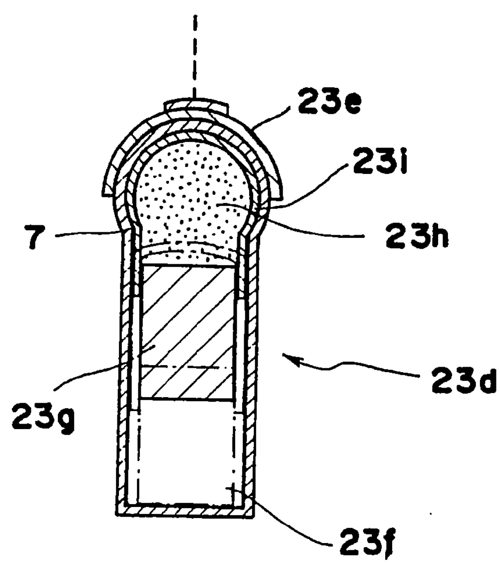


Fig. 8



(19) Europäisches Patentamt
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EP 0 281 762 A3

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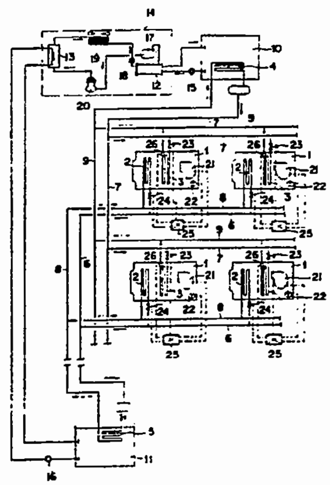
(54) Air conditioning system for buildings.

(57) The system includes a plurality of air conditioners (1) on the different floors, a cooling heat pipe (7, 9) of a gravity type for connecting a vaporizer (3) in the air conditioners with a cold thermal source (10) installed on a top of the building and a warming heat pipe (6, 8) of a gravity type for connecting a condenser (2) of the air conditioners with a hot thermal source (11) installed on a basement of the building. The heat pipes are, respectively, provided with flow control valves (23, 24), at inlet to the vaporizer and the outlet from the condenser. Each air conditioner is provided with a thermistor (22) for detecting the temperature of the returning air and a liquid level detection switch (26) for detecting the liquid level of the thermal medium in the vaporizer. The flow control valves are respectively controlled based on the detection signals from the thermistor and the liquid level detection switch.

Fig. 1



European Patent Office

EUROPEAN SEARCH REPORT

Application Number

EP 88 10 1622

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int. Cl.4)
A	US-A-4 237 859 (GOETTL) * Column 6, lines 15-63; figure 1 * ---	1	F 24 F 3/08 F 24 F 11/06 F 24 F 5/00 F 24 D 11/02 F 16 K 31/06
A	US-A-3 127 929 (RINGQUIST) * Column 2, lines 59-71; figure * ---	1	
A	DE-A-2 403 059 (VEB KOMBINAT LUFT- UND KÄLTETECHNIK) ---		
E	EP-A-0 266 680 (TAKENAKA KOMUTEN) * Whole document * -----	1	
			TECHNICAL FIELDS SEARCHED (Int. Cl.4)
			F 24 F F 24 D

The present search report has been drawn up for all claims

Place of search	Date of completion of the search	Examiner
THE HAGUE	16-03-1989	PESCHEL G.

CATEGORY OF CITED DOCUMENTS

X : particularly relevant if taken alone
Y : particularly relevant if combined with another document of the same category
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(11) **EP 1 143 778 A1**

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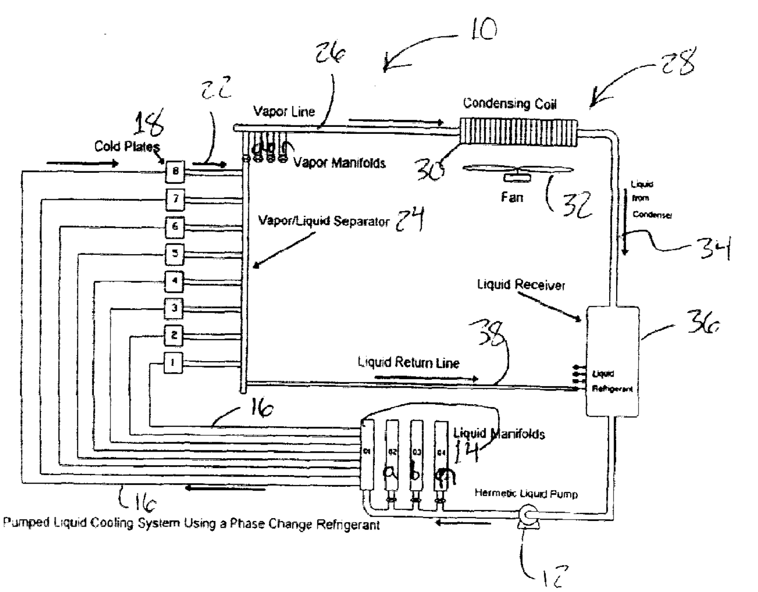
(54) **Pumped liquid cooling system using a phase change refrigerant**

(57) An improved cooling system provides cooling away from the surface of electrical and electronic components with very low parasitic power consumption and very high heat transfer rates. The component to be cooled is in thermal contact with a cold plate evaporator device. Refrigerant is circulated by a liquid refrigerant pump to the cold plate evaporator device, and the liquid refrigerant is at least partially evaporated by the heat generated by the component. The vapor is condensed by a conventional condenser coil and the condensed liquid along with any unevaporated liquid is returned to the pump. The system operates nearly isothermally in both evaporation and condensation.



Pumped Liquid Cooling System Using a Phase Change Refrigerant

Fig 1

EP 1 143 778 A1

Description

Related Applications

[0001] This is a regularly filed application, based on provisional application Serial No. 60/194,381, filed April 4, 2000.

Technical Field

[0002] The present invention relates to cooling of electrical and electronic components, and more particularly, to a liquid refrigerant pump to circulate refrigerant to multiple cold plate/evaporators in thermal contact with the electrical or electronic component to be cooled.

Background of the Invention

[0003] Electrical and electronic components (e.g. microprocessors, IGBT's, power semiconductors etc.) are most often cooled by air-cooled heat sinks with extended surfaces, directly attached to the surface to be cooled. A fan or blower moves air across the heat sink fins, removing the heat generated by the component. With increasing power densities, miniaturization of components, and shrinking of packaging, it is sometimes not possible to adequately cool electrical and electronic components with heat sinks and forced air flows. When this occurs, other methods must be employed to remove heat from the components.

[0004] One method for removing heat from components when direct air-cooling is not possible uses a single-phase fluid which is pumped to a cold plate. The cold plate typically has a serpentine tube attached to a flat metal plate. The component to be cooled is thermally attached to the flat plate and a pumped single-phase fluid flowing through the tube removes the heat generated by the component.

[0005] There are many types of cold plate designs, some of which involve machined grooves instead of tubing to carry the fluid. However all cold plate designs operate similarly by using the sensible heating of the fluid to remove heat. designs operate similarly by using the sensible heating of the fluid to remove heat. The heated fluid then flows to a remotely located air-cooled coil where ambient air cools the fluid before it returns to the pump and begins the cycle again. This method of using the sensible heating of a fluid to remove heat from electrical and electronic components is limited by the thermal capacity of the single phase flowing fluid. For a given fluid to remove more heat, either its temperature must increase or more fluid must be pumped. This creates high temperatures and/or large flow rates to cool high power microelectronic devices. High temperatures may damage the electrical or electronic devices, while large flow rates require pumps with large motors which consume parasitic electrical power and limit the application of the cooling system. Large flow rates may also cause erosion of the metal in the cold plate due to high fluid velocities.

[0006] Another method for removing heat from components when air-cooling is not feasible uses heat pipes to transfer heat from the source to a location where it can be more easily dissipated. Heat pipes are sealed devices which use a condensable fluid to move heat from one location to another. Fluid transfer is accomplished by capillary pumping of the liquid phase using a wick structure. One end of the heat pipe (the evaporator) is located where the heat is generated in the component, and the other end (the condenser) is located where the heat is to be dissipated; often the condenser end is in contact with extended surfaces such as fins to help remove heat to the ambient air. This method of removing heat is limited by the ability of the wick structure to transport fluid to the evaporator. At high thermal fluxes, a condition known as "dry out" occurs where the wick structure cannot transport enough fluid to the evaporator and the temperature of the device will increase, perhaps causing damage to the device. Heat pipes are also sensitive to orientation with respect to gravity. That is, an evaporator which is oriented in an upward direction has less capacity for removing heat than one which is oriented downward, where the fluid transport is aided by gravity in addition to the capillary action of the wick structure. Finally, heat pipes cannot transport heat over long distances to remote dissipaters due once again to capillary pumping limitations.

[0007] Yet another method which is employed when direct air-cooling is not practical uses the well-known vapor compression refrigeration cycle. In this case, the cold plate is the evaporator of the cycle. A compressor raises the temperature and pressure of the vapor, leaving the evaporator to a level such that an air-cooled condenser can be used to condense the vapor to its liquid state and be fed back to the cold plate for further evaporation and cooling. This method has the advantage of high isothermal heat transfer rates and the ability to move heat considerable distances. However, this method suffers from some major disadvantages which limit its practical application in cooling electrical and electronic devices. First, there is the power consumption of the compressor. In high thermal load applications the electric power required by the compressor can be significant and exceed the available power for the application. Another problem concerns operation of the evaporator (cold plate) below ambient temperature. In this case, poorly insulated surfaces may be below the dew point of the ambient air, causing condensation of liquid water and creating the opportunity for short circuits and hazards to people. Vapor compression refrigeration cycles are designed so as not to return any liquid refrigerant to the compressor which may cause physical damage to the compressor and shorten its life by diluting its lubricating oil. In cooling electrical and electronic components, the thermal load can be highly variable, causing unevaporated refrigerant to exit the cold plate and enter the compressor. This can cause damage and shorten the life of the compressor. This is yet another disadvantage of vapor compression cooling of components.

[0008] It is seen then that there exists a continuing need for an improved method of removing heat from components when air-cooling is not feasible.

Summary of the Invention

[0009] This need is met by the pumped liquid cooling system of the present invention wherein cooling is provided to electrical and electronic components with very low parasitic power consumption and very high heat transfer rates away from the component surface. This invention also reduces the temperature drop required to move heat from the component to the ambient sink.

[0010] In accordance with one aspect of the present invention, a liquid refrigerant pump circulates refrigerant to cold plate/evaporators which are in thermal contact with the electrical or electronic component to be cooled. The liquid refrigerant is then partially or completely evaporated by the heat generated by the component. The vapor is condensed by a conventional condenser coil, and the condensed liquid, along with any unevaporated liquid, is returned to the pump. The system of the present invention operates nearly isothermally in both evaporation and condensation.

[0011] Accordingly, it is an object of the present invention to provide cooling to electrical and electronic components. It is a further object of the present invention to provide such cooling to components with very low parasitic power consumption and very high heat transfer rates away from the component surface. It is yet another object of the present invention to reduce the temperature drop required to move heat from the component to the ambient sink.

[0012] Other objects and advantages of the invention will be apparent from the following description, the accompanying drawings and the appended claims.

Brief Description of the Drawings

[0013]

Fig. 1 is a schematic block diagram illustrating the pumped liquid cooling system in accordance with the present invention; and

Fig. 2 illustrates a plurality of cold plate evaporator devices, each in thermal contact with a component to be cooled.

Detailed Description of the Preferred Embodiments

[0014] Referring now to Fig. 1, there is illustrated a cooling system 10 which circulates a refrigerant as the working fluid. The refrigerant may be any suitable vaporizable refrigerant, such as R-134a. The cooling cycle can begin at liquid pump 12, shown as a Hermetic Liquid Pump. Pump 12 pumps the liquid phase refrigerant to a liquid manifold 14 , where it is distributed to a plurality of branches or lines 16. Additional liquid manifolds 14a, 14b and 14n are shown to indicate where more branches (or lines) could be attached. The actual number of branches will depend on the number of components to be cooled by the system. From the manifold 14, each branch or line 16 feeds liquid refrigerant to a cold plate 18.

[0015] As illustrated in Fig. 2, each cold plate 18 is in thermal contact with an electrical or electronic component or components 20 to be cooled, causing the liquid refrigerant to evaporate at system pressure. None, some, or all of the liquid refrigerant may evaporate at cold plate 18, depending on how much heat is being generated by component 20. In most cases, some of the refrigerant will have evaporated and a two-phase mixture of liquid and vapor refrigerant will leave each cold plate 18, as shown by arrow 22.

[0016] In a preferred embodiment of the present invention, at this point in the operation of the system, each cold plate 18 discharges its mixture of two-phase refrigerant to vapor/liquid separator 24, as illustrated in Fig. 1. For most applications, the vapor/liquid separator 24 is a vertical tube of sufficient diameter to allow the heavier liquid refrigerant to fall to the bottom of the tube by gravity, while the lighter vapor rises to the top of the tube. In this manner, any unevaporated refrigerant is separated from the vapor and each phase may be treated separately within the system.

[0017] The vapor/liquid separator 24 is attached to a vapor line 26 leading to condenser 28, comprised of a condensing coil 30 and a fan 32. Additional vapor/liquid separators 24a, 24b, and 24n, may be connected through the use of vapor manifolds so that the cooling capacity of the system may be increased. Condenser coil 30, attached to vapor line 26, condenses the vapor phase back to a liquid and removes the heat generated by the electronic components 20. In Fig. 1, an ambient air-cooled condenser 28 is shown, using fan 32, although it will occur to those skilled in the art that any suitable form of heat rejection may be used without departing from the scope of the invention, such as an air cooled condenser, a water or liquid cooled condenser, or an evaporative condenser.

[0018] The condenser 28 operates at a pressure which corresponds to a temperature somewhat higher than the temperature of the ambient air. In this way, it is impossible for condensation to form, since no system temperature will be below the ambient dew point temperature. The condenser operating point sets the pressure of the entire system by means of the entering coolant temperature and its ability to remove heat from the condenser, thus fixing the condensing temperature and pressure. Also, since vaporized refrigerant is being condensed to a liquid phase, the condenser 28 sets up a flow of vaporized refrigerant from the vapor/liquid separator 24 into the condenser 28, without the need for any compressor to move the vapor from the cold plate-evaporator 18 to the condenser 28. The liquid refrigerant exits the condenser 28, as indicated by arrow 34, and moves by gravity to a liquid receiver 36, which holds a quantity of liquid refrigerant.

[0019] In one embodiment of the invention, connected to the liquid receiver 36 is a second and optional liquid return line 38 from the vapor/liquid separator 24. Alternatively, all liquid can be returned to the pump 12 via line 26, passing through the condenser 28 to change vapor back to liquid. With the addition of liquid return line 38, there are two sources of liquid refrigerant. One source of liquid refrigerant is from the condenser and the other is from the separator. Either line 26, or line 38, or both, can be used to carry any unevaporated liquid refrigerant from the separator 24 to the liquid receiver 36, where it may be used again in the cycle. The liquid receiver, therefore, can receive liquid from the condenser or from the separator. The quantity of liquid refrigerant held in the liquid receiver 36 provides a liquid head over the inlet of the pump 12 so the pump operates reliably. The liquid receiver 36 also handles changes in the amount of liquid refrigerant in the system 10 by providing a reservoir to store refrigerant. The outlet of the liquid receiver is connected to the inlet of the liquid refrigerant pump 12. At the pump 12, the pressure of the refrigerant is raised sufficiently to overcome the frictional losses in the system and the cooling cycle begins again. The pump 12 is selected so that its pressure rise is equal to or exceeds the frictional loss in the system at the design flow rate.

[0020] Unlike the pumped liquid single-phase system, the present invention operates isothermally, since it uses change of phase to remove heat rather than the sensible heat capacity of a liquid coolant. This allows for cooler temperatures at the evaporator and cooler components than a single-phase liquid system. Low liquid flow rates are achieved through the evaporation of the working fluid to remove heat, keeping the fluid velocities low and the pumping power very low for the heat removed. Parasitic electric power is reduced over both the pumped single-phase liquid system and the vapor compression refrigeration system.

[0021] An advantage over the heat pipe system is obtained with the system 10 of the present invention because the liquid flow rate does not depend on capillary action, as in a heat pipe, and can be set independently by setting the flow rate of the liquid pump. Dry out can thus be avoided. The cold plate/evaporator system of the present invention is insensitive to orientation with respect to gravity. Unlike heat pipe systems, the thermal capacity of the evaporator 18 of the present invention does not diminish in certain orientations.

[0022] Another advantage of the present invention over heat pipe and vapor compression based systems is the ability to separate the evaporator and condenser over greater distances. This allows more flexibility in packaging systems and design arrangements. In accordance with the present invention, liquid and vapor are transported independently, allowing for optimization of liquid and vapor line sizes. The present invention easily handles variation in thermal load of the components 20 to be cooled. Since any unevaporated liquid refrigerant is returned to the pump, multiple cold plates at varying loads are easily accommodated without fear of damaging a compressor. Since the current invention does not operate at any point in the system 10 at temperatures below ambient dew point temperature, there is no possibility of causing water vapor condensation and the formation of liquid water.

[0023] Having described the invention in detail and by reference to the preferred embodiment thereof, it will be apparent that other modifications and variations are possible without departing from the scope of the invention defined in the appended claims.

Claims

1. An improved cooling system comprising:

   at least one component generating heat and required to be cooled;
   at least one cold plate evaporator device in thermal contact with the at least one component;
   a liquid refrigerant pump;
   a vaporizable refrigerant circulated by the liquid refrigerant pump to the at least one cold plate evaporator device, whereby the refrigerant is at least partially evaporated by the heat generated by the at least one component, creating a vapor;
   a condenser for condensing the vapor, creating a condensed liquid; and
   at least one return line to return liquid refrigerant to the pump.

2. An improved cooling system as claimed in claim 1 further comprising a separator means for separating the unevaporated liquid refrigerant from the vapor.

3. An improved cooling system as claimed in claim 2 wherein the at least one return line comprises an unevaporated liquid refrigerant return line and a liquid return line from the condenser.

4. An improved cooling system as claimed in claim 1 further comprising at least one liquid manifold for receiving the refrigerant from the liquid refrigerant pump.

5. A method for cooling one or more electrical or electronic components generating heat and required to

be cooled, the method comprising the steps of:

locating at least one cold plate evaporator device in thermal contact with the one or more electrical or electronic components;
providing a liquid refrigerant pump;
providing a refrigerant;
using the liquid refrigerant pump to circulate refrigerant to the at least one cold plate evaporator device, whereby the refrigerant is at least partially evaporated by the heat generated by the one or more electrical or electronic components, creating a vapor;
condensing the vapor to create a condensed liquid; and
providing at least one return line to return liquid refrigerant to the pump.

6. A method as claimed in claim 5 further comprising the step of providing a separator means for separating the unevaporated liquid refrigerant from the vapor.

7. A method as claimed in claim 5 wherein the step of providing at least one return line comprises the steps of:

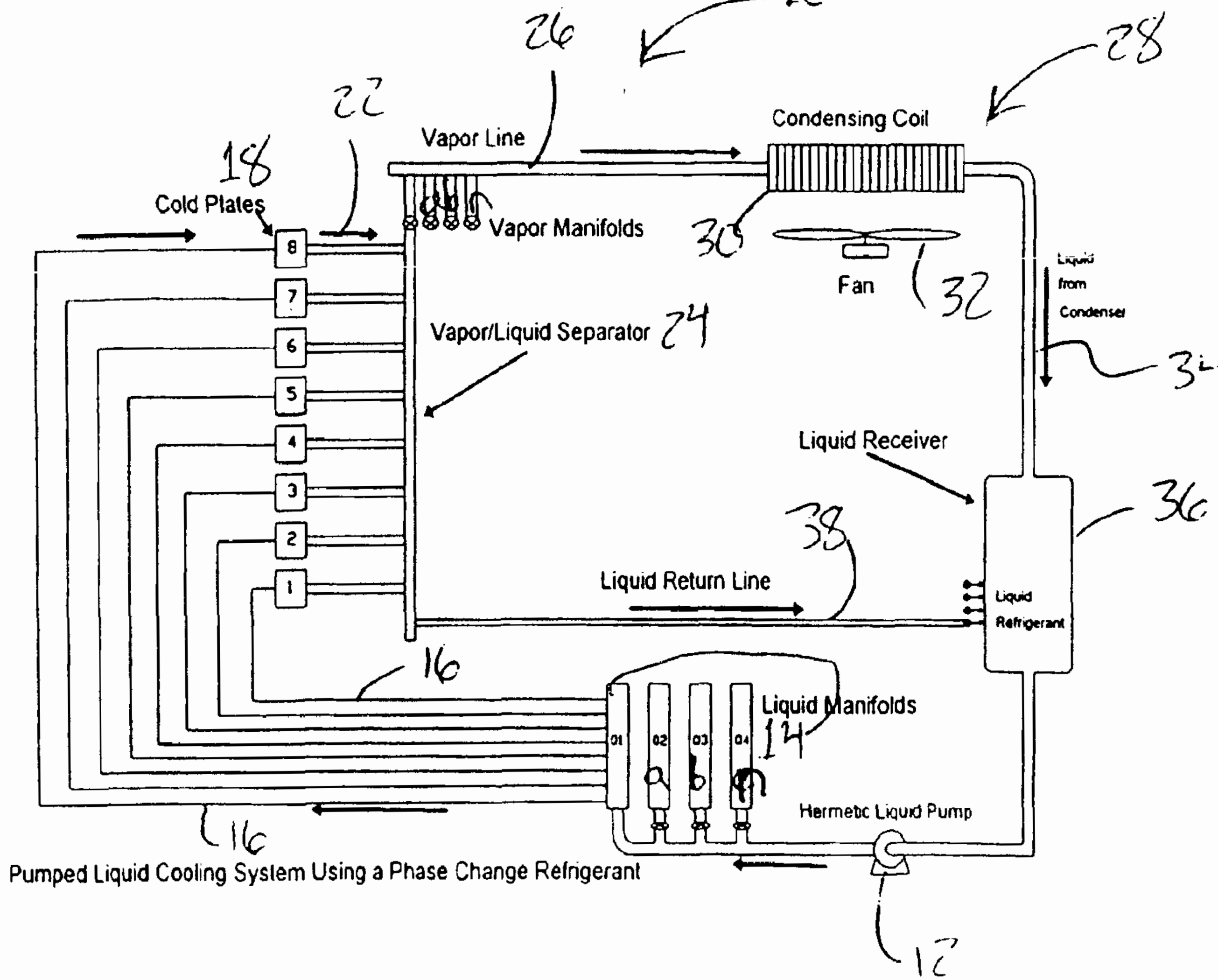
providing an unevaporated liquid return line; and
providing a liquid return line from the condenser.

8. A method as claimed in claim 7 further comprising the steps of;

providing a liquid receiver for receiving the liquid refrigerant; and
returning the liquid refrigerant to the liquid refrigerant pump.

9. A method as claimed in claim 8 wherein the liquid receiver receives unevaporated liquid from the unevaporated liquid return line and receives condensed liquid from the condenser liquid return line.

10. A method as claimed in claim 5 further comprising the step of providing at least one liquid manifold for receiving the refrigerant from the liquid refrigerant pump.



Pumped Liquid Cooling System Using a Phase Change Refrigerant

Fig 1

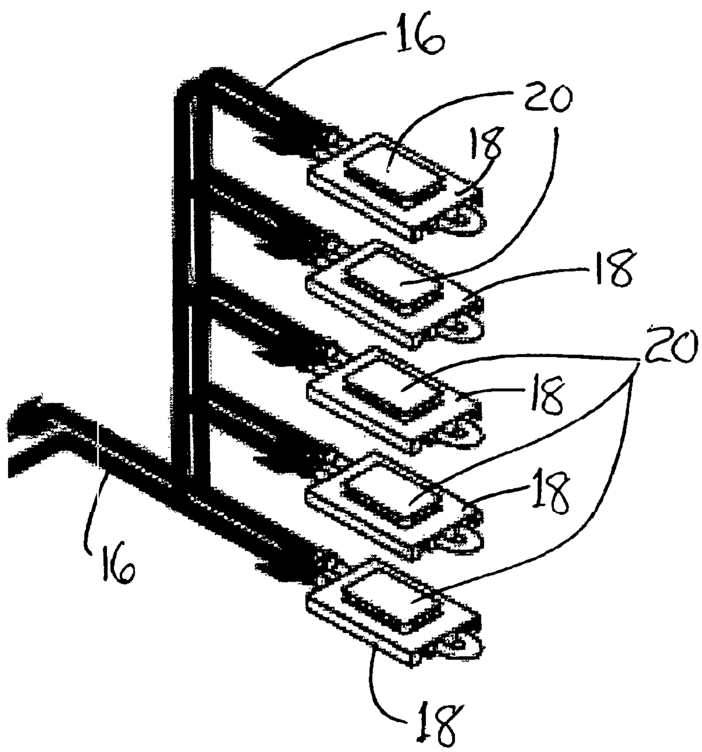


Fig. 2

European Patent Office

EUROPEAN SEARCH REPORT

Application Number
EP 01 30 3042

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int.Cl.7)
X	FR 2 071 964 A (IBM) 24 September 1971 (1971-09-24) * figures 1,1A * * page 2, line 40 * * page 4, line 1,2 * * page 4, line 32-34 * -----	1-10	H05K7/20 F25B23/00 F28D15/02 H01L23/427
			TECHNICAL FIELDS SEARCHED (Int.Cl.7)
			H05K F25B F28D H01L

The present search report has been drawn up for all claims

Place of search	Date of completion of the search	Examiner
THE HAGUE	20 July 2001	Castagné, O

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ANNEX TO THE EUROPEAN SEARCH REPORT ON EUROPEAN PATENT APPLICATION NO. EP 01 30 3042

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20-07-2001

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		DE 2056699 A	24-06-1971
		GB 1319091 A	31-05-1973
		US 3586101 A	22-06-1971

EPO FORM P0459

For more details about this annex : see Official Journal of the European Patent Office, No. 12/82

EMERGING COOLING REQUIREMENTS & SYSTEMS IN TELECOMMUNICATIONS SPACES

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ABSTRACT

During the last several years, power density trends, and consequently thermal density trends in telecommunications spaces have become topics of increasing interest. This paper will identify several of the underlying drivers of these trends, project possible outcomes, and assess the impact on cooling system design for these spaces.

INTRODUCTION

The installation of Servers, Fiber Optic Switches, and other Internet required and related devices within traditional Central Offices, and the creation of the Internet Data Center as a new communication category, are increasing the power and heat density of these spaces. To illustrate this trend, the microprocessor can be used as one example of the level and direction of power usage. Figure 1 plots the three primary drivers of microprocessor power & density: the number of transistors per processor, the processor clock speed, and track spacing. Transistors turning on and off consume power, speed determines how often they turn on and off, and track spacing establishes the transistor density. Since 1975 transistors per processor have increased by approximately a factor of 4,000, clock frequency by a factor of 600, and spacing has decreased by a factor of 50. Note that the vertical scale is logarithmic. Plotting these curves on a linear scale, Figure 2, shows more dramatically the recent increases. The Pentium 4 microprocessor now has 42,000,000 transistors, is approaching 2GHz in speed, has a track spacing of 0.12 micron, and consumes approximately 50 watts at full power. Recent statements from Intel such as "The number of transistors on our processors should pass 200 million by 2005, and reach well in excess of 1 billion by the end of the decade", and recent announcements by Sandia Labs of success in achieving a track spacing of 0.07 micron using Extreme Ultraviolet Lithography (EUVL), give a high degree of probability that these trends will continue in the near term. Based on this, Figure 3 shows projected microprocessor power usage. The International Technology Roadmap for Semiconductors projects microprocessor "maximum power" to be 170 watts in 2005.

These, and other trends have prompted a group of computer and communications providers to issue a report entitled "Heat Density Trends in Data Processing Computer Systems, and Telecommunications Equipment", projecting "Heat Load per Product Footprint" for Communication equipment to be as high as 60,000 W/m^2 in 2005, and Servers to be as high as 15,000. See Figure 4. Heat densities of this magnitude will require substantial changes in the way heat is currently transported out of the space.

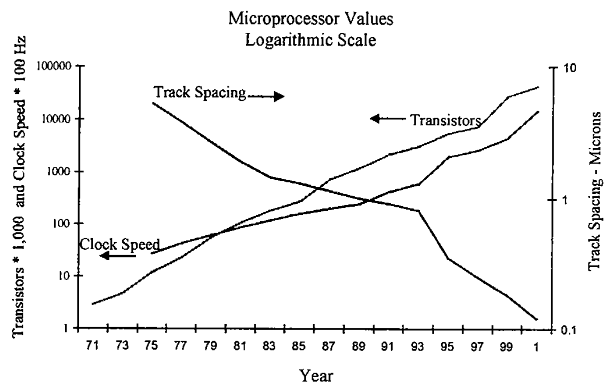

Figure 1

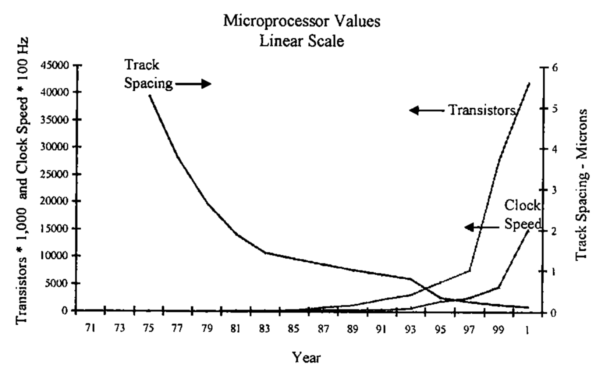

Figure 2

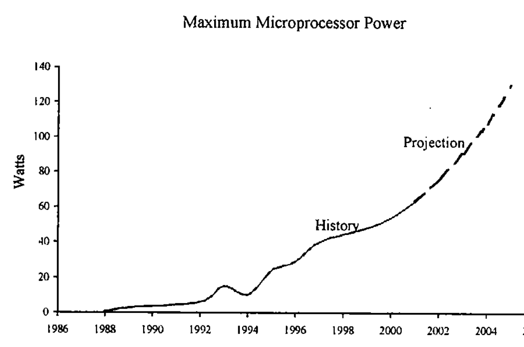

Figure 3

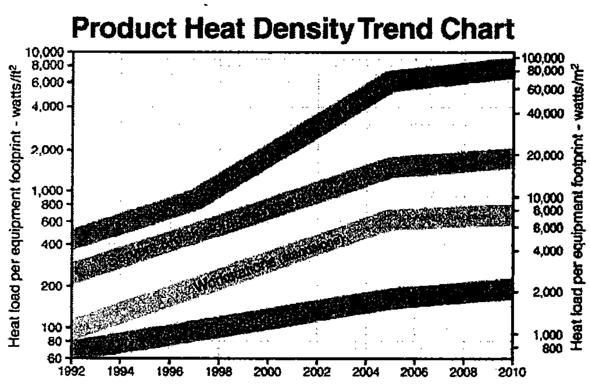

Figure 4

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CURRENT COOLING PRACTICES

The most common current practice for cooling Telecommunications facilities is to use air as the transport medium. This can be done with remote (out of the space) air handling systems utilizing overhead air distribution ducts, or multiple modular systems (in the space) using overhead ducts or mounted on a raised floor supplying air through the under-floor plenum. Common "rules of thumb" for these types of systems are design duct velocity = 5 m/s, airflow = 0.06 m^3/s-kW, vertical space for duct = 0.5 m. Modular cooling unit floor space = 0.02 m^2/kW, with an additional 0.02 for service. A common design heat density for these spaces has been 500 W/m^2. Using these values in a 1,000 m^2 facility with a 4-meter slab to slab height as an example, results in duct width, air changes per minute, and air-conditioner floor space values that are rather easy to accommodate.

Heat Density: 500 W/m^2
Total Heat = 500 kW
Total Airflow = 500*0.06 = 30 m^3/s
Total supply duct cross section = 30/5 = 6.0 m^2
Supply duct width = 6 / 0.5 = 12.0 m
Duct Width as a Percent of Room Width = 12.0/31.6 = 38%
Air changes per minute = (30 * 60)/(1000*4) = 0.45
Assuming the need for 15 % of standby capacity, the modular cooling unit footprint would be:
Total Cooling Unit Footprint (with service space) = 500*1.15*0.04 = 23.0 m^2
Cooling unit space as a percent of room area = 2.3%

FUTURE COOLING REQUIREMENTS

If heat densities continue to increase as forecast, and design practices do not change, Table 1 summarizes the resulting cooling system values at heat densities up to 5,000 W/m^2. Duct dimensions, air changes per minute, and floor space occupied by the cooling system reach generally unacceptable or unachievable levels. Figure 5 demonstrates the floor space occupied by modular cooling equipment at the 5,000-W/m^2 point. Since the Duct requirement would be a 2.0-meter high duct occupying 98% of the ceiling space, it is not possible to demonstrate an acceptable duct layout. Smaller duct sizes could be used, with consequently higher air velocities. However, that would result in more expensive duct construction and greater fan power, increasing operating costs and adding more motor heat for the cooling system to absorb. It appears that current system practices reach their limit at approximately 2,000 W/m^2. It should be noted that these results apply to any system (such as rooftop air conditioners or central station air handlers) that use ducted air as the heat transport medium.

Up until this point, diversity of heat release has not been discussed. Reviewing the curves in Figure 4, it can be seen that there is a great difference in the heat density of different equipment classes. It is not uncommon, currently, for different zones in a communications facility to vary in density by a factor of five or more. As an example, it would not be unusual in a facility with an average density of 500, to have areas with a density of 833, and areas with a density of 167. If projected heat density trends continue, this diversity factor will increase to eight or more within the next five years. This would mean that a facility with an average density of 3,000 might have zone maximums of 5,336 and minimums of 667, further increasing the difficulty of satisfying local heat releases via traditional techniques. Again referring to Figure 4, localized heat release may be as high as 60,000 W/m^2.

Heat Density (w/m2)	500	1000	2000	3000	4000	4500	5000
Total Heat (kW)	500	1000	2000	3000	4000	4500	5000
Room Airflow (m3/s)	30.2	60.4	120.8	181.2	241.6	271.8	302.0
Air Changes per Minute	0.5	0.9	1.8	2.7	3.6	4.1	4.5
Total Duct Area (m2)	6.0	12.1	24.2	36.2	48.3	54.4	60.4
Duct Height (m)	0.5	0.5	1.0	1.5	1.5	1.75	2.0
Total Duct Width (m)	12.1	24.2	24.2	24.2	32.2	31.1	30.2
Duct as a % of Room Width	38%	76%	76%	76%	102%	98%	96%

Table 1

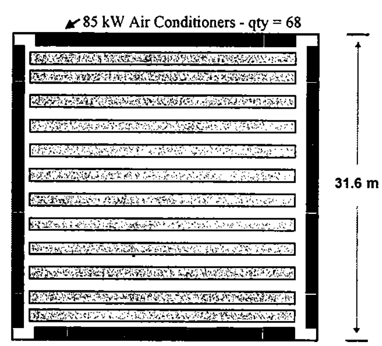


Figure 5

ALTERNATIVE COOLING METHODS

As the average heat release and the diversity of heat release increases a more effective way to absorb this heat may be localized cooling systems. Several system concepts are described below. Also, other materials may be more effective in transporting the heat from the local environment. Fluids and refrigerants are good alternative materials for the absorption and transport of heat. Table 2 compares the volumetric heat transport capacity of air, water, refrigerant R407C, and dielectric fluid HFE 7100.

Material	Air	Water	Refrigerant R407C Liquid	Refrigerant R407C Vapor	Dielectric HFE 7100
Specific Heat J / kg - C	1006	4187	1448	1015	1163
Density kg / m3	1.22	999	1202	32	1543
Enthalpy J / kg			67,498	270,502	
Delta Temp - C	11	6	0	0	6
Volume Heat Capacity – J / m3	13,500	25,096,878	243,955,746	6,552,686	10,767,054
Volume Heat Comparison	1	1,859	18,071	485	797

Table 2

As can be seen, each of these materials has measurably greater heat transport capabilities than air.

Heat densities in communications facilities will very likely increase gradually over time as old equipment is replaced by new, and new product categories are added. Therefore, it can be expected that various products and techniques will be employed at different levels of density and diversity. The following are several possible designs or techniques.

Fan Assisted Enclosure: One method for extending the heat density range of current air cooled environments, particularly those with under-floor air distribution, is shown in Figure 6. This concept draws cold air directly into the enclosure from under-floor (avoiding mixing with warmer air in the aisle) and discharging the heated air out the upper rear (again avoiding mixing with cooler air in the aisle, and returning warmer air to

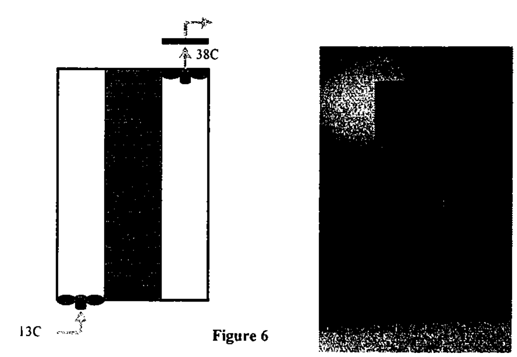


Figure 6

the air conditioner). This particularly helps with equipment that has a high discharge air temperature, which might affect equipment in adjacent aisles.

Higher Capacity, Smaller Footprint Air Conditioners: Another method for extending the range of the current air-cooled environment is to increase the cooling capacity per unit, while decreasing the footprint per kW of cooling. Typical Computer Room Air Conditioners, have capacity ratings of 100 – 120 kW, and footprints of 0.02 kW/m^2. Manufacturing products with capacity ratings of 200 – 240 kW, while reducing the footprint to 0.015 kW/m^2 would reduce the number of units per site by half, and the installation space by 25 %, potentially improving site reliability and saving valuable floor space.

Enclosure Mounted Heat Exchanger: Recent published heat releases for "1U" high servers using dual Pentium III processors, are as high as 292 watts per server. Therefore a "42U" cabinet dedicated to these servers may release as much as 12.2kW of heat. Using a common enclosure footprint of 0.6 m2, this translates to a footprint heat density of 20,333 W/m^2. One concept of cooling such a heat release is shown schematically in Figure 7. It consists of a fin and tube coil commonly used in the HVAC industry, mounted on the rear access door of the enclosure. Heat transport is accomplished via water or a dielectric fluid. The intent of this concept is to absorb the heat released by the communication devices as it leaves the equipment and prior to reentering the space, thereby making the cabinet "load neutral" so far as the general air cooling system is concerned.

Laboratory tests have shown that as much as 18kW of heat can be absorbed by such a technique. Supplemental fans are required to pull the server air through the exchanger. Note the nominal air temperatures from entering to leaving the enclosure. Since the air temperature entering the exchanger is approximately 36C, this increases its heat absorption capacity. As service access is required to the rear of the enclosure, the door must remain hinged. This requires that flexible hoses be used with quick connect fittings for easy opening, installation and removal. Advantages of this solution are it can be added or removed as needed, and only installed on those enclosures that have a high heat release.

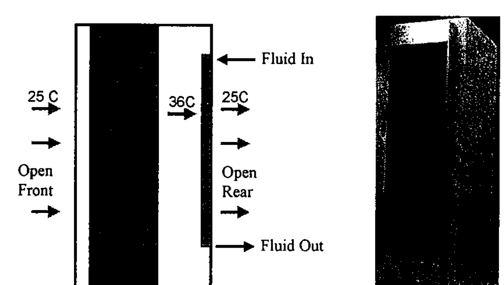


Figure 7

To ensure that no condensation occurs while cooling, the fluid temperature must always be above the dew point of the air. For this reason, and also to ensure proper flow, system pressures, fluid purity, and isolation from the main cooling loop a Coolant Distribution Unit (CDU) should be used, as shown in Figure 8. The CDU contains a fluid to fluid heat exchanger, pumps, control valves, and control. As the CDU provides fluid to multiple exchangers, it would likely have a capacity of between 100 and 200 kW. The power consumption of the fans and CDU pump allocated to the 8 kW model of this enclosure-mounted heat exchanger is 288 watts, or 36 watts/kW. The fan of a traditional cooling system consumes approximately 80 w/kW. This represents a 55 % reduction in power.

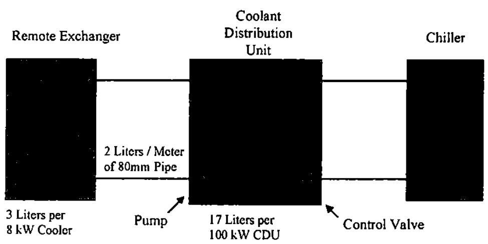


Figure 8

Ceiling Mounted Heat Exchanger: As indicated previously, as heat density increases, it becomes increasing difficult to bring air to the area requiring cooling, using traditional ducting methods. However, a modular ceiling system consisting of a fin & tube coil plus fans, and using a fluid or refrigerant as the heat transport medium, can provide local air movement, and high heat density cooling. One concept for such a device is shown in Figure 9. This concept includes movable fan trays, to optimize the air supply and return path based on the communication equipment beneath it. Laboratory tests have shown that such a system can support heat densities of up to 5,000 W/m^2. This system requires a CDU, for the same reasons as mentioned previously, if a fluid is used as the transport medium. If a refrigerant is used as the transport medium, a remote condensing unit is required, and if multiple ceiling units are connected to the same condensing unit, variable capacity is required. Again, the refrigerant temperature must be maintained above the air dew point. The power consumption of the fans and CDU pump allocated to this 20 kW model of the ceiling mounted heat exchanger is 1,020 watts, or 51 watts/kW, representing a 36% reduction in power.

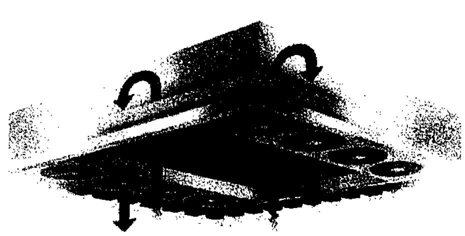


Figure 9

Horizontal Air Displacement Units: Another method of providing local air cooling without extensive ducting, and also cooling in a manner that enhances the natural tendency of warmer air to rise is shown in Figure 10. This concept delivers low velocity cool air at lower levels of the equipment aisle, naturally displacing warmer air as it travels. Heated air exiting the equipment is drawn back to the top of the cooling unit. The sound level of this system tends to be low. This system has been applied with heat densities of up to 1,000 W/m^2.

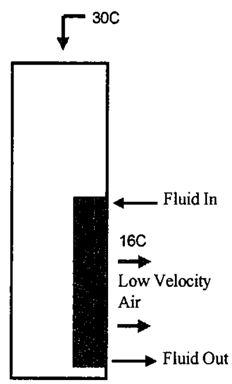



Figure 10

Fluid Assisted Heat Sinks: All of the above systems remove heat from the air that has been heated by the communication equipment. As the packaging density of equipment increases, and the power and power density of the microprocessor increases, it may become necessary for the thermal path to go directly from the processor to a fluid or refrigerant circuit, without air as an intermediary. One concept for such a path is shown in Figure 11. Since conduction is superior to convection in transferring heat, the potential advantages of such a system are: greater heat removal per unit of volume, lower chip operating temperatures, and the ability to support higher equipment footprint heat densities. This solution may be incorporated in a proprietary way by individual manufacturers, or via industry standard construction and dimensions for all manufacturers. The need for this solution may occur as microprocessors approach 150 watts in power consumption and/or 150 W/cm^2 in surface density. Again, a

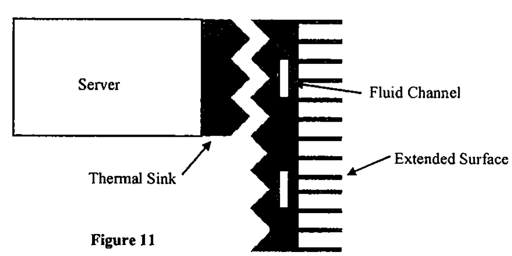


Figure 11

CDU would be necessary for the reasons expressed earlier.

Other Methods: It is not expected or intended that the devices described above are a complete list of all current and future methods of cooling high density heat releases. Spray cooling is an example of a method in use today in Cray computers, thermal conduction modules were employed in previous generations of IBM mainframes, and research is being conducted at the university level on two phase boiling techniques. Each manufacturer may find a unique solution to these challenges.

The Future Cooling System: It is probable that the future complete cooling system for a communications facility will be a combination of many, if not all of the above methods. The thermal demands imposed by increasing average, and increasingly diverse heat densities, will require solutions tailored to these densities. However, it is not expected that the current method of heat absorption via air-cooling will vanish as a primary means of maintaining a suitable environment. Certainly many pieces of communications equipment will continue to be thermally satisfied using current methods of air cooling. General temperature and humidity control are examples of tasks that they will continue to perform.

Thermal Conditions During a Loss of Cooling: While it may be possible to satisfy increasing heat densities with the methods or devices described above, another significant operating condition that must be considered is the temperature and temperature rise of the space in the event of a catastrophic loss of cooling. Examples of a catastrophic cooling loss are, a power failure during which the communications equipment operates on battery power but the cooling system does not, a failure of the cooling system, an inadvertent power disconnect of the cooling system, etc. Figure 12 is actual measured data in a 42 m^2 test laboratory, with the walls and ceiling surrounded by a 20 C space temperature. Note that when heat densities approach 4,844 W/m^2, the air temperature entering the enclosure remains below 51C for only five minutes. Therefore, either the cooling system has to be restarted, and achieve full capacity in less than five minutes, or some means of thermal "ride through" must be provided. Also note that at 4,844 W/m^2 the temperature rise is approximately 5.2C / minute, which far exceeds current test standards for communication equipment, (typically 30C / hour).

Therefore a combination of both thermal storage for "ride through", (to decrease the temperature rise) and rapid return to full cooling (to reestablish conditions) is an optimal solution, and should be considered in any facility with high heat density. Thermal ride through can be achieved through the use of fluid storage tanks, or other techniques. An analysis of which cooling components, such as pumps, controls, and fans, must be powered by battery power is required for each system configuration. Depending on the degree of ride through that is desired the thermal storage volume may be quite large. Switchover to redundant equipment must be automatic and swift, and the capacity and speed with which standby generation is activated must be assured. Computational Fluid Dynamics (CFD) or other modeling techniques can be employed to create these plots for larger facilities, where temperature gradients may be much greater.

Energy Use, and Potential for Savings: Energy saving techniques applied to the cooling system can achieve substantial reductions in energy use and cost. Assuming a cooling system with the fans consuming 13% of system power, pumps 7%, condenser 17%, and compressor 63%, the following savings are possible. By, applying a variable frequency drive (VFD) to the pumping system it may be possible to achieve a 30% savings in annual pump power consumption, a 2% overall savings. If ceiling or enclosure mounted heat exchangers were employed to absorb half of the heat release, an additional 18 to 28% reduction in fan power could be achieved, a 2 – 4 % overall savings. And if a "free cooling" circuit were used to reduce compressor power in a climate such as Frankfurt's, (see Figure 13), a 40 - 50 % savings in annual compressor power could occur, a 25 - 31 % overall savings. Therefore, attention to system design and product selection could result in 29 – 37 % overall savings in annual cooling power. Assuming a cooling system COP of 2.5, a power cost of 0.10 Br Lbs / kWh, a site size of 1,000 m^2, and a heat density of 2,000 W/m2, this would mean an annual savings of between 202,000 and 260,000 Br Lbs / year. To convert these values to Euros, multiply by approximately 1.5.

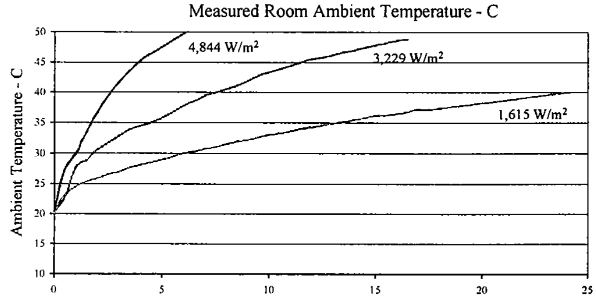


Figure 12

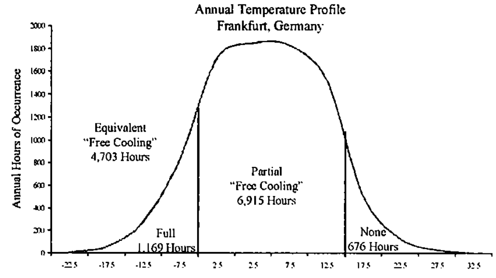


Figure 13

Summary: Current and projected trends of heat density in telecommunications spaces will result in new cooling solutions to high density heat releases. Product heat density may reach 60,000 W/m^2 by the year 2005. These cooling products and systems will likely be modular and local, and may utilize a different heat transport medium than air. This medium may be water, a refrigerant, a dielectric fluid, or other material that has a higher volume heat capacity. Multiple different systems may be employed in the same facility, in order to satisfy diverse densities. Thermal conditions during a loss of cooling will require methods of thermal storage and rapid re-establishment of cooling, in order to avoid equipment shutdown. Energy saving methods can result in substantial operating cost reductions.

Recommended Additional Reading:

[1] Uptime Institute ,2000, Heat Density Trends in Data Processing, Computer Systems, and Telecommunications Equipment.

[2] International Roadmap Committee & the International Technology Working Groups, 2000, International Technology Roadmap for Semiconductors – 2000 Update, p. 23.

[3] D. Abi-Zadeh, P. Samain, 2001, A Transient Analysis of Environmental Conditions for a Mission Critical Facility after a Failure of Power.

[4] K. Azar, 2000, The History of Power Dissipation, Electronics Cooling, January 2000, Vol. 6, No. 1.

[5] R. Schmidt, 2000, Low temperature electronic cooling, Electronics Cooling, September 2000, Vol. 6, No. 3.

[6} C. Patel, 2000, Enabling Liquid Loop Cooling: Justification and the Key Technology & Cost Barriers, High Density Packaging Conference, 2000.

[7] I. Mudawar, 2000, Assessment of High-Heat-Flux Thermal Management Schemes, Itherm 2000 Conference.

Low Profile Heat Sink Cooling Technologies for Next Generation CPU Thermal Designs

Marlin Vogel and Guoping Xu
Sun Microsystems

Introduction
The electronic industry requires increased forced-air cooling limits to cool high-end server CPUs adequately. Improving air-cooled heat sink thermal performance is one of the critical areas for increasing the overall air-cooling limit. One of the challenging aspects for improving heat sink performance is the effective utilization of relatively large air-cooled fin surface areas when heat is being transferred from a relatively small heat source (CPU) with high heat flux. To meet the next generation CPU thermal requirements with a low profile heat sink, we will describe three heat sink technologies (i.e., embedded heat pipe; vapor chamber; and oscillating/pulsating heat pipe) and their associated prototypes. Each of these technologies uses internal liquid-to-vapor phase change to spread the local CPU heat efficiently to the air-cooled fin structure.

The increased electrical performance required by the computer industry has created thermal design challenges due to both increased power dissipation from the CPU and spatial envelope limitations. Local hot spot heat fluxes within the CPU are exceeding 100 W/cm^2, while the maximum junction temperature requirement is 105°C or less. The heat sink requirements for this study are,

- Sink-to-Air Thermal Resistance: 0.18°C/W
- Heat Source Size: 16 x 16 mm

- Flow Direction: Front-to-back (perpendicular to gravity)
- Spatial Envelope (H x W x Flow Length): 50 x 115 x 150 mm
- Mass: 680 grams
- Altitude: Sea level
- Orientation: Side heating

An all-metal heat sink was optimized by means of a commercial CFD software tool to yield the minimum sink-to-air thermal resistance, while not exceeding the pressure loss and mass requirements. This heat sink has a 4 mm thick copper base and 70 aluminum fins. Each fin is 0.28 mm thick. This article shows that for the above requirements, the thermal resistance of the all-metal heat sink design is 44% greater than the measured thermal resistance of the prototype heat sinks tested. Additionally, the thermal performance achieved by these prototype heat sinks exceeded the thermal performance levels for all the submitted competitive designs and prototypes, which utilized similar fluid phase change technologies.

Experimental Verification Testing and Setup

The above requirements for each prototype were verified through experimental testing initially conducted at the companies providing the prototypes. Final verification testing was conducted at the Sun Microsystems R&D facility in San Diego, CA. Test results attained by the prototype suppliers agreed relatively well with the final verification test results. The maximum difference in measured thermal resistance attained by all of the test facilities was less than 10% for any given flow rate.

Heat sink thermal performance and pressure drop experiments were conducted at Sun Microsystems in a wind-tunnel system (Figure 1). The test set-up consisted of a fully ducted airflow channel, a heater block under the heat sink, an airflow test chamber to measure airflow rate, and an air-mover. The duct cross sectional area was the same as that of the heat sink to create a fully ducted airflow. Two pressure taps and two thermocouples located upstream and downstream of the heat sink were used to measure the static pressure and airflow inlet and outlet temperatures respectively. The airflow chamber. The airflow rate was calculated from measured pressure difference across the nozzle.

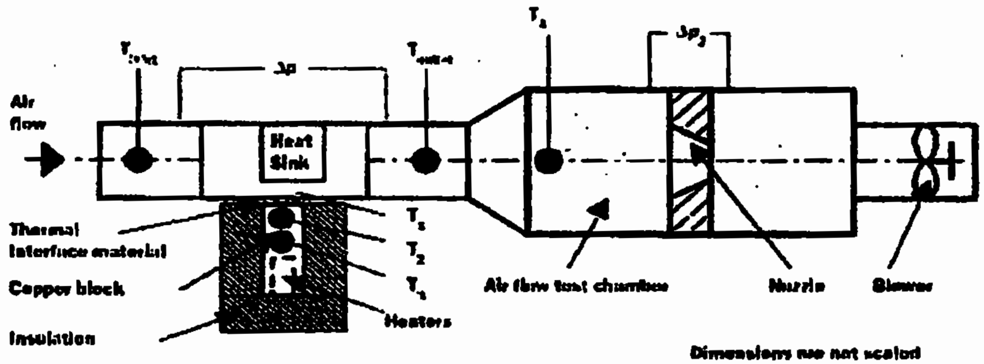


Figure 1. Verification test setup and wind tunnel.

A fiberglass-insulated copper block with embedded AC cartridge heaters provided power input for the heat sink. Two thermocouples T_2 and T_3, located below the heater block/heat sink interface surface, were used to measure the actual power input to the heat sink by assuming one-dimensional conduction along the heater block. A commercially available, high-performance thermal grease (with an effective thermal conductivity of 3.5 W/mK) served as the thermal interface material between the heat sink and the heater surface. The heat sink base temperature was measured by embedding thermocouple T_1 0.5 mm above the bottom surface of the heat sink base. All thermocouples were calibrated in a temperature controlled water bath with resolution of 0.01°C. In the verification test work the power level was fixed at 150 W and data was recorded after temperatures in the heat sink base and heater block were allowed to reach steady-state for a given air flow rate.

Embedded Heat Pipe Heat Sink

The embedded heat pipe [1,2] heat sink prototype appears in Figure 2. The internal and external heat pipe geometry to minimize the intrinsic temperature drop in the heat pipes. This approach distributes the heat over the base of the heat sink. The joining processes minimize the interfacial temperature drops to get the heat into and out of the heat pipes. Other embedded designs have a lower performance because they do not achieve this balance successfully.

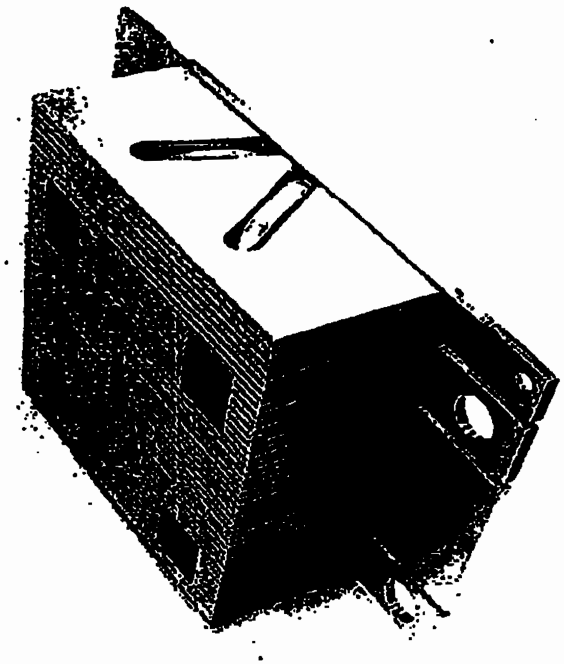


Figure 2. Embedded heat pipe heat sink prototype.

A heat pipe may be defined as a two-phase (liquid-vapor) device that

which exists in both liquid and vapor states within the container. The working fluid absorbs heat and evaporates wherever the container is hot, carries the heat by physically flowing as a gas through the vapor core to the cooler parts of the container, liberates heat and condenses into a liquid there, and returns through the wick to the hot regions, thus completing a thermodynamic cycle. The temperature differences between the hot and the cold regions within the vapor core of the heat pipe depend only on the pressure drop that the vapor experiences as it flows through the vapor core. This temperature drop is typically quite small, giving the heat pipe an effective thermal conductance significantly greater than that of copper. The wicking structure enhances evaporative heat transfer at the evaporator and provides the motive force to return the liquid phase back from the condenser to the evaporator by capillary action when gravity cannot be utilized to return the liquid.

Vapor Chamber Heat Sink

The prototype vapor chamber [3] heat sink, appearing in Figure 3, is a 3-dimensional heat pipe located in the heat sink base. It represents a relatively new technology that became commercially available during the mid-1990s, versus the traditional unidirectional heat pipe technology available for over 25 years. The prototype supplier carried out an aggressive development effort, which allowed the wick thermal resistance to decrease by 50%, thereby offering an advantage over other heat sinks incorporating vapor chamber technology.

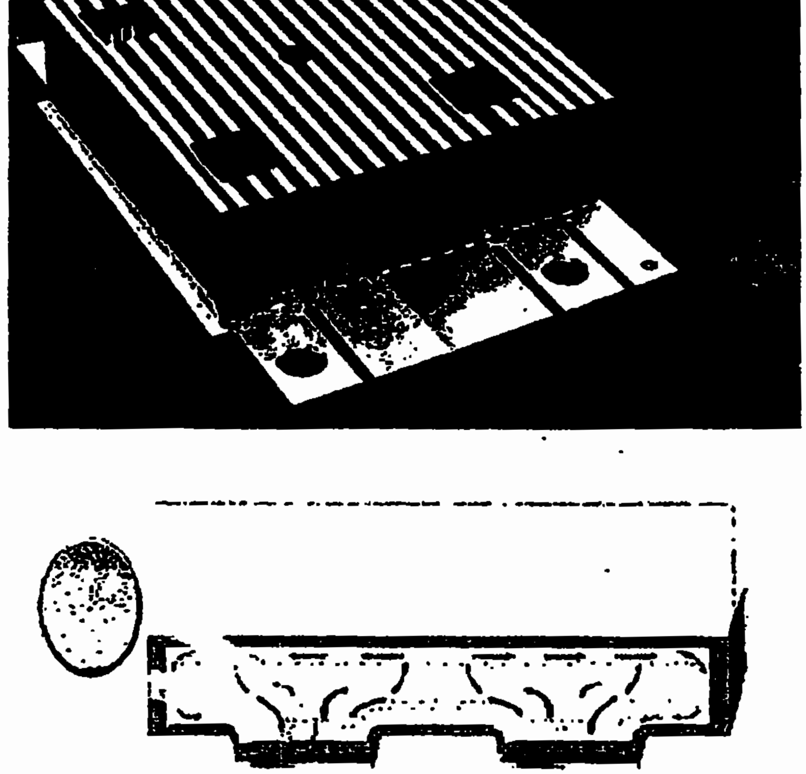
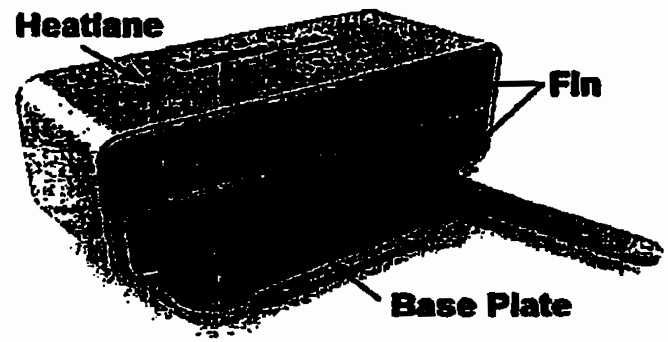


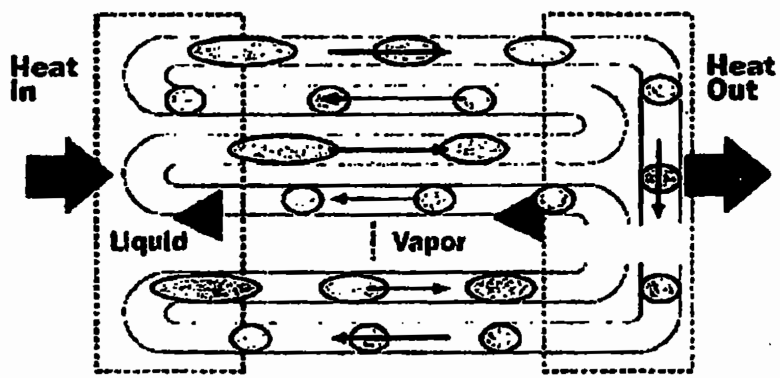
Figure 3. Vapor chamber heat sink prototype and illustrated liquid-vapor heat/mass transfer.

The vapor chamber minimizes the spreading resistances in the heat sink base by allowing the heated vapor to make full contact between the heat input region and the base of the heat sink fin structure. Figure 3 also illustrates the liquid-to-vapor heat/mass transfer operation when two heat sources are

Oscillating/Pulsating Heat Pipe Heat Sink

The oscillating heat pipe [4] heat sink prototype appears in Figure 4. In this case, the fin structure flow length is 47.5 mm. Figure 4 also illustrates heat transfer and fluid flow direction. Unlike the previously described prototypes, the oscillating heat pipe heat sink does not contain a wicking structure.





http://www.electronics-cooling.com/html/2005_feb_a2.html several times. Blue circles are vapor bubbles. Blue oblongs are vapor slugs. The remaining white background region inside the serpentine-shaped ducting is liquid. Black arrowheads indicate which regions of the flow through the ducting are liquid and which are vapor.

Similar to vapor chamber technology, the oscillating/pulsating heat pipe is a relatively new technology, invented in 1994 by Mr. Akachi [5]. A flat extruded, aluminum plate incorporates both the working fluid and an undulating turned capillary tube. For successful operation, liquid slugs and vapor bubbles must coexist inside the capillary tube along its length (Figure 4). The extruded aluminum plate is formed into a scroll-like shape (Figure 4) to spread heat not only horizontally, but also vertically, thereby providing 3-dimensional heat spreading.

Dr. Polasek [6] described the physics for the oscillating fluid flow and heat transfer as follows: "When one end of the undulating capillary tube is subjected to high temperature, the working fluid inside evaporates and increases the vapor pressure, which causes the bubbles in the evaporator zone to grow. This pushes the liquid column toward the low temperature end (condenser). The condensation at the low temperature end will further increase the pressure difference between the two ends. Because of the interconnection of the tube, the motion of liquid slugs and vapor bubbles at one section of the tube toward the condenser also leads to the motion of slugs and bubbles in the next section toward the high temperature end (evaporator). This works as the restoring force". As a result the force of gravity has a minimal effect on fluid flow direction.

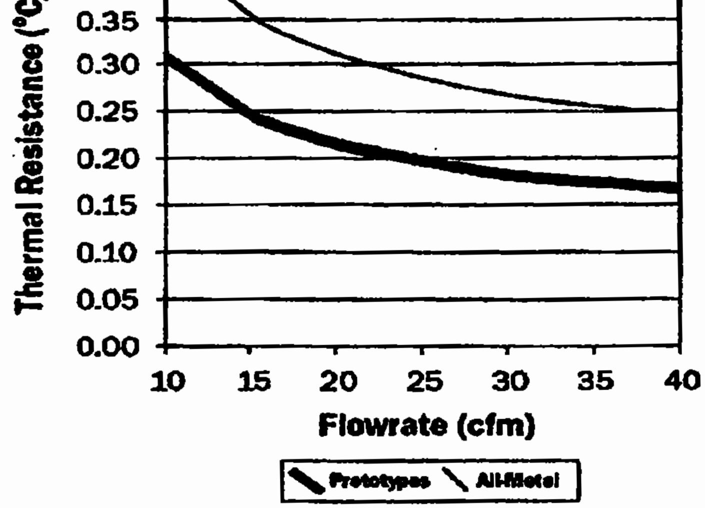


Figure 5. Sink-to-air thermal resistances for the prototypes and the optimized all-metallic heat sink.

Results

The measured sink-to-air thermal resistance for the tested prototypes appear in Figure 5. At the design flow rate of 35 cfm (0.0165m^3/s), the maximum resistance of these prototypes met the requirement of 0.18°C/W. Figure 5 also shows that, for any given flow rate, the range in thermal resistance for these prototypes was typically less then 0.013°C/W. In addition, the results show that at the 35 cfm (0.0165m^3/s) airflow rate, the optimized all-metal heat sink yielded a thermal resistance of 0.26°C/W.

Summary

The measured test results verified that all of the prototypes, which utilized liquid-to-vapor phase change technologies, met the heat sink design requirements, including the critical 0.18°C/W sink-to-air thermal resistance.

technology needed to meet the requirements should be determined on an application-by-application basis, and be based on the technological strengths developed within individual companies. This can only be done when thermal solution providers and customers work together to understand fully the problem at hand, allowing the individual companies to optimize and apply the specific technologies that they have developed towards meeting a common set of design requirements.

Acknowledgments
The authors acknowledge the contributions towards this article made by the prototype development engineers and their respective companies that provided the prototypes: Brad Whitney (Aavid Thermalloy - embedded heat pipe) Matt Connors (Modine Thermacore - vapor chamber), and Takahiro Katoh (TS Heatronics - oscillating heat pipe).

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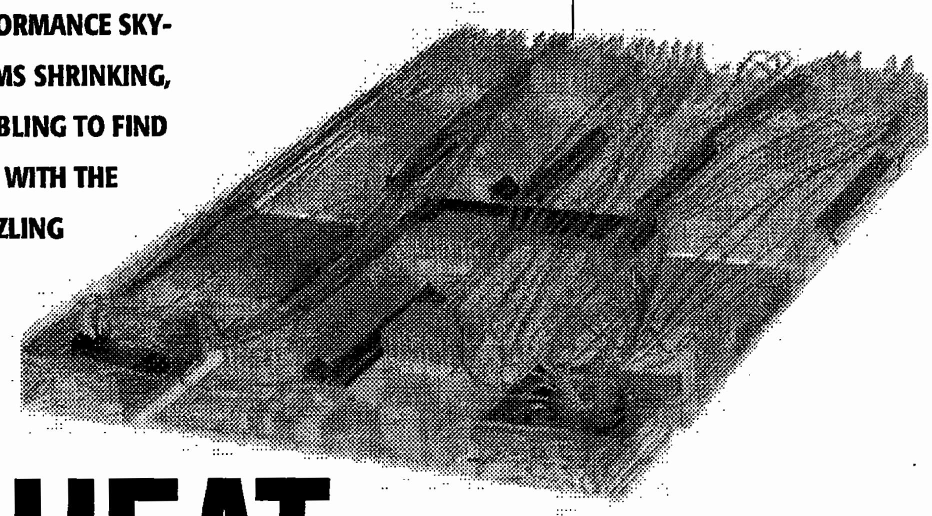
 By Warren Webb, Technical Editor



The thermal model of a 1U server using the Icepak analysis package shows colors representing surface temperatures and flow ribbons indicating airflow.

WITH PROCESSOR PERFORMANCE SKYROCKETING AND SYSTEMS SHRINKING, DESIGNERS ARE SCRAMBLING TO FIND CREATIVE WAYS TO LIVE WITH THE HEAT THAT TODAY'S SIZZLING CIRCUITS GENERATE.

Take the HEAT

Cool that hot embedded design

At a glance 50
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THE DESIGN GOALS of many new projects include miniaturization, increased performance, longer battery life, and silent operation. All of these goals run counter to effective thermal management, and designers are facing the prospect of sacrificing performance to handle the excessive heat that modern ICs generate. Cooling problems plague all electronic systems, from the simplest portable devices to complex board-level systems. Although mechanical-packaging engineers typically handle thermal analysis on larger projects, most design teams rely on rules of thumb, experience with similar projects, guidelines from board or chassis manufacturers, simple thermal analysis, and the old standby—trial and error.

Much of this thermal consternation is due to the effects of Moore's Law: the exponential growth in the number of transistors on a chip. Over the last 30 years, ICs have progressed from 10,000-nm resolutions with 3000 transistors to less than 100-nm resolutions with hundreds of millions of transistors. With these added transistors, designers have increased IC performance by adding parallel execution paths and therefore increased power dissipation. The power consumption in CMOS ICs is proportional to fCV^2, where C is the sum of all transistor gate, drain, and interconnect capacitances; f is the frequency of operation; and V is the core voltage. It is easy to see that, as the number of transistors and the frequency increase, the power climbs. Although reducing the voltage saves power, leakage current through ultrathin gate structures also contributes to overall power consumption. This leakage current is nonlinear and becomes significant at less than 1V. **Figure 1** shows the relentless growth of power density in each successive microprocessor generation.

Along with shock, vibration, and humidity, temperature is the major source of environmental stress in electronic systems. Temperature can reduce performance by lowering output-voltage swings, reducing switching speeds, lowering noise margins, and generally reducing overall signal quality. In addition to the performance hit, temperature stresses also reduce system reliability. Mechanically, high temperature can lead to wire-bond failures, die fractures, and increased corrosion. Excessive heat may also overstress electrical components, causing gate-oxide breakdown, electromigration, ion diffusion, and, ultimately, failure. The key to improving reliability and maintaining performance is to ensure that each component in the system is operating within its temperature parameters.

AT A GLANCE

- ▷ Less-than-100-nm technology delivers extremely high power density and new thermal-management problems for embedded-system designers.
- ▷ As temperatures rise, embedded systems are rapidly approaching the limits of conventional forced-air-cooling techniques.
- ▷ Automatic analysis software provides thermal maps from board-layout data to identify hot spots and reliability problems.
- ▷ Liquid-heat-transfer techniques are on drawing boards to cool the next generation of electronic systems.

HOT AIR RISES

Designers have many options for dissipating the heat generated by electronic circuits. Convection, a passive form of electronic cooling, transfers heat by airflow due to temperature gradients; in other words, hot air rises. This type of cooling is most common for small, portable devices, such as cell phones and PDAs, and you can augment it by increasing the surface area of dissipating devices with a heat sink. For simple systems, designers can estimate the thermal performance directly from component-data-sheet parameters. The most common measure of package thermal performance is θ_{JA}, the thermal resistance measured or modeled from junction to ambient. You use θ_{JA} to measure the temperature difference between the component and the ambient atmosphere when the component has consumed 1W of power.

If you know the thermal resistance and the operating power, P, of the component, you can approximate the junction temperature, T_J, from $T_J = T_A + \theta_{JA} {*} P$.

Data sheets include another parameter, θ_{JC}, the thermal resistance from junction to case, which may be more useful when the component is cooled by forced air, heat sink, or liquid cooling. In this case, $T_J = T_C + \theta_{JC} {*} P$, where T_C is the temperature of the case or packaging. The efficiency of convection cooling improves when you supplement the air movement with a fan. Although forced-air cooling has for years served the electronics industry, it is now stretched to its limit, with each new high-performance design calling for increased heat-sink area and higher airflow rates. With the notable exception of PC/104, most bus-based systems include forced-air cooling, and their specifications include guidelines for maximum dissipation per plug-in module.

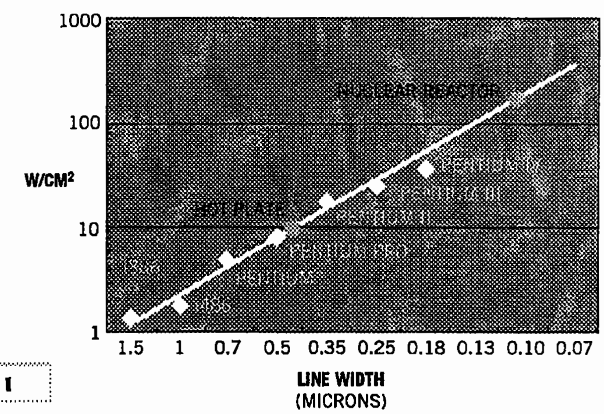
As the number of components on a board or system grows, designers need an automatic analysis technique to detect thermal stress. The Betasoft-Board package from Dynamic Soft Analysis is an example of software specially designed for board-level thermal analysis. It outputs board temperature and gradient maps, component and junction temperatures, and the amount those temperatures exceed their respective limits (**Figure 2**). The software models boards with as many as 1500 components per side and performs 3-D modeling of the complex flow and thermal fields based on heat conduction, convection, and radiation.

Betasoft-Board interfaces with several board-layout programs for automatic transfer of component-placement and parameter information. Advanced features include multilayer boards, irregular shapes, bolt-on heat sinks, conduction cooling through wedge locks, or sealed compartments. The flow field can be natural or forced-convection, and heat exchangers can cool closed systems. The software models the effects of gravity, air pressures, and flow directions. You can also attach heat sinks, heat pipes, chip fans, and conduction pads to components. You can transfer the junction temperatures calculated by the Betasoft-Board analysis into reliability software to improve the reliability predictions.

THERMAL-ANALYSIS SOFTWARE

Fluent Inc offers electronics-cooling-simulation software based on computational-fluid dynamics for mechanical-packaging engineers. Its Icepak software allows an engineer to build a computer model of a product or system and then virtually prototype and test it under real-world conditions. Icepak accepts input from standard CAD-analysis tool sets and includes a downloadable library of fans, heat sinks, and IC packages for manual model building. The latest version includes a new user environment that features a model manager, advanced object wizards, alignment tools, and four-

Figure 1



Extremely narrow lines and millions of available transistors have led to microprocessor power density approaching that of a nuclear reactor (courtesy Intel).

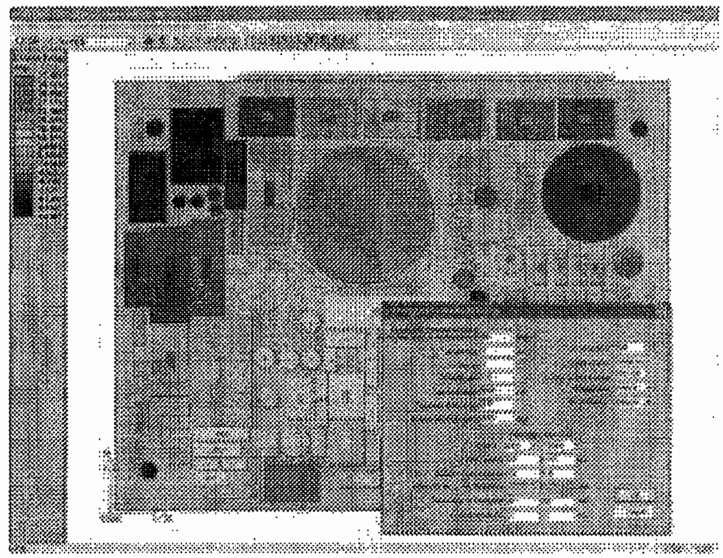


Figure 2 **The Betasoft-Board package from Dynamic Soft Analysis provides board gradient maps, junction temperatures, and thermal-limit checks.**

window simultaneous views. The thermal model in the photo on pg 49 is of a 1U server cooled using heat sinks and two blowers at the back of the chassis. The colors correspond to surface temperature, and the flow ribbons indicate the pattern air takes coming through the inlet vents, over and around the components, angled DIMMs, heat sinks, power supplies, and baffles before entering the blowers and exiting the system. Icepak is available for Unix and Windows platforms as an annually renewable license. A typical entry-level price is $20,000, which includes the software, training, and unlimited technical support.

Although you may have a well-designed system that reacts thermally as predicted by manual or automatic analysis, several situations may add undue stress and cause unexplained failures. For example, consider a board-based system that was designed and analyzed with a full complement of plug-in boards but now has empty slots due to changed requirements or degraded operation.

The resulting air leakage through the empty slots upsets the laminar flow of air inside the card cage and reduces the airflow over the remaining boards. Card-cage manufacturers generally offer baffle boards that you can install in an unused slot to both close the front panel and block the airflow to the unused slot. You may encounter other thermal problems in board-based systems, such as loading one area in the chassis with all the high-power boards instead of evenly distributing the heat producers. Board-layout problems may also create thermal shad-

FOR MORE INFORMATION...

For more information on products and technologies such as those discussed in this article, contact any of the following manufacturers or institutions directly, and please let them know you read about them in *EDN*.

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www.cooligy.com

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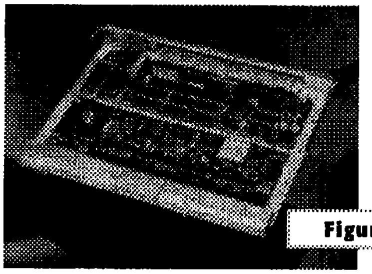

Figure 3

Vista Controls' PPC G4C single-board computer incorporates conduction cooling for an expanded operating-temperature range.

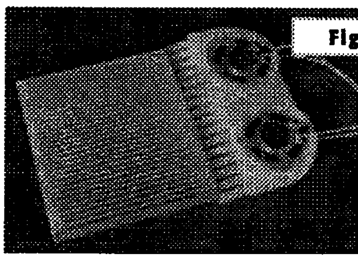

Figure 4

SynJets cooling modules produce two to three times as much cooling as conventional fans with two-thirds less energy input.

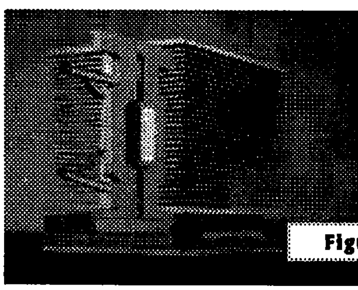

Figure 5

Cooligy's heat collector uses liquid and an electrokinetic pump to transfer excessive heat via tiny microchannels to any location.

ows where a tall component sits ahead of a short component in the air stream. A turbulent eddy often forms over the short component, trapping the heated air and causing severe overheating.

Thermal-induced failures can also result from an unmanaged power-down sequence. Most systems are designed with a single power switch that simultaneously removes voltage from the power supply, the boards, and the fans. If the system happens to be running near its upper thermal limit, and airflow is halted, the internal temperature will continue to rise after shutdown even though the components are no longer powered. Because of the component density, normal convection cannot remove the remaining heat, and elevated temperatures may last 10 to 20 minutes, possibly damaging the IC-die level, packaging, and solder joints. Solutions are to simply to add a time delay to allow fans to operate for a period after power-down or to add a separate switch to control air circulation. You can easily evaluate the thermal performance of a forced-air system by measuring the temperature in the intake and exhaust air stream at each board position. You should measure temperatures at several points along the board to check for hot spots or uneven airflow. A difference of 10°C or greater between the intake and exhaust temperatures may indicate the need for increased airflow.

HEAT-PIPE COOLING

Even if a design calls for a high-power processor or system on chip, advanced techniques are available to cool a hot spot. For example, Thermacore offers a line of passive heat-dissipation devices based on heat-pipe technology. A heat pipe consists of a vacuum-tight envelope, a wick structure, and a working fluid. The heat pipe is evacuated and then backfilled with a small quantity of working fluid, just enough to saturate the wick. The atmosphere inside the heat pipe is set near the equilibrium between liquid and vapor. Heat entering at the evaporator upsets this equilibrium, generating vapor at a slightly higher pressure. This higher pressure vapor travels to the condenser end, where the slightly lower temperatures cause the vapor to condense, giving up its latent heat of vaporization. The condensed fluid is then pumped back to the evaporator by the capillary forces developed in the wick structure. This continuous cycle transfers large quantities of heat with low thermal gradients.

To battle the heat that high-performance systems generate, many manufacturers of extended-temperature boards have turned to conduction cooling to help tame thermal problems. Thermal planes within a board and conductive bars across and along the edge of the board shunt heat away from high-power components to the chassis and eventually to the outside air. Vista Controls' PPC G4C is a recent example of a single-board computer incorporating conduction cooling (**Figure 3**). The 6U CompactPCI board incorporates a PowerPC 7457 processor along with dual Gigabit Ethernet ports. Additionally, the board offers dual 64-bit, 66-MHz PMC sites, 2 Mbytes of external L3 cache, 512 Mbytes to 1 Gbyte of onboard SDRAM, 64 kbytes of nonvolatile RAM, and 128 Mbytes of program flash. The PPC G4C is available in extended-temperature, rugged air-cooled, and conduction-cooled versions. The board is priced at $7100.

Several new technologies on the horizon offer alternative techniques for cooling electronic systems. For example, SynJets, or synthetic jet-ejector arrays, developed by the Georgia Institute of Technology's School of Mechanical Engineering consist of a diaphragm mounted within a module that has one or more orifices. Electromagnetic or piezoelectric drivers cause the diaphragm to vibrate 100 to 200 times per second. The rapid cycling of air into and out of the module creates pulsating jets that you can direct to the precise locations that require cooling. SynJets produce two to three times as much cooling as conventional fans with two-thirds less energy input. With no friction parts to wear out, SynJet cooling modules are much smaller than fans, and you can mount them directly within the cooling fins of heat sinks (**Figure 4**). Although the jets move much less air than similarly sized fans, the turbulent and pulsating airflow breaks up thermal boundary layers.

VIDA (vibration-induced droplet atomization), another advanced cooling technology from Georgia Tech, uses atomized liquid coolants to carry heat away from electronic components. VIDA uses high-frequency vibration produced by piezoelectric actuators to create sprays of tiny cooling liquid droplets inside a closed cell attached to an electronic component. The droplets form a thin film on the heated surface, allowing thermal energy to be removed by evaporation. The heated vapor then condenses, either on the exterior walls of the cooling cell or on tubes carrying liquid coolant through the cell. The liquid is then pumped back to the vibrating diaphragm for reuse. To date, researchers have been able to cool about 420W/cm² and ultimately expect to increase that figure to 1000W/cm².

Georgia Tech has licensed both of these cooling technologies to industry, and they should soon be available.

ION-DRAG PUMP

Another advanced cooling technology, developed at Stanford University's Mechanical Engineering Department, uses common materials to produce a noiseless, closed-loop, active-cooling system for high-power ICs. The university has now licensed the technology to Cooligy. The active-microchannel system employs a fluid pumped in a sealed cooling loop. A heat collector is attached to the chip to absorb heat that hot spots generate. The heat travels a small distance into fluid flowing through the microchannels in the collector, 20 to 100 microns wide each, which transport the heat away from the chip to a radiator, where the heat is exhausted to the outside air (**Figure 5**). The fluid then travels through Cooligy's solid-state electrokinetic pump to complete the cooling loop. Using fluid to transfer heat means that the cooling system can pump large amounts of heat away from the chip to any location the system designer chooses. Cooligy's electrokinetic pump is based on an interaction between a fluid and glass. The walls of a fluid-filled glass tube carry a negative charge that is balanced by positive ions in the fluid accumulating near the walls of the tube. When you apply an electric field along the length of the tube, the excess positive ions near the tube's wall move parallel to the wall and push the fluid through the tube. The core of Cooligy's pump is a glass disk with millions of paths through which fluid is pumped using this electrokinetic effect.

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Because your customers will always want the latest technology to squeeze the best performance from a system, you can expect that each new project will have thermal issues. To avert these temperature problems, IC-design and heat-analysis software may become standard tools for future development projects. As technology continues to keep pace with Moore's Law, we must learn to live with its inevitable side effect: heat.

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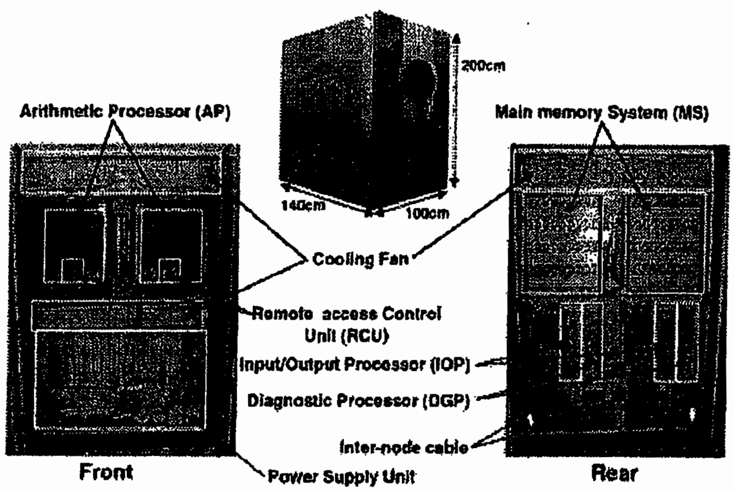
Processor Node (PN) Cabinet



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PUMPED LIQUID MULTIPHASE COOLING

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ABSTRACT

The performance limits of conventional cooling technologies are being reached in both military and commercial electronics and electro-optical systems. Pumped Liquid Multiphase Cooling (PLMC) provides significantly enhanced thermal management capabilities for these systems using mostly-conventional components and working fluids. PLMC is highly scalable and reliable. Energy efficiency is very high, surpassing conventional techniques by up to two orders of magnitude. This paper describes the basic PLMC technology and presents experimental results from several prototype embodiments of the technology

INTRODUCTION

The thermal control of electronic and electro-optical devices and systems is an integral part of their design and performance. Historically, commercial electronic systems such as computers and telecommunications switches have been cooled by natural and forced convection using ambient air. Wherever possible, military and defense-related systems also use air cooling, but in many cases single-phase liquid cooling (usually using chemically-treated water as the working fluid) has been required. Examples of such systems include high-energy laser arrays and high-power radars.

For microprocessor-based commercial systems, cooling is recognized as a key technology required for continued progress. In general, there is agreement as to the general trends. Shown in Figure 1 are the predictions in 1993, 1999, and 2002 for the heat generated by high-end devices – microprocessors and microcontrollers [1,2,3].

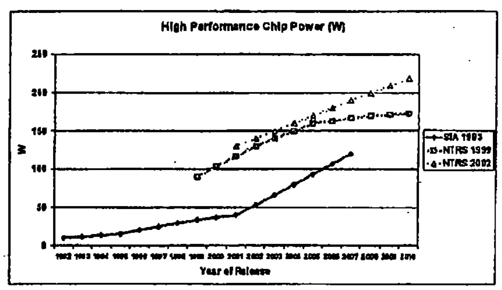

Figure 1: Industry power dissipation forecast

It seems clear that chip powers will exceed 150 W by the middle of the decade; this is generally beyond the capabilities of even "heroic" air cooling (~100W) given chip packaging constraints and operating temperature limits. It is also clear that expectations for power dissipation are *rising* as time goes on. Another key metric is the heat flux at the chip level, predictions for which are shown in Figure 2 [1,2,3].

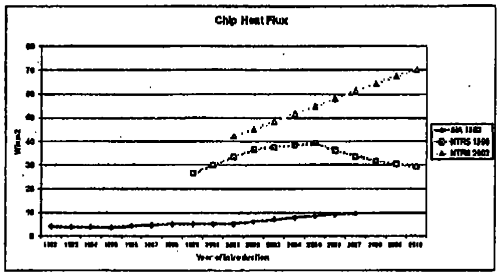

Figure 2: Chip heat flux forecast

While much attention has been placed on these thermal limits, cooling is also an issue at the data center level. Figure 3 portrays the trends in terms of overall power dissipation per unit floor area. Both telecommunications equipment rooms (both central offices and remote huts) and computer rooms have reached their limit for thermal density given current cooling approaches relying on in-room air conditioning units.

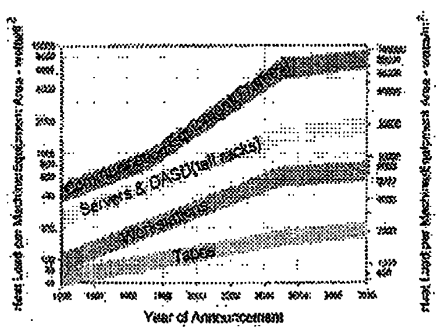

Figure 3: Rack/equipment room heat load forecast [4]

Military systems typically must operate in significantly more taxing environmental conditions than their commercial counterparts. For certain systems, such as phased array radars, isothermal cooling (that is, thermal control subsystems providing near-isothermal conditions across the system) is important. For others, such as high-energy lasers, near-isothermal conditions are coupled with a requirement to handle extremely high heat fluxes. PLMC is capable of providing such thermal management. Thermal Form and Function LLC (TFF) is participating in a program sponsored by the Missile Defense Agency (MDA) that will characterize PLMC as applied to the radar cooling problem; the results of this program will be complementary to TFF's product development efforts for computer and other electronics and electro-optical cooling.

This paper will describe the basic PLMC technology, provide early results for commercial applications at several levels of the hardware hierarchy, and outline the application of this uniquely capable cooling technology to other military and industrial systems.

PUMPED LIQUID MULTIPHASE COOLING

In its simplest form, PLMC may be thought of as an *externally pumped heat pipe* thermal management system. Heat pipes are well known for their effectiveness in handling high heat fluxes with a very low temperature rise, but are inherently inflexible in their application (e.g., they are sensitive to orientation) and limited in the distance over which the collected energy can be transported. Heat pipes use capillary pumping to return liquid from the condenser end to the evaporator (heat load) end of the device.

PLMC eliminates wick structures for pumping and instead employs a high reliability, low flow rate pump to return condensed liquid to the evaporator / cold plate. The basic PLMC flow loop is shown in Figure 4.

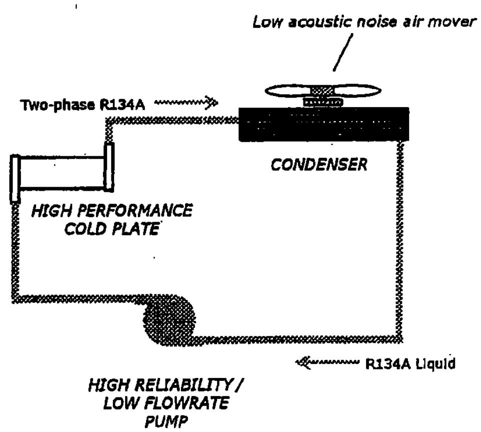

Figure 4: PLMC flow loop

The fluid (in this case refrigerant R134A) essentially goes through a Rankine cycle. In the cold plate, heat from the device to be cooled partially evaporates the coolant, thus providing a very high effective heat transfer coefficient (TFF has devised patent-pending cold plate structures that optimize the heat transfer in the cold plate). The coolant (ideally a two-phase mixture with a quality of perhaps 30%) then flows to an air- or water-cooled condenser (in Figure 4, an air-cooled condenser is shown) where heat is rejected and the fluid returns to the single-phase liquid state. A low-power, low-flow rate pump then returns the liquid to the cold plate.

PLMC has many advantages over other cooling techniques. Very high local heat transfer coefficients are possible in properly designed cold plates. Heat transport is very efficient; a kilowatt can be transported hundreds of meters with a pump that requires only a few watts of power. PLMC uses component technology that for the most part is proven and used in other applications. Near-isothermal cooling is possible for a number of devices in a system. When properly designed, a PLMC system is guaranteed not to suffer from environmental water condensation. In short, PLMC technology provides the highest performance, lowest cost, and most compact cooling for electronics and other systems available today. Finally, the approach is highly scalable – effective and efficient for applications with 100W heat loads (computers) as well as applications with 10s of kW heat loads (radar).

It is instructive to compare and contrast PLMC with single-phase liquid cooling using either standard cold plates or advanced microchannel heat sinks for electronics devices. Table 1 below provides comparative data for a 200W heat load operating in normal ambient conditions.

	Single-phase liquid loop	PLMC
Working fluid	Water	R134A
Flow rate (l/h)	35	7.5
Pump power (W)	20	2
Coolant temperature rise (deg C)	10	~0
Relative condenser size	2	1

Table 1: PLMC vs. liquid cooling

As can be seen, even if liquid cooling could provide the requisite thermal performance to maintain the operational limits of a 200W processor chip, PLMC provides significant benefits from efficiency, size, and weight perspectives.

APPLICATION TO RADAR SYSTEM COOLING

Phased array radar systems can benefit from the advantages of PLMC, which promises significantly reduced cost, size, weight and power over the currently used single-phase liquid cooling and vapor compression two-phase designs.

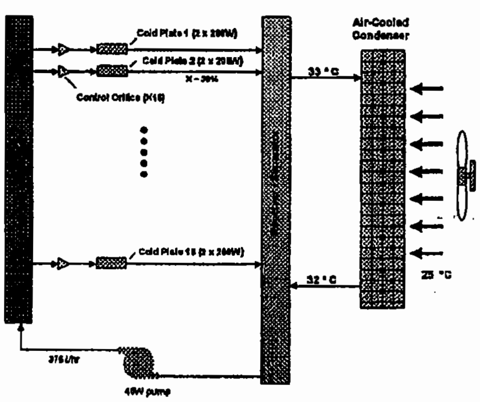

Figure 5: Demonstration radar cooling unit

Thermal Form and Function has built and tested a demonstration cooling system similar in size and power density to important radar cooling applications. The goal of the project was twofold: first, to demonstrate in hardware a PLMC system capable of isothermally cooling a total of 16 400W heat loads (for a total of 6.4 kW) at working-fluid temperatures below 33 °C, and second, to successfully design and test a 400W heat exchanger / cold plate for the system with a fluid-to-cold-plate-surface thermal resistance of less than 0.02 K/W. The demonstration system block diagram is shown in Figure 5.

The demonstration unit was thoroughly instrumented to measure temperatures and pressures at key points in the working fluid cycle, and was equipped with sight-glass monitors to assess fluid quality. At the design heat load of 6.4 kW (16-400W cold plates) and 25 °C ambient air temperature, the working fluid temperature was maintained at a (saturation) temperature of ~32 °C with a relatively modest 376 l/h coolant flow rate. A 30% quality was desired, and achieved, at the exit of the cold plates; this provides a safety factor for cold plate dry-out. The system was stable, easily controllable (through fan speed), and provided essentially isothermal conditions for all cold plates.

In order to meet design objectives, custom cold plates were fabricated using TF&F's patented copper convoluted fin construction. Two 200W thermal loads (thin film heaters) were used for each cold plate. The cold plate design is shown in Figure 6 below.

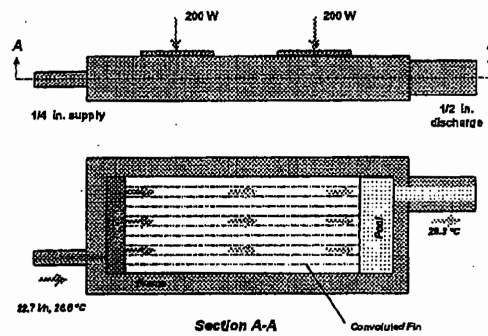

Figure 6: Demonstration unit cold plate (2.875 x 21.9 x 1.6 cm)

Typical operating conditions are indicated in the figure. Test data from 21 separate runs were used to determine the effective fluid-to-frame thermal resistance; values ranged from 0.009 K/W to 0.011 K/W, thus meeting the design objectives.

COMMERCIALIZATION

PLMC technology is applicable to a variety of electrical, electronic, and optical systems. The primary market target for TF&F is the information technology area (computers and telecommunications equipment).

As was pointed out in the Introduction, thermal management has become a serious problem at several levels, from hot spots on microprocessor chips, through processor packages, boxes, racks, and equipment rooms. PLMC is uniquely capable as a platform for computer cooling because it can address cooling issues at all these levels.

For personal computers and workstations, acoustic noise generation is a serious ergonomic problem. Even if thermal performance goals can be met with active air-cooled heat sinks, a significant acoustic benefit can be gained by using PLMC. This is because for moderate heat loads the PLMC condenser can be cooled via natural convection or using a low-speed, inherently quiet air mover.

For high-performance 1U and blade server configurations, thermal performance has become a limiting factor both for microprocessor operating frequency and processor density (that is, the number of processors per box). TF&F is currently working with OEMs on the implementation of PLMC for these configurations; a mockup of a 1U server application with dual processors, each dissipating 150W is shown as Figure 7.

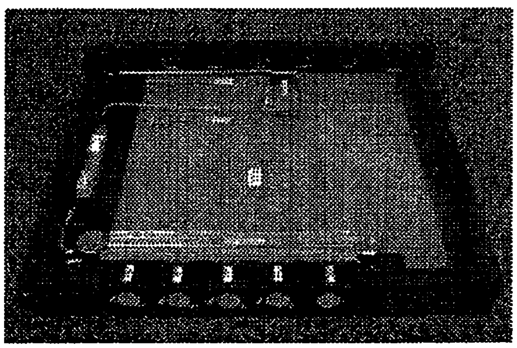

Figure 7: 1U PLMC mockup

Dense, rack-based systems can be effectively addressed using a single pump, single condenser unit that allows hot swapping of PLMC-cooled modules. TF&F has prototyped such a unit aimed at the VITA34 backplane market with an equipment OEM; a drawing of this system is shown in Figure 8.

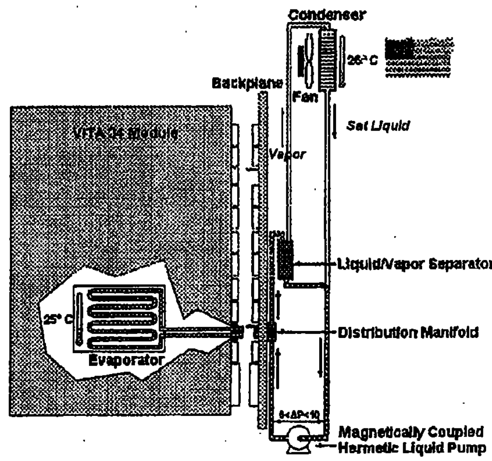

Figure 8: VITA34 multi-module system

A major pain point for operators of major computer room installations is the per-rack power dissipation, which will more than double over the next few years. Energy efficiency is also of serious concern at this level; over 40% of the energy used in "typical" installations is consumed by cooling equipment, according to a major computer room air conditioning manufacturer. State and federal governments are considering energy efficiency statutes for such installations.

PLMC is applicable for this problem as well. Shown in Figure 9 is a concept for a rack intercooler unit that provides thermal management within a densely populated equipment frame and draws on PLMC's capabilities in moving heat over long distances – the condenser can be located either at a remote location within the equipment room or outside the room altogether (for example, on the building roof).

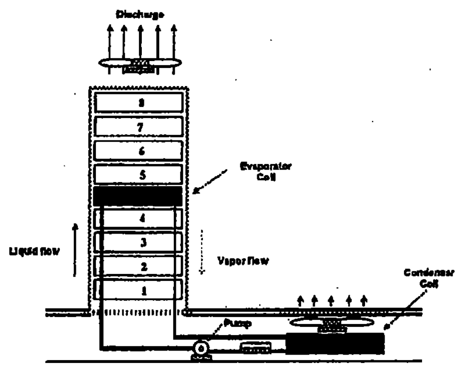

Figure 9: Rack Intercooler System

CONCLUSION

Thermal performance limits are being reached in a variety of applications, from microprocessors to large radar systems. Pumped liquid multiphase cooling (PLMC), essentially an externally pumped heat pipe cooling approach, promises to be effective across many applications. It is highly scalable, very energy efficient, and can provide significant performance, cost, size, and reliability benefits as compared with existing active air cooling or simple liquid-loop systems. While PLMC can be implemented with a variety of working fluids, Thermal Form & Function has favored R134A refrigerant, which is non-toxic, widely available, and environmentally friendly.

TF&F's technology and product development has been materially aided by applying PLMC to military and industrial systems, and is currently being commercialized as the ideal solution for the growing thermal management problems in commercial microelectronics.

The concepts described in this paper are protected by issued and pending US and foreign patents.

ACKNOWLEDGEMENTS

The authors gratefully acknowledge the support of Raytheon Integrated Defense Systems in Sudbury, MA as well as the technical contributions of Steve Lindquist and John Bowman of Hewlett-Packard and Vik Jegers of APW, Inc.

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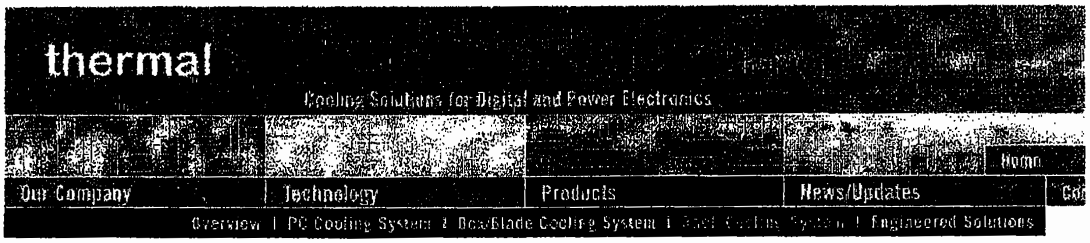


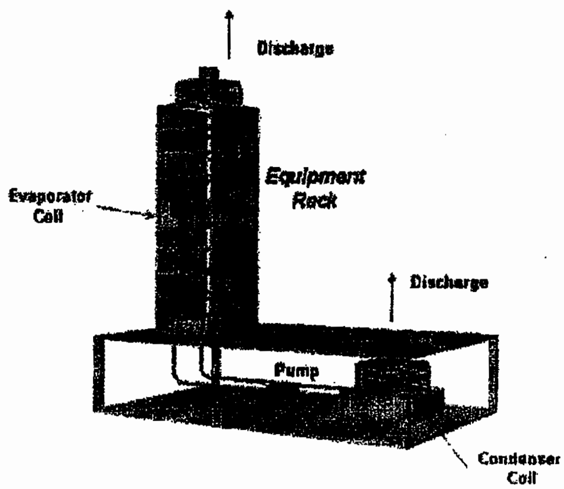
Rack Cooling System

Data center hot spots – racks with over 15kW of power dissipation – pose an increasingly prevalent problem.

Our Rack Cooling System (RCS) consists of a modular evaporator with up to 10 kW of heat removal capacity that can be mounted in high-power-density racks. The condenser unit can be remotely located, under the raised floor, outside the computer room, or even on the building roof.

The RCS has been designed for retrofit applications, and flexible refrigerant tubing is used for installation flexibility.

The RCS modules can be specifically designed for a variety of mounting schemes and form factors. Air or water-cooled condenser units can be used. Overall energy savings can be significant – a major OpEx advantage.



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Thermal Management System Using Pumped Liquid R-134a with Two Phase Heat Transfer

Marty Pitasi, Thermal Form & Function LLC, Manchester, MA

1. Summary

A new and versatile high performance, isothermal-cooling design, currently capable of cooling 6.4 kW at 33°C (91°F) with a COP of 100 is discussed. The system is a pumped liquid two-phase (2F) cooling design that uses a 1/20 HP motor, hermetic pump prime mover. The fluid is refrigerant 134a (R-134a), an ecologically friendly refrigerant used in car air conditioners. The design consists of pumping R-134a through a series of metered distribution lines that delivers refrigerant to heat exchangers/cold plates (evaporators). Heat is absorbed by the refrigerant changing it from a liquid to a vapor. The vapor is then converted back to a liquid via an air or liquid cooled condenser. This technique is scalable. A simplified 0.5kw system was built as a demonstration. Specifics of the design are discussed below.

2. Introduction

In anticipation of a 200wt. ALPHA processor (or equivalent) for the next generation of Enterprise Computer, Compaq Computer's Alpha Server Division surveyed all available cooling technology. Their goal was to establish a cooling strategy that would be applicable for future system and component cooling needs. As a result of the survey a pumped liquid 2F cooling system was identified and developed. This system is able to meet all design constraints including envelope size, cabinet integration, reliability, parasitic power limits, cooling demands, and control issues.

The proof of concept flow diagram (Figure 1) shows the R-134a fluid being pumped into a distribution manifold. Flow metering orifices uniformly distribute the R-134a into 16 parallel ¼" D lines. Custom 400 wt heaters simulate 2 x 200 wt processor packages converting the liquid to vapor. To enhance the system performance the vapor is discharged into a vapor-liquid separator where gravity causes the saturated liquid to displace the vapor to the condenser. Sub-cooled fluid flowing from the condenser mixes with the saturated fluid prior to the pump.

To prevent heater burn out, orifices are sized to deliver flow at rates that guarantee a vapor quality not to exceed 30%.

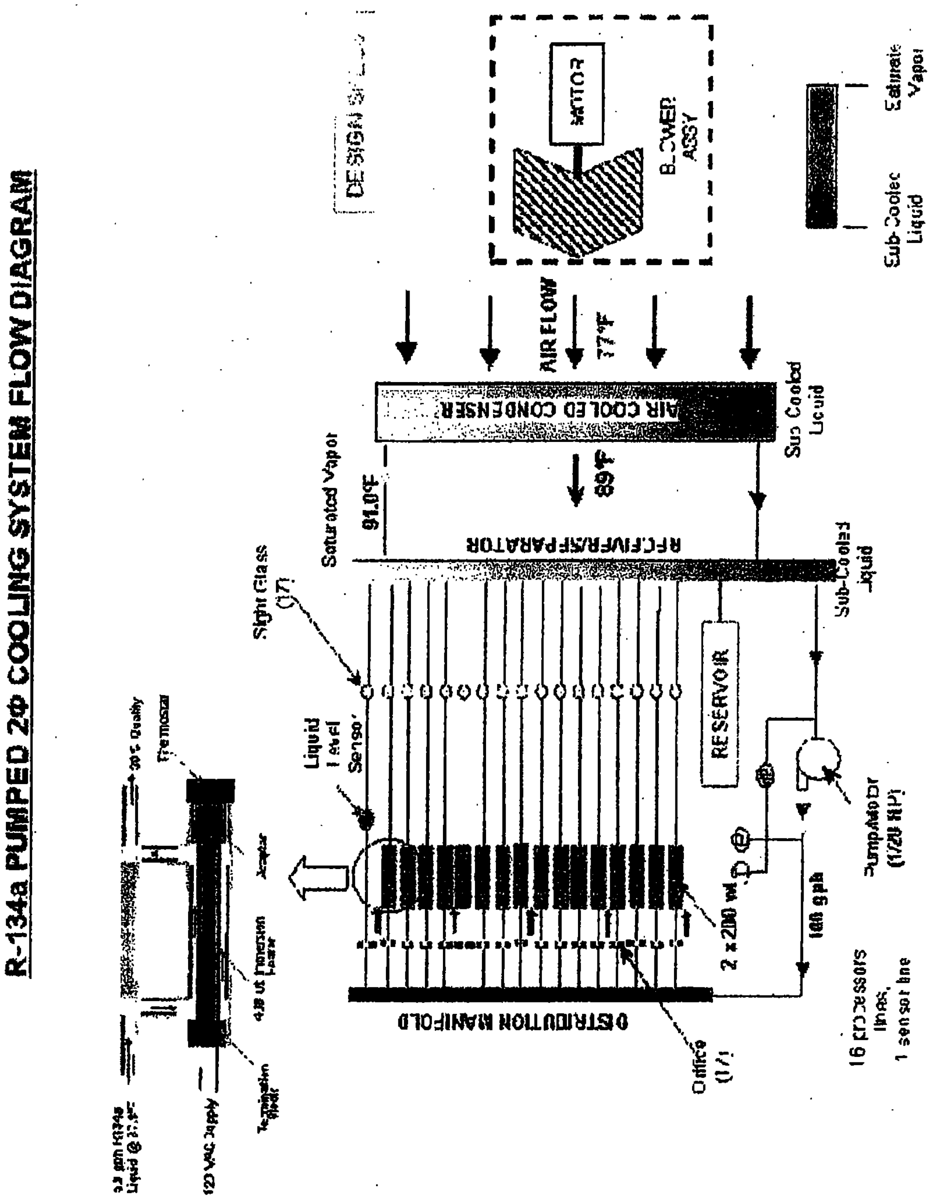

Figure 1 shows an additional line. Line 17 is at the top of the system and is used to continuously monitor the refrigerant charge.

The thermal budget limits the thermal resistance from the cold plate wetted surface to the saturated refrigerant to 0.011 °C/wt. See Table 1 in Figure 6 for details. Flow analysis of the problem showed that an inline convoluted strip fin design with a pitch of 10 fins/ inch and measuring 0.10"H x 2.00"W would meet the 0.011°C/wt requirement. The subsequent design shown in enclosure 2 depicts the heat exchanger flow components. Flow enters the heat exchange via a ¼" D line and pools behind a distribution manifold, then uniformly flows across the fins into a vapor reservoir prior to discharge via 1/2" D line. To ensure a uniform flow across the fins, therefore uniform cooling, the manifold flow loss is designed as the primary loss exceeding the fin, fluid expansion, and entrance and exit effect. The heat exchanger accommodates 2/200wt processor packages approximately 2" W x 2" D, resulting in an overall proof-of-concept cold plate size of approximately 2 7/8" W x 8 5/8"' L x 5/8" H. A sketch of the proposed final deign is shown in enclosure 3.

Briefly, the two goals are to build and demonstrate: (1) a proof-of-concept pumped liquid, two-phase (2F) cooling system capable of uniformly cooling 16/400 wt (6.4kw.) loads at temperatures below 33°C, and (2) a cold plate/heat exchanger with a refrigerant to wetted surface thermal resistance = 0.011°C/wt.

3. Results

A diagram of the proof-of-concept hardware is shown in Figure 1. As described earlier the primary system components of motor-pump, distribution manifold, orifices, vapor/liquid separator, and condenser are clearly visible. Other support hardware such as reservoir, liquid level sensor, and sight glasses are also shown.

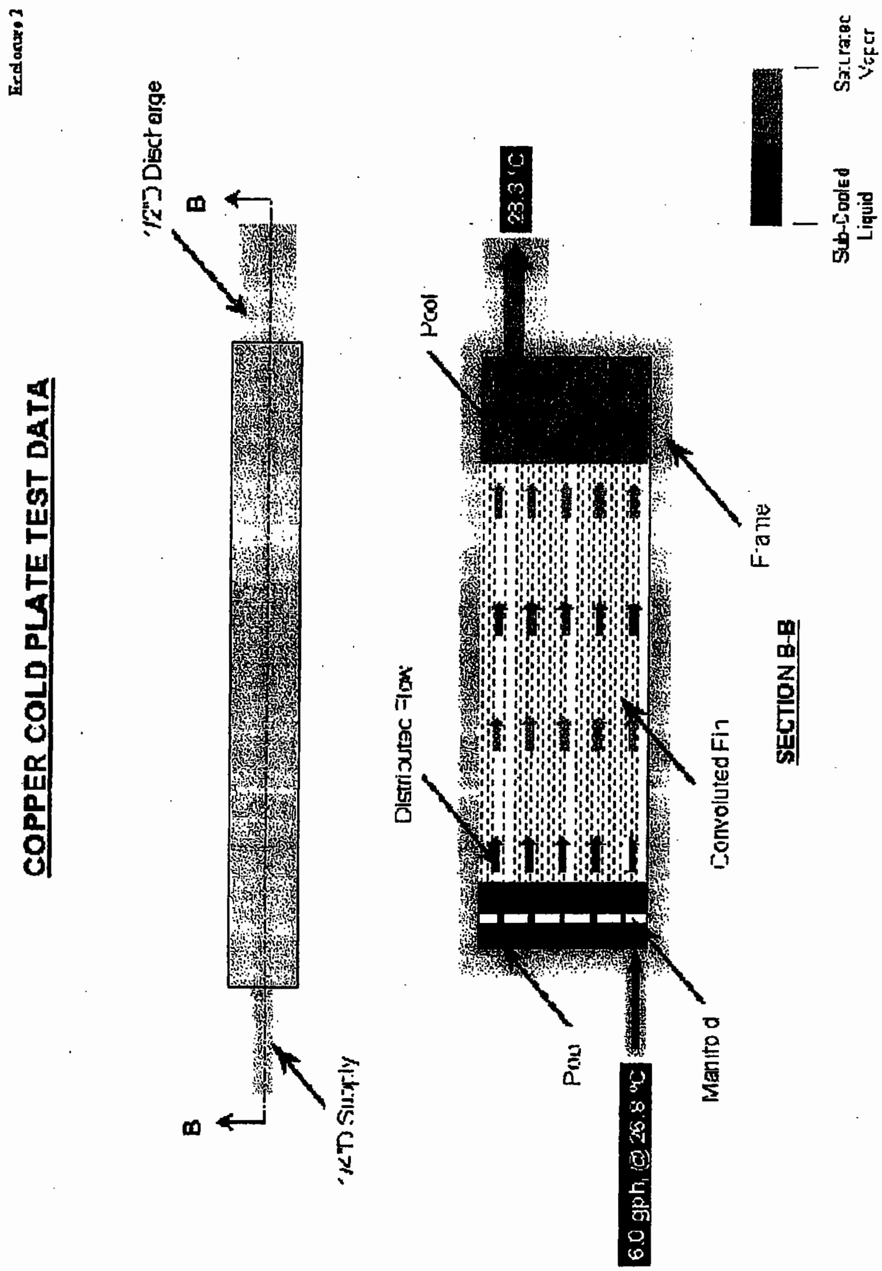

Figure 2. PROTOTYPE COLD PLATE FLOW CONCEPT

Specifics regarding the system design point were extracted from the R-134a phase diagram (Reference a) and the pump performance curve (Reference b). Please note that all pump flow rates are gallons per hour. The vapor enthalpy (hfg) for 30% quality and 33°C (91°F) produces approximately 6.4 wt-Hr/Lb. Combining this value with an overall load of 6.4kw and adjusting for the 17th leg results in a total flow requirement of 100 gph. Using the pump performance curve, figure 1 of Figure 4, a maximum allowable design pressure of 7.6 psi was extracted and used to determine the orifice sizes. As shown in item 2 of Figure 4, a minimum of 5.9 gph of R-134a at 33°C is required for each leg. However, in practice with off-the-shelf orifices, the flows ranged from 6.6 to 7.5 gph, thus, resulting in a maximum vapor quality of 27%.

Enclosure 3

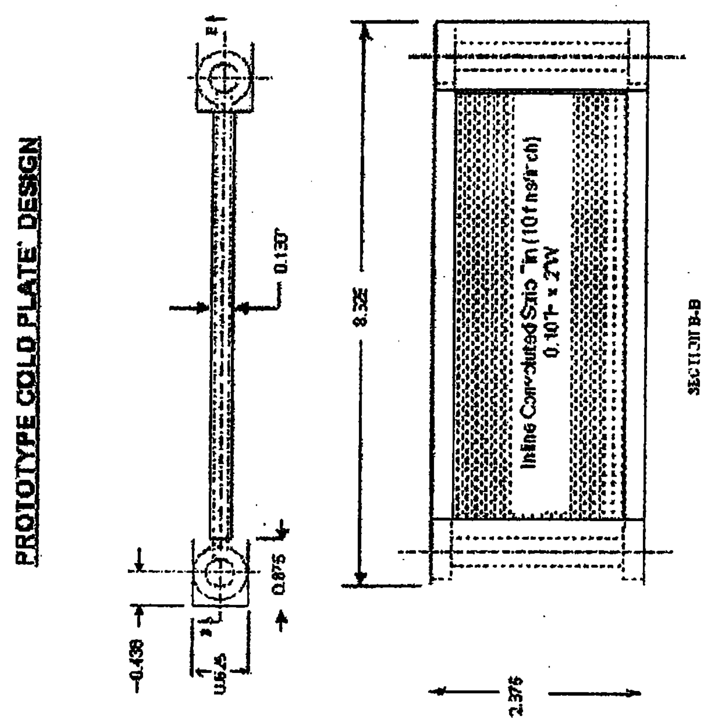

Figure 3. PROPOSED COLD PLATE

The assembled system integrated into Compaq GS-320 19" cabinet is shown in Figure 5. Note both air and liquid condensers included. Coefficient-of-Performance (COP) test results range from 17 to 100+ for air and liquid cooled condenser, respectively.

Enclosure 4

SYSTEM PERFORMANCE

Figure 1 Design Performance

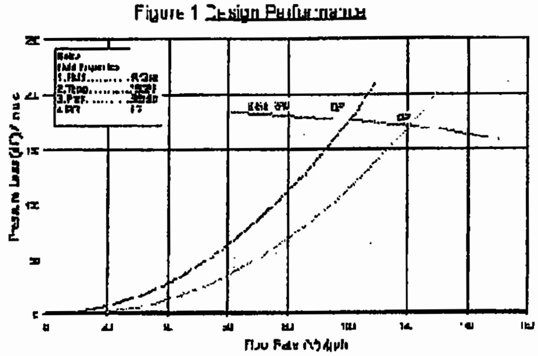

Figure 2: Performance Comparison

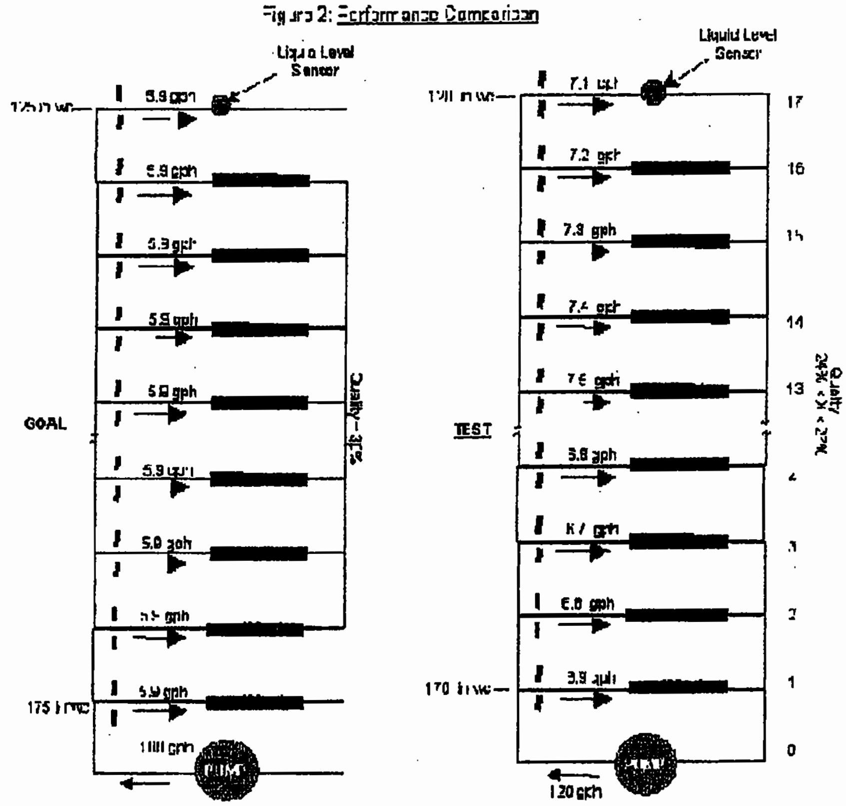

Figure 4. SYSTEM PERFORMANCE

To determine the thermal performance of the heat exchanger a heat balance was performed. Two 2" x2" film heaters simulate the package envelope, dissipating 200 wt each. They were mounted to the cold plate via an indium/gallium amalgam interface material. Thermal insulation was placed over the heaters to eliminates loses. Input power, flow, and critical temperatures were measured. The resulting maximum heat transfer coefficient extrapolated from the data was 0.011°C/wt. Figue 6 summarizes the data used.

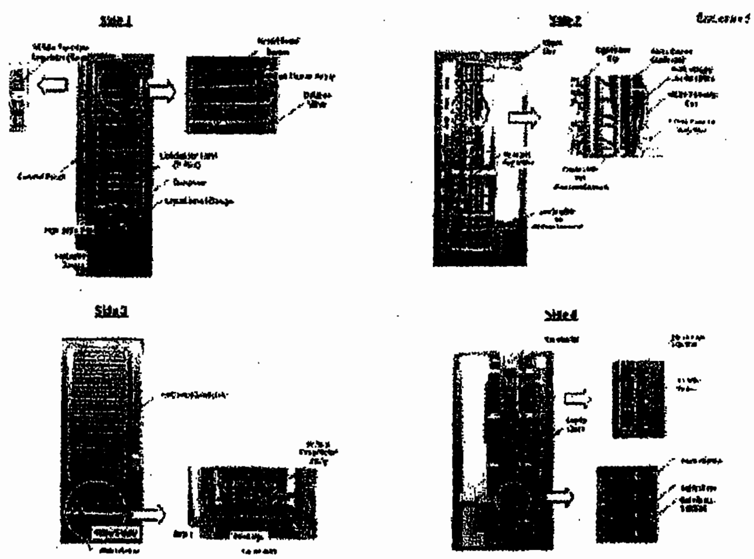

Figure 5. HARDWARE INTERGRATION

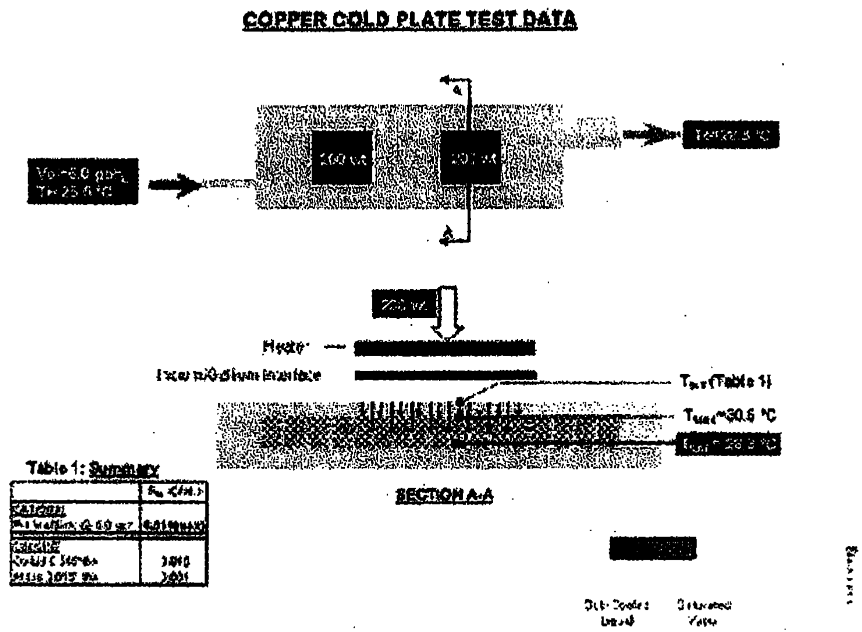

Figure 6. COLDPLATE PERFORMANCE SUMMARY

4. Discussion of Results

Test data from twenty-one (21) separate tests were used to statistically determine the cold plate/heat exchanger's thermal resistance. Values ranged from of 0.009 °C/wt to 0.011°C/wt.

Figure 5 shows the proof-of-concept system integrated into a typical Enterprise cabinet. The design requires approximately 5% of the cabinet volume and doesn't encroach into the sensitive electronics envelope. Not show, is the design strategy that allows for quickly hot swapping a board using quick disconnects (reference c)

By operating the system at 33 °C ambient, condensation issues are avoided. Any leakage takes the form of a non-toxic, non-contaminating gas. The 1/20 HP motor-hermetic pump moves 6.4 kW at 33 °C and has a documented minimum MTBF of greater than 50,000 hours (reference d). Control is inherent in the design. R-134a operating temperatures are managed by the condenser performance, and load variability is managed at the receiver/separator.

Coefficient of performance (COP) values range from 17 to 100+ for air and liquid cooled condensers, respectively. The only difference between the two is the fan power needed for the air-cooled condenser.

5. Patent Disclosure
The concepts described in this paper are covered by pending US and foreign patents.

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- Pump Reliability, Hy-Save Energy Conservation Technologies letter, Subject "Refrigerant Pump Reliability," date August 23,2001

Pumped Liquid/Two Phase Coo
for
High Performance Systems

Joe Marsala - Thermal Form & Function LLC

13 May 2003 – Scottsdale, AZ

thermalForm & function LLC

Options

- Air Cooled Heat Sinks
- Heat Pipe Technology
  - Circular Heat Pipes
  - Loop Heat Pipes
  - Vapor Chamber
- Thermoelectric Devices
- Pumped Liquid
  - Single Phase
- Vapor Compressi
  Refrigeration
- Pumped Liquid
  - Two Phase TF&F System

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- Parasitic Power Consumption
- COP
- Control
  - Condensation/Dew Point Management
  - Variable Load/ No Load Operation
- Reliability
- Cost
- Scalability
- Hot Swap/ Modul
- Size/ Weight
- Interface to Processor
  - Cold Plate/ Evapor
    UA Product

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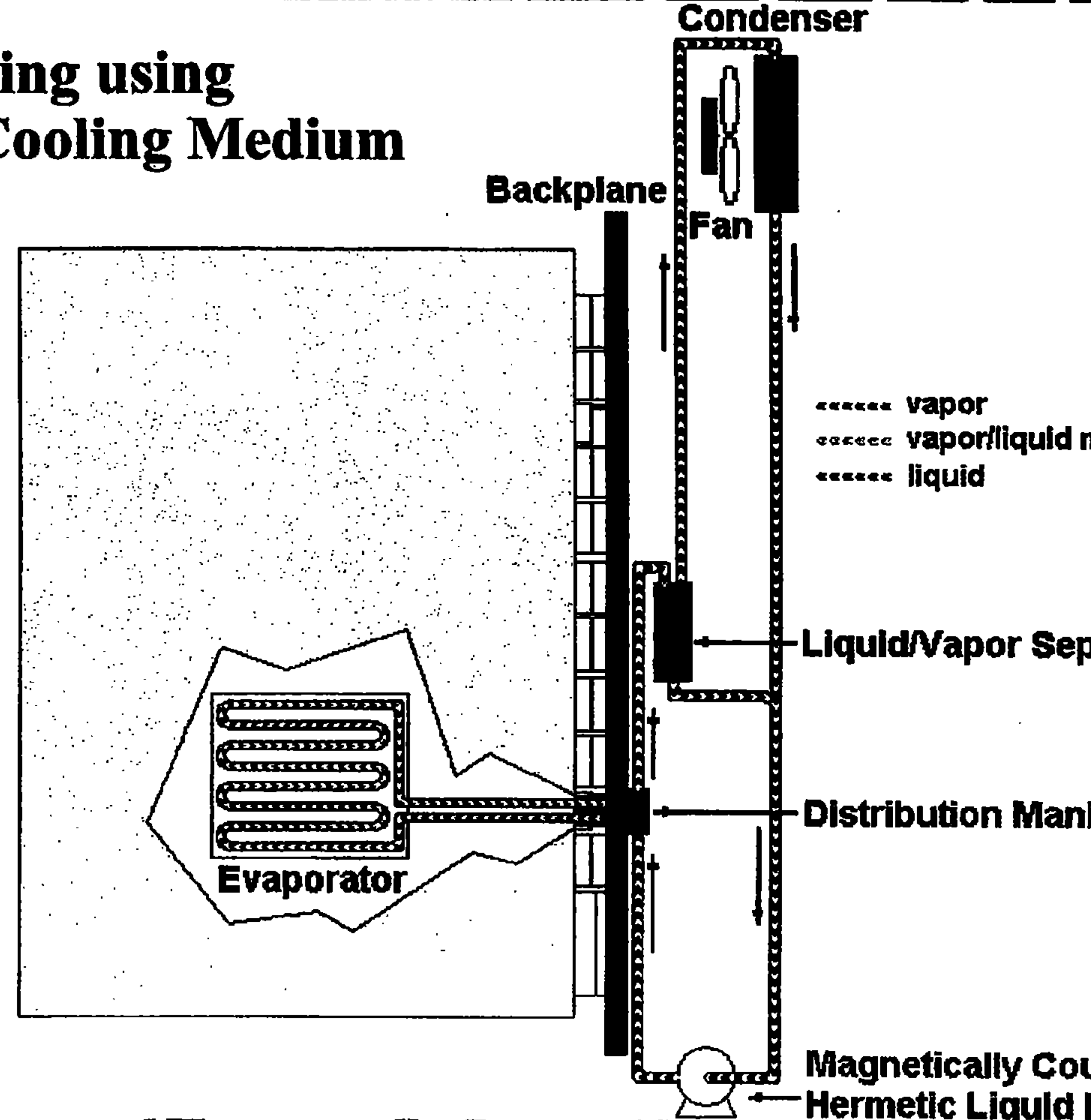
- Two Phase Cooling using Refrigerant as Cooling Medium



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	Pumped Liquid Single Phase	Vapor Compression Refrigeration	Pumped Two
Parasitic Power	[redacted]	2133W	[redacted]
COP air/water cooled	[redacted]	3	[redacted]
Control			
Condensation/Dew Point	[redacted]	[redacted]	[redacted]
Ambient Air Sink	[redacted]	[redacted]	[redacted]
Variable Load/No Load	[redacted]	[redacted]	[redacted]
Reliability	[redacted]	12K-20K hrs (compressor)	[redacted]
Cost	Intermediate	[redacted]	[redacted]
Cold Plate/Evaporator U	[redacted]	[redacted]	[redacted]
Required Area	[redacted]	[redacted]	[redacted]

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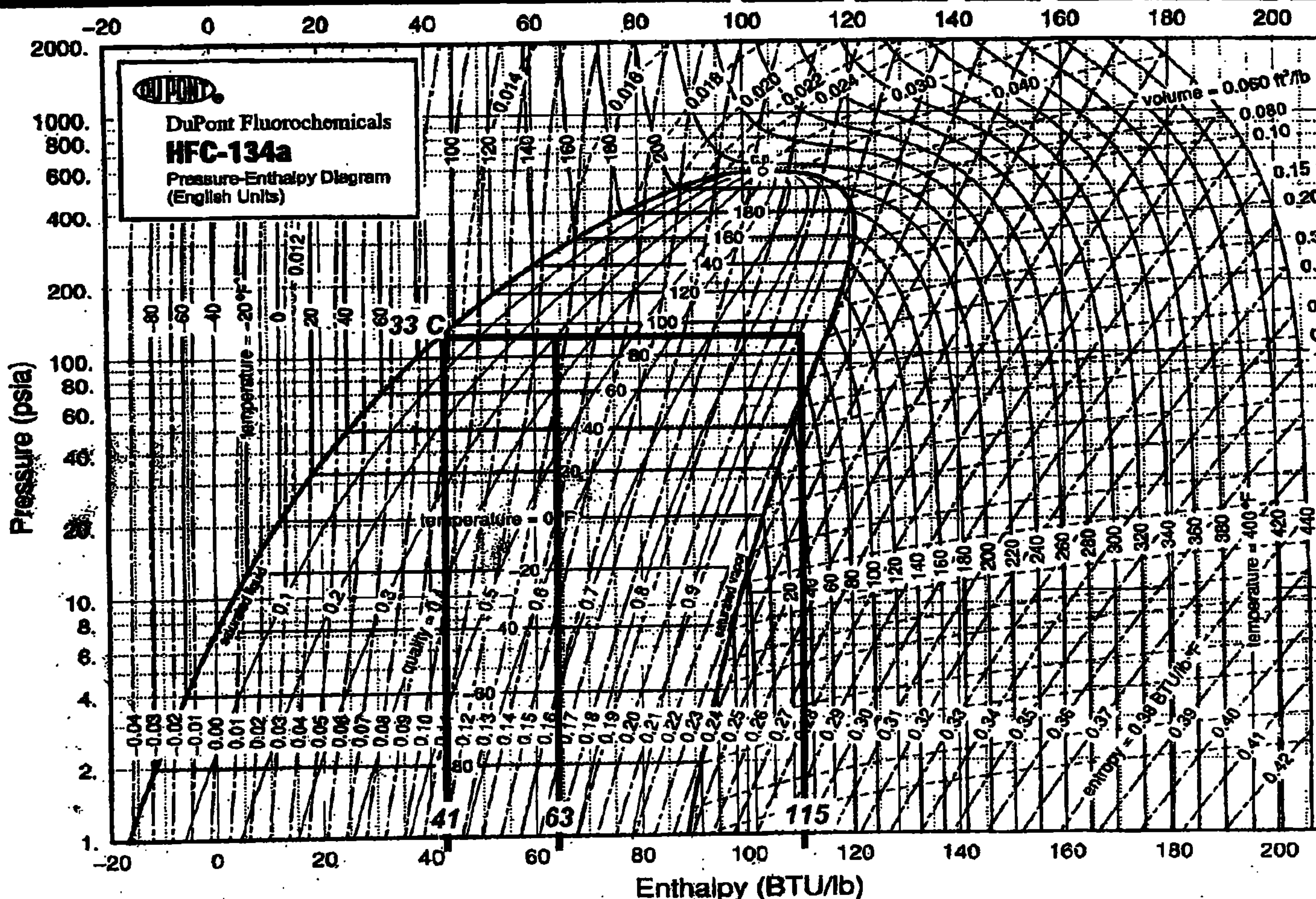

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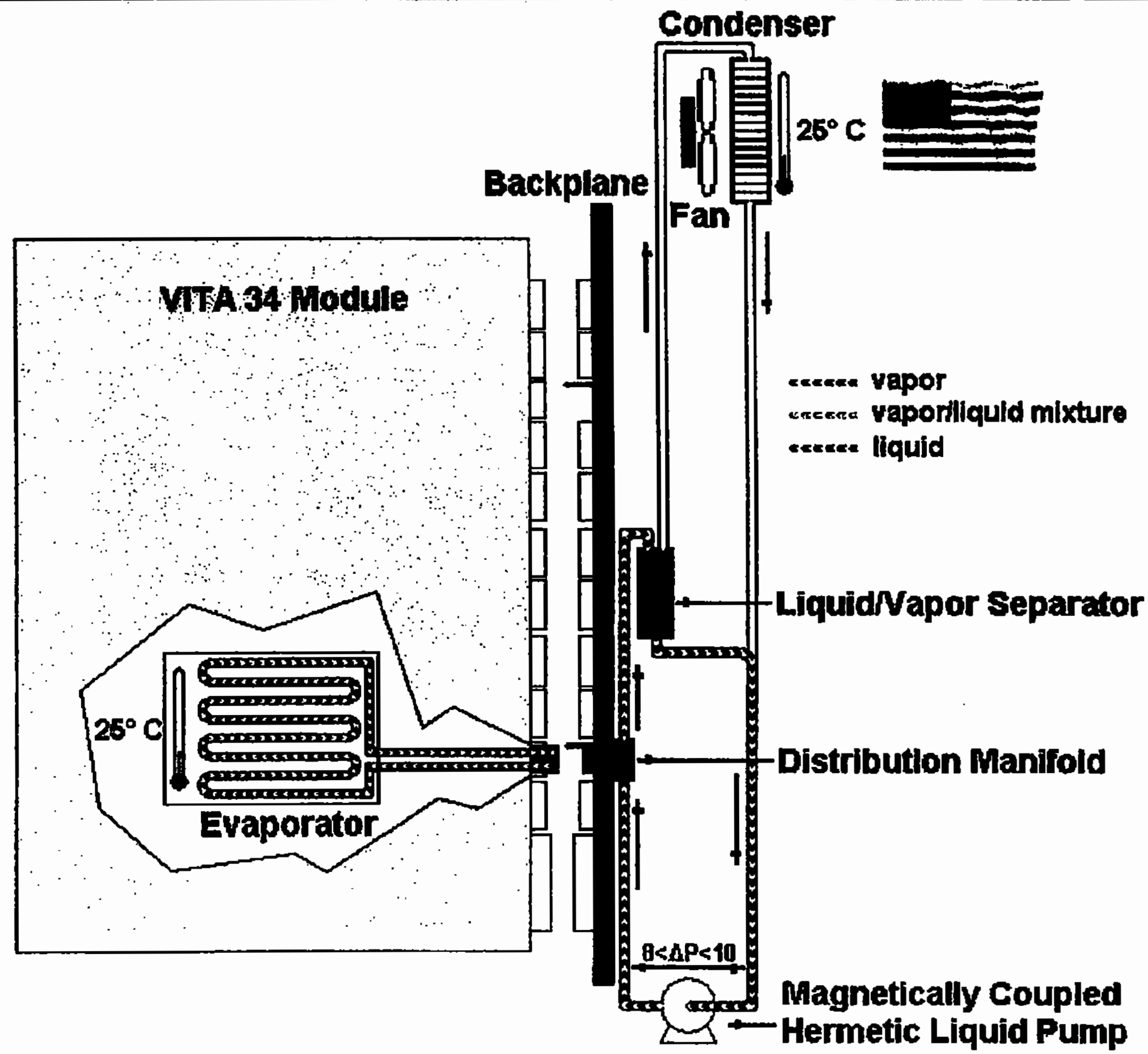

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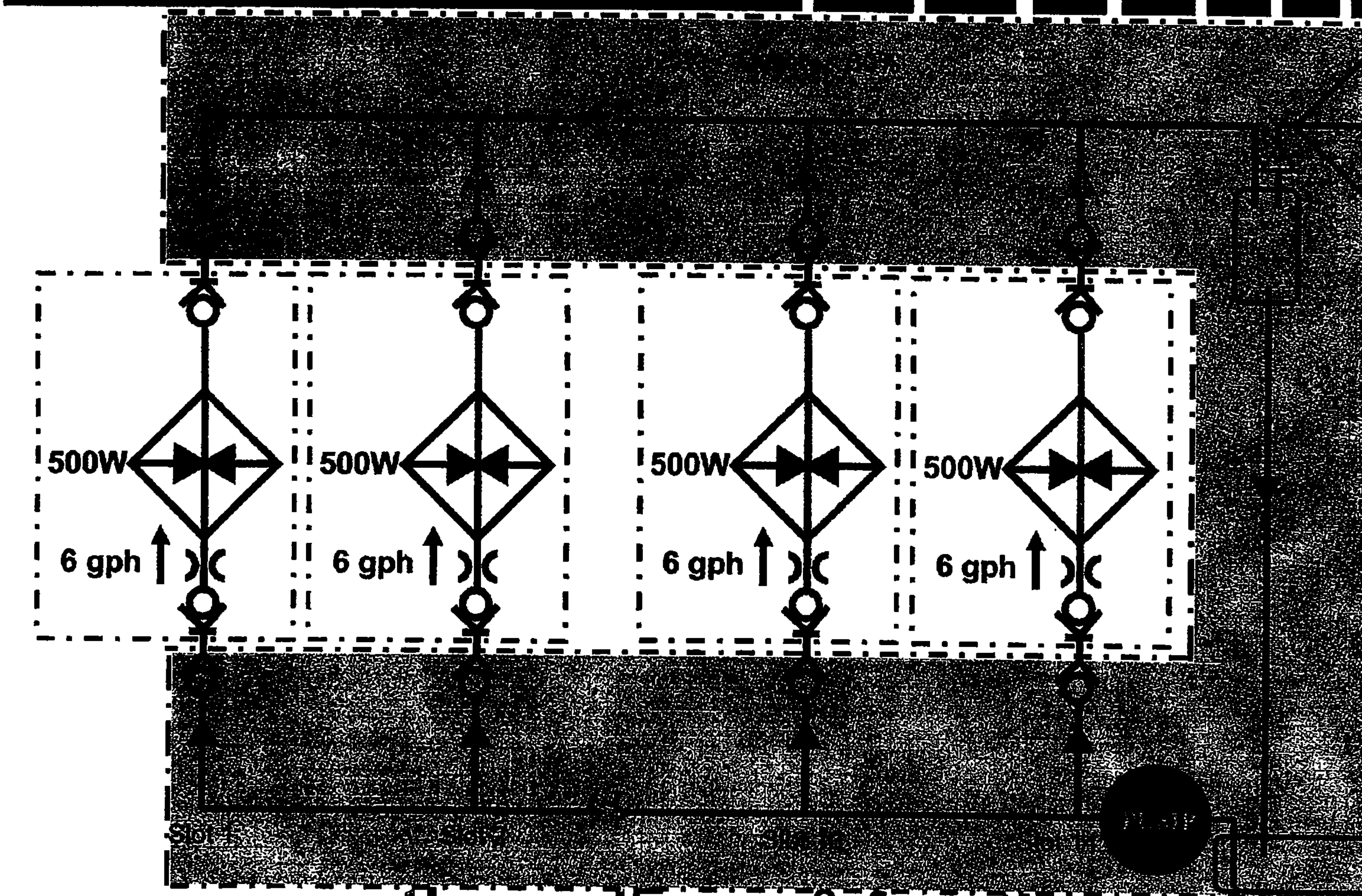

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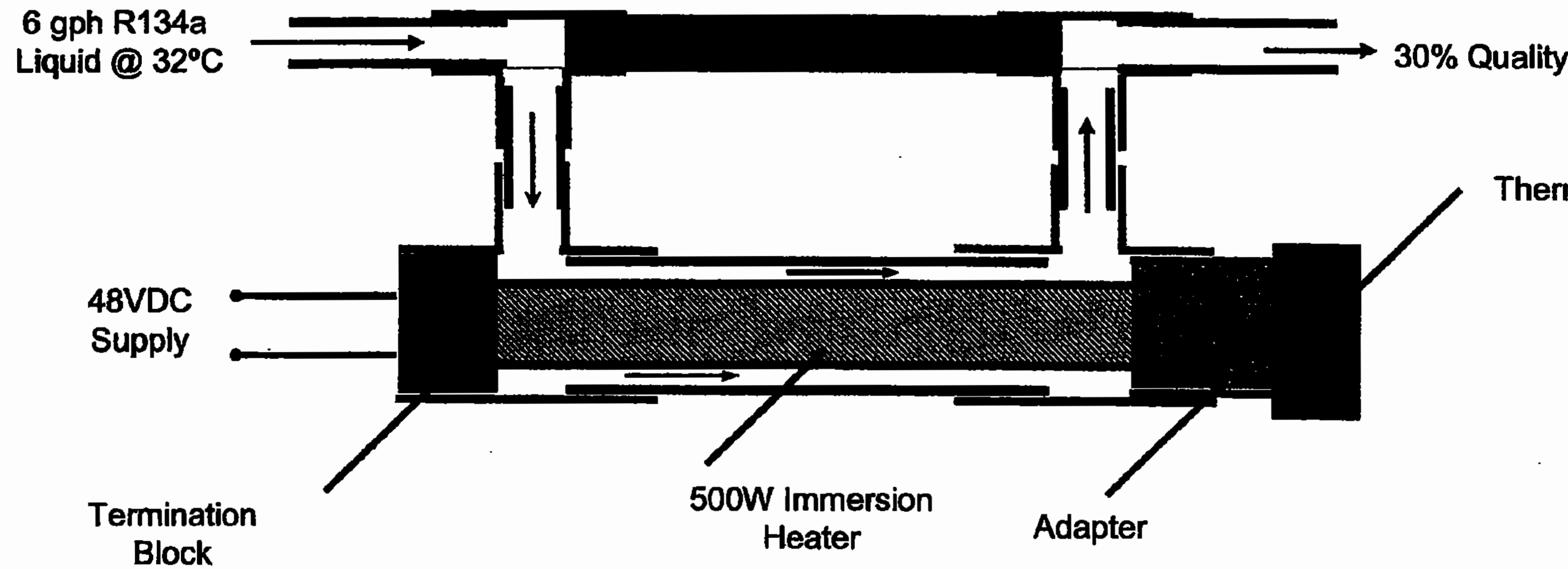

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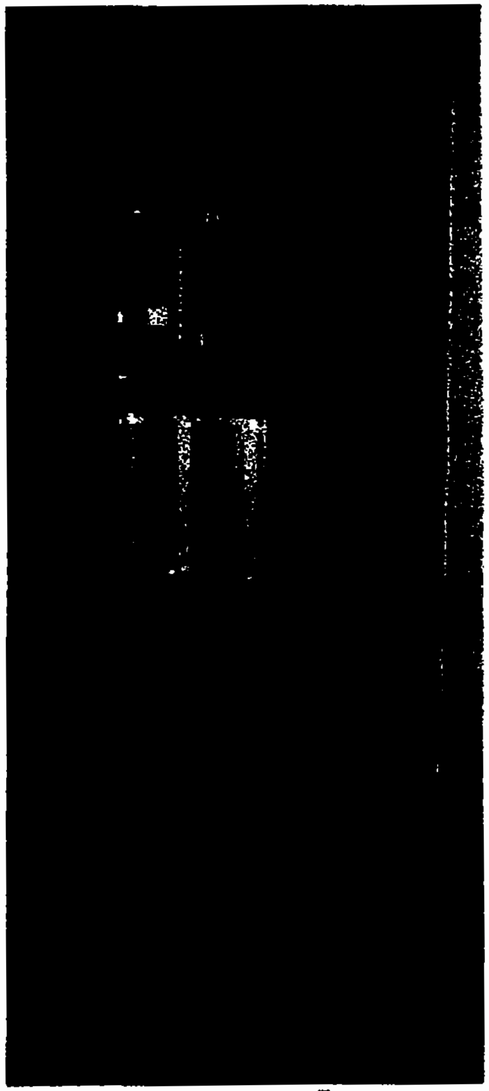
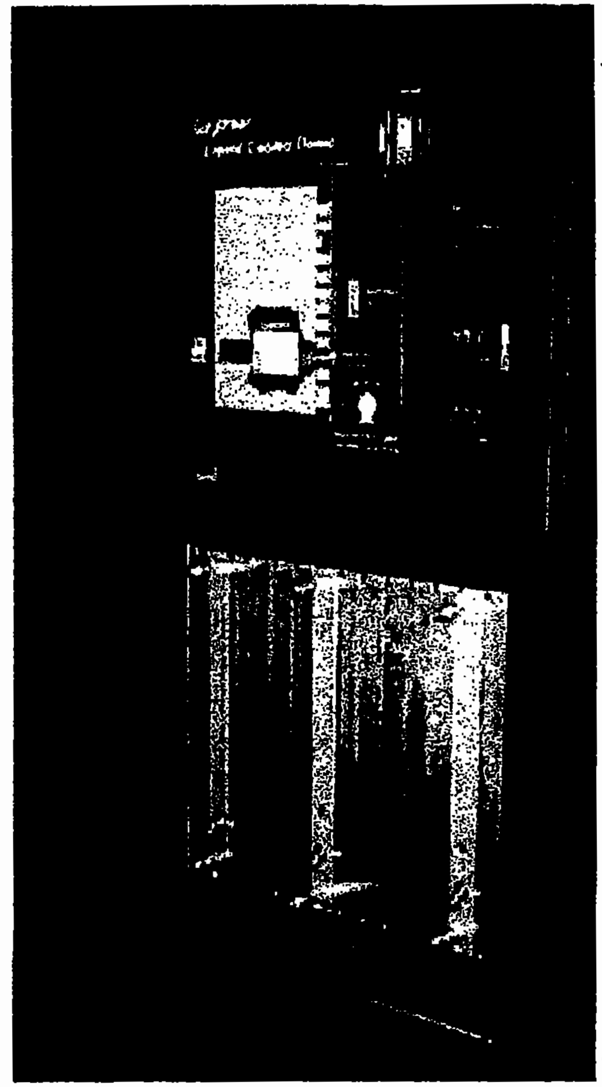
Implemented in a VITA 34 Prototy





Display

VITA 34 P
Chassis w
500W loa

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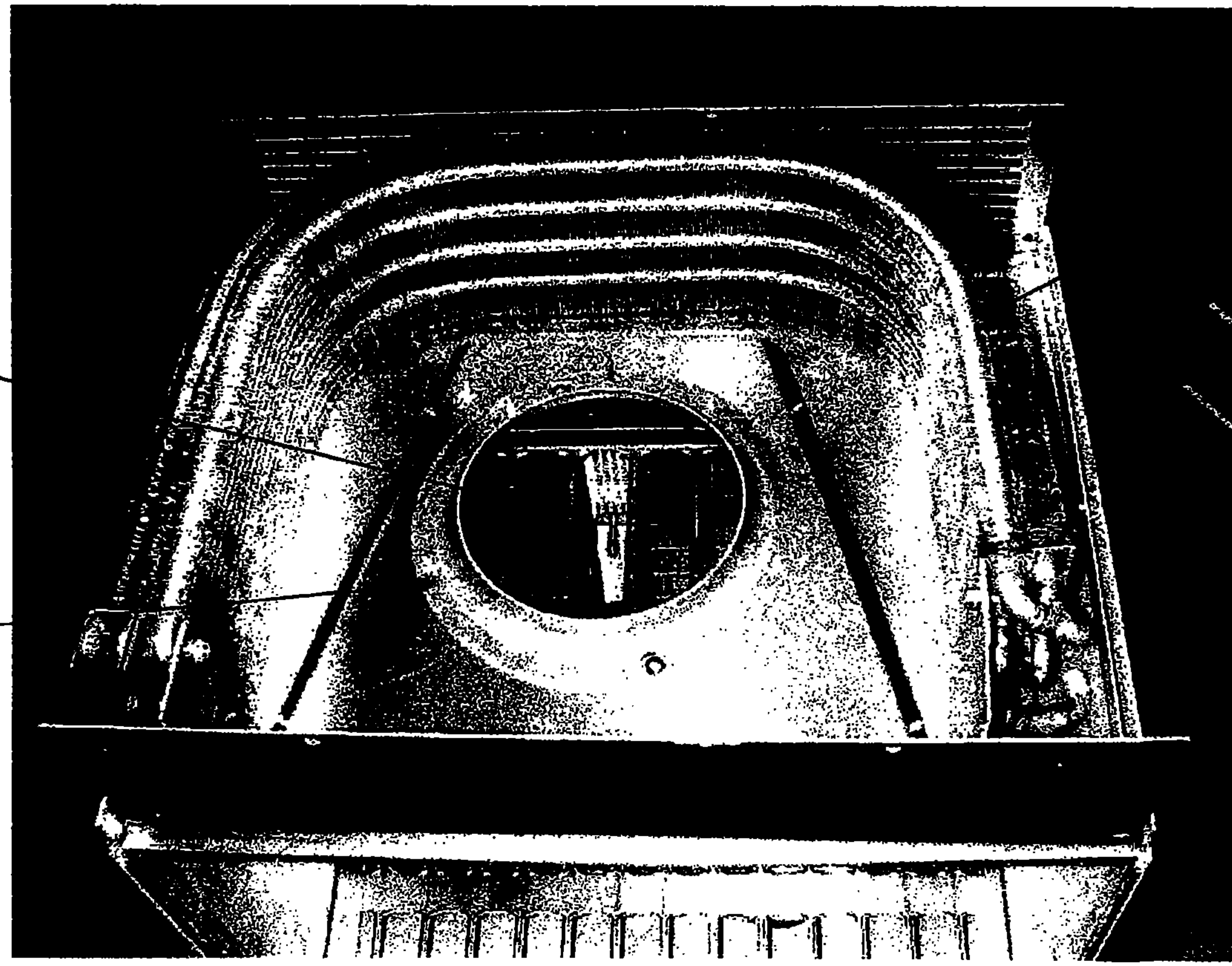
Implemented in a VITA 34 Prototy

VITA 34 Backplane

Fan Inlet Ring

C



Top View with top cover removed – Condenser Assembly

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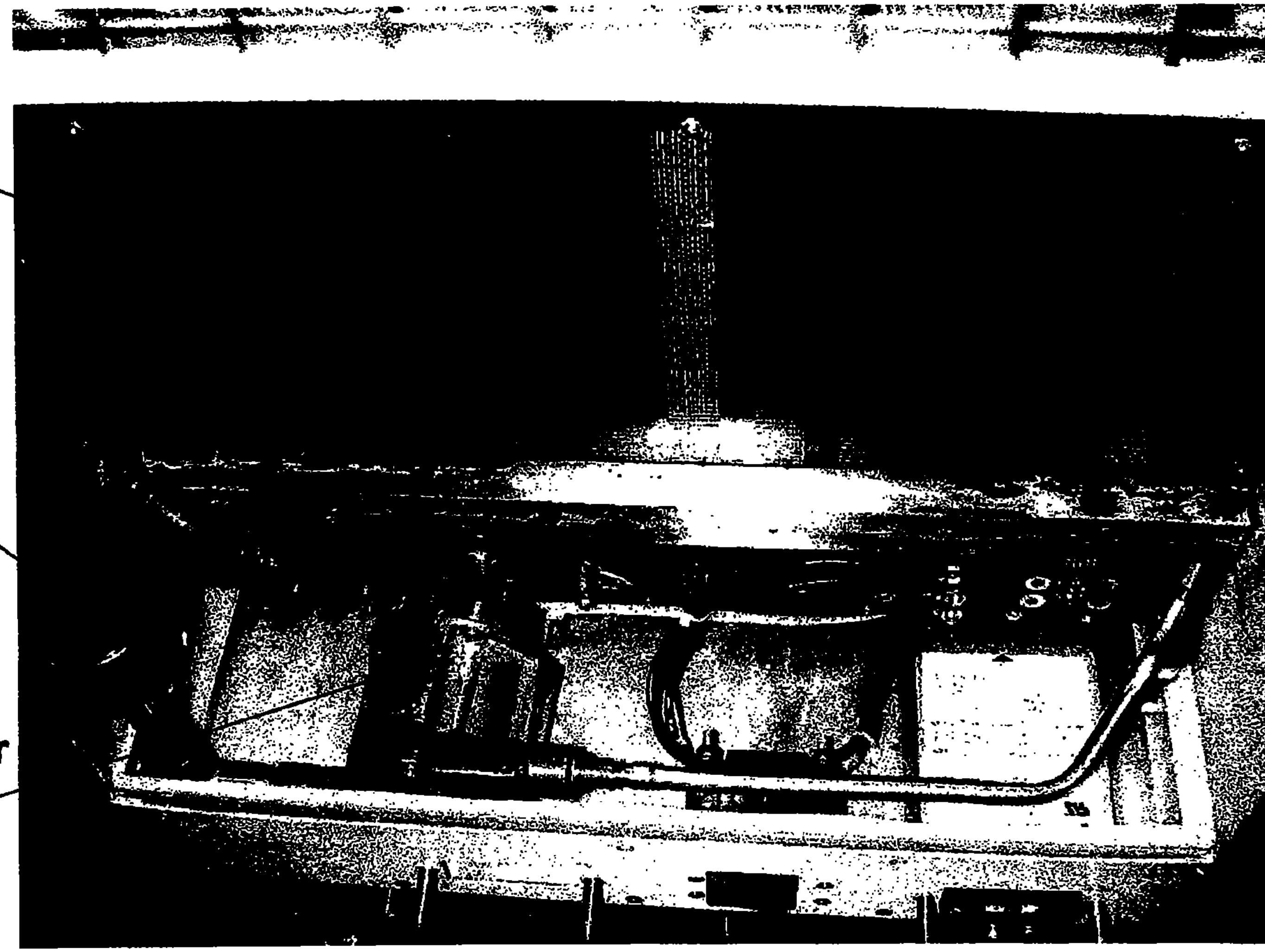

Rear View looking down into the chassis

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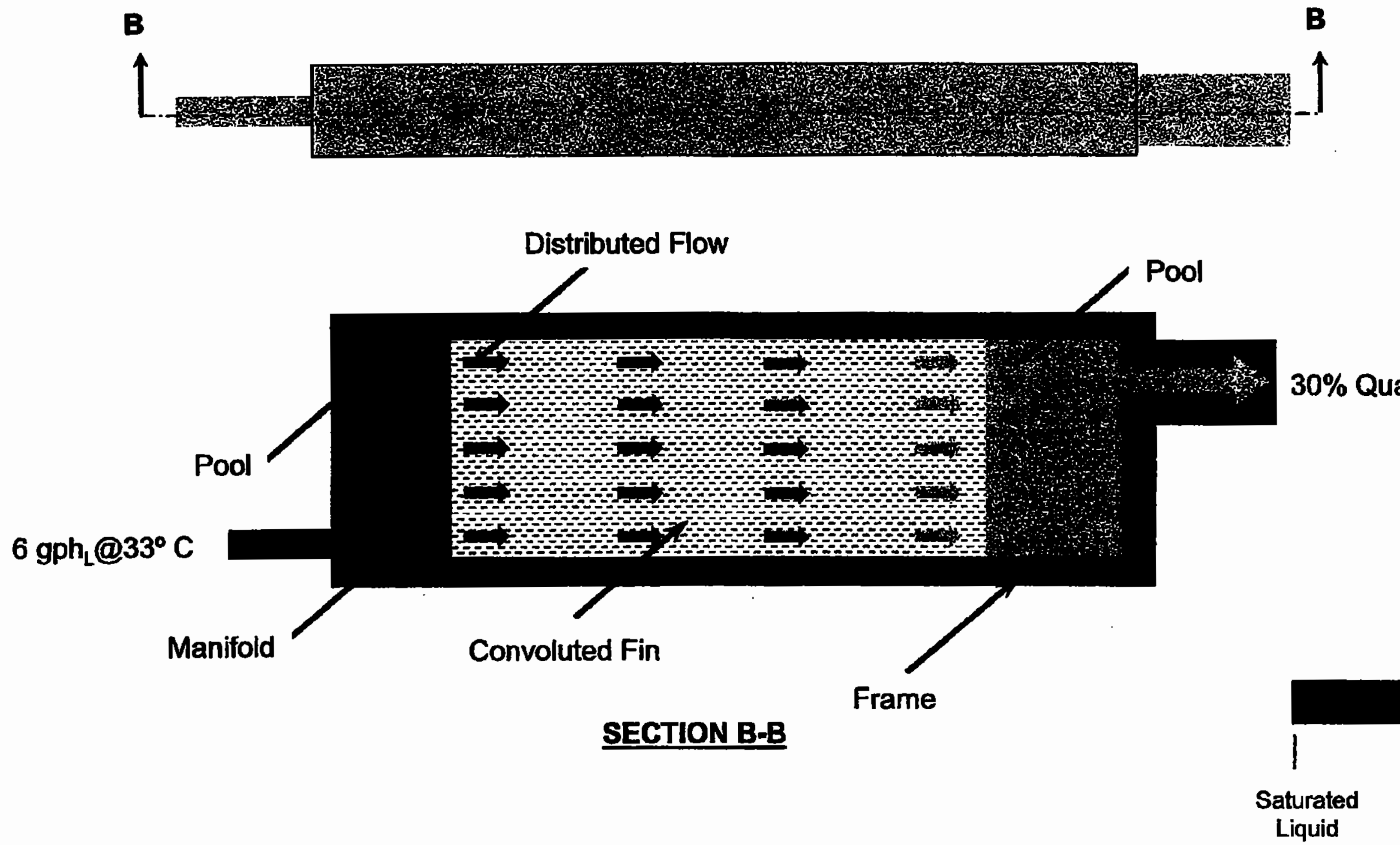

SECTION B-B

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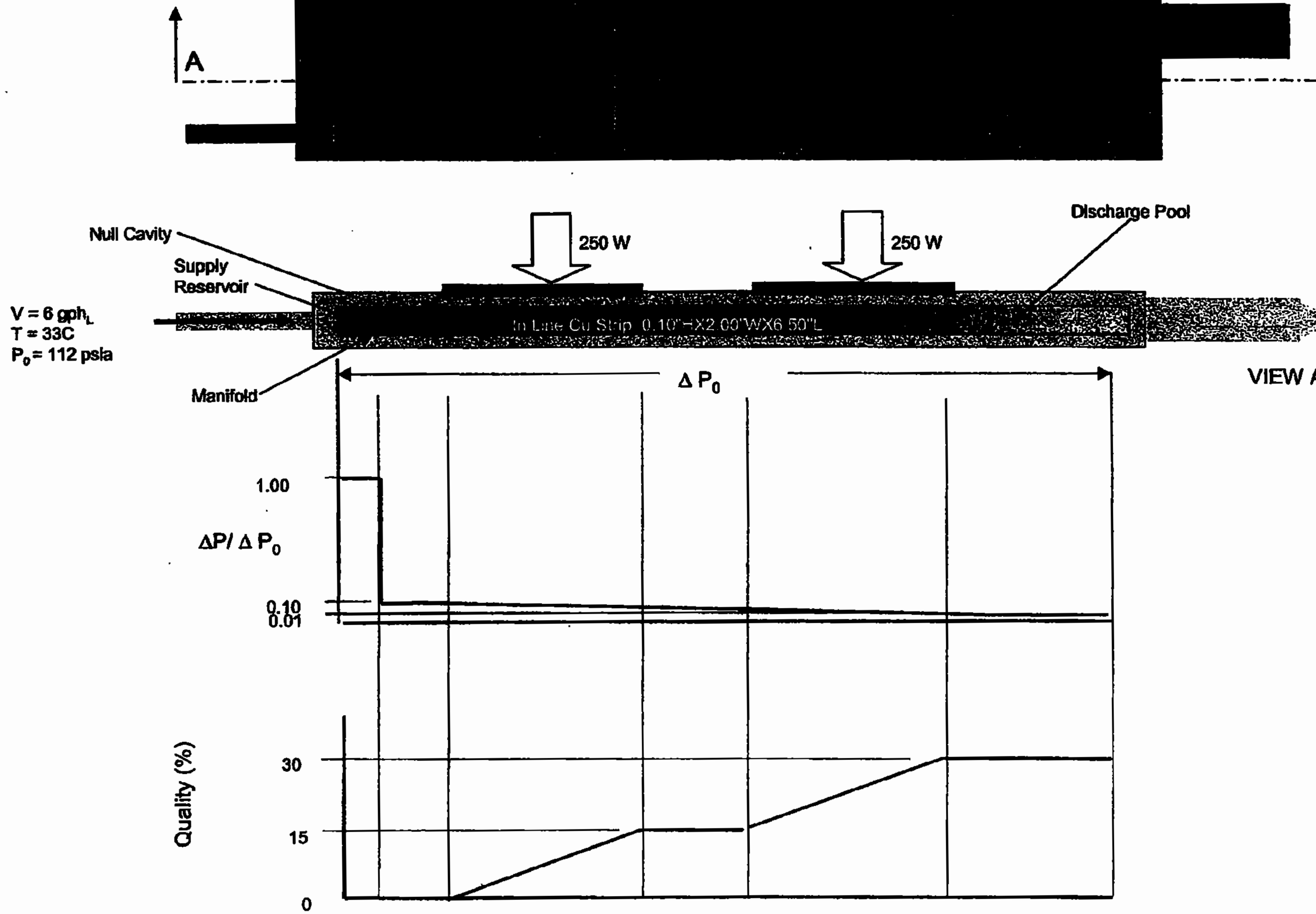
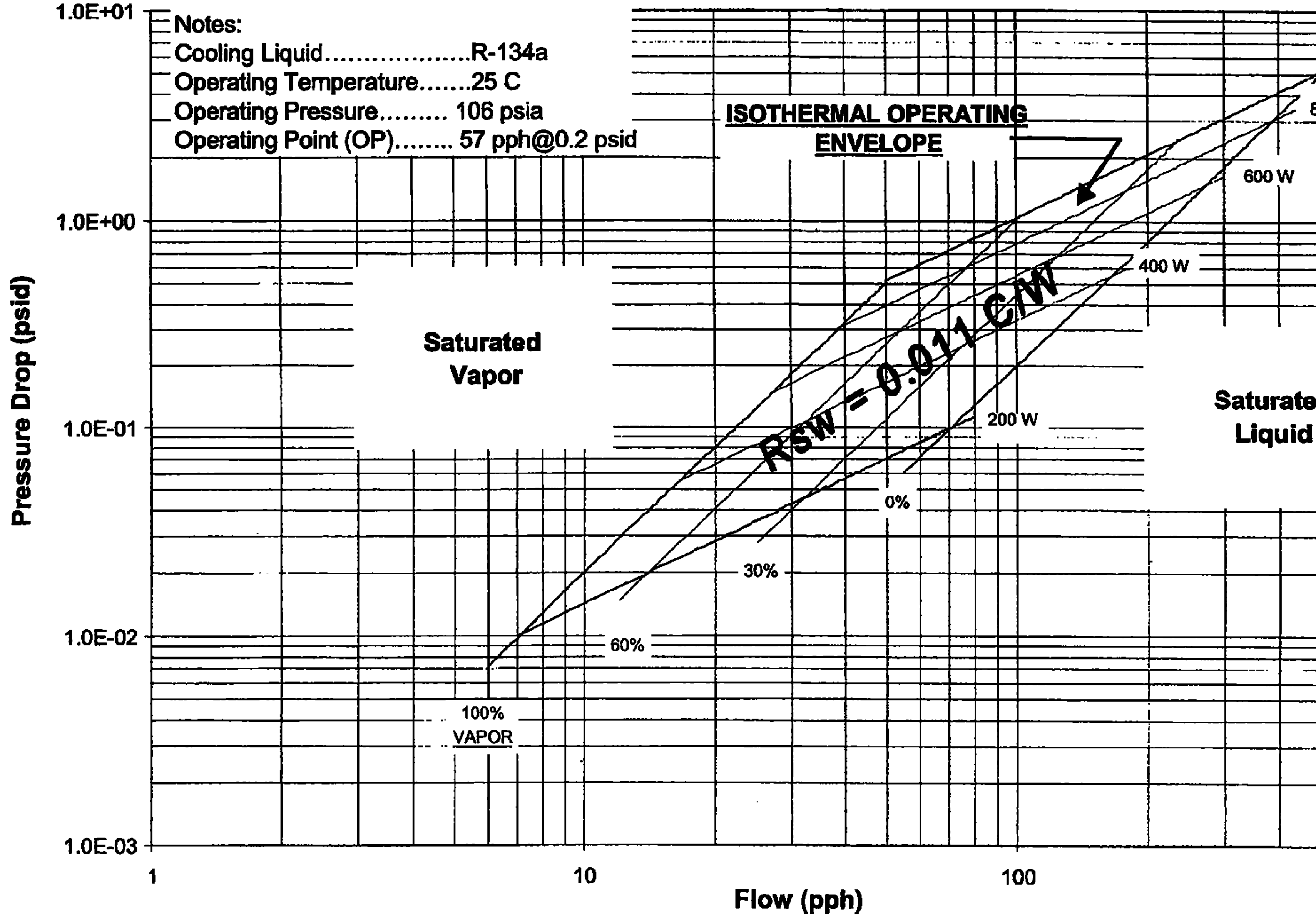

A
Null Cavity
Supply
Reservoir
V = 6 gph_L
T = 33C
P_0 = 112 psia
Manifold
250 W
250 W
Discharge Pool
In Line Cu Strip 0.10"HX2.00"WX6.50"L
Δ P_0
VIEW A
1.00
ΔP/ Δ P_0
0.10
0.01
Quality (%)
30
15
0


Notes:
Cooling Liquid....................R-134a
Operating Temperature.......25 C
Operating Pressure......... 106 psia
Operating Point (OP)........ 57 pph@0.2 psid
ISOTHERMAL OPERATING ENVELOPE
RSW = 0.011 C/W
Saturated Vapor
Saturated Liquid
600 W
400 W
200 W
0%
30%
60%
100% VAPOR
Pressure Drop (psid)
1.0E+01
1.0E+00
1.0E-01
1.0E-02
1.0E-03
1
10
100
Flow (pph)

- **Thermal Form & Function LLC owns issued and pending patents covering the concepts described in this presentation**

thermalForm & function LLC

- **Pumped liquid/ two phase coolin technology**
  - **High COP cooling method**
  - **Combines high heat removal of two phase systems with the reliability of liquid pumping**
  - **Uses well known cost effective components and materials from the HVAC industry in a novel configuratio**
  - **Modular, scalable and hot swappable**

thermalForm & function LLC

- Air is for breathing
- Water is for drinking
- Refrigerants are for heat transfer

thermalForm & function LLC

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Liebert

A Leading Practice White Paper From Liebert

Managing Extreme Heat

Cooling Strategies for High-Density Computer Systems

EMERSON. Network Power

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APPLICATION GUIDELINES FOR THE LIEBERT XD FAMILY..10

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SUMMARY

As computer equipment gets smaller and more powerful, unprecedented heat densities are being created — densities traditional approaches to cooling cannot accommodate.

As computer manufacturers pack more and more processing power into smaller packages, the challenge of data center cooling becomes more complex – and more critical.

New servers and communication switches generate as much as ten times the heat per square foot as systems manufactured just ten years ago.

As these new systems are installed alongside previous generation systems, they create hot zones within the data center that cannot be effectively managed using traditional approaches to cooling.

New strategies and technologies must be implemented in the data center to provide the levels of cooling high-density systems require to deliver the availability expected from them.

The Liebert XD™ family of extreme density cooling systems extends the capacity of existing cooling systems by introducing a new cooling technology to the data center and by bringing cooling closer to the source of heat. The XD family was specifically designed to address the challenge of high-density computing systems and represents the most efficient and practical solution to a problem that will impact a growing number of data centers in the coming years.

MORE POWER

MORE HEAT

The history of the computer industry is the story of the quest to pack more processing power into a smaller footprint. True to Moore's Law, these advances have occurred exponentially, with processing power doubling on average every 18 months.

This has been an overwhelmingly positive development in almost every respect. From the data center perspective, it has meant the floor space required to achieve a constant quantity of computing and storage capability has shrunk geometrically. Of course, this has not resulted in an actual decrease in data center floor space – the quantity of computing and storage capability has increased more than the size of systems has decreased.

The density of power consumed and the heat dissipated per square foot has actually increased significantly as the size of equipment has decreased.

What it has done is allow data center managers to continually do more with their systems without continually expanding the data center.

However, one misconception that has resulted from this trend is the idea that power consumption and heat generation are shrinking along with equipment size.

This has not proven to be the case. The density of power consumed and the heat dissipated per square foot has actually increased significantly as the size of equipment has decreased. In fact, technology advances have driven the power densities of electronics to unprecedented levels and this has pushed existing data center support systems to the high end of their practical capacity.

To illustrate this trend, consider the microprocessor. There are three primary drivers of microprocessor power and density:

1. *Number of transistors per processor*
   The switching on-and-off of transistors consumes power. The more transistors per processor, the greater the power consumption.

2. *Processor clock speed*
   Clock speed determines how fast transistors turn on-and-off. Faster clock speeds increase the frequency with which transistors turn on-and-off, resulting in higher power consumption.

3. *Track spacing*
   Spacing determines processor density, or the ability of manufacturers to add more transistors without increasing the physical size of the processor.

In the course of just two years, the popular Pentium microprocessor has seen a 319% increase in power consumption.

It will be no surprise to anyone that each of these drivers has changed significantly since the introduction of the microprocessor. Yet the amount of change that has occurred since 1975 is still startling: the number of transistors per processor has increased by a factor of 4,000; clock frequency has been driven up by a factor of 600; and spacing has decreased by a factor of 60.

The result is smaller, more powerful processors that consume an ever increasing amount of power.

Take the Pentium 4 as an example. With its approximately 55,000,000 transistors, a clock speed approaching 3GHz and a track spacing of 0.09 micron, it consumes about 83 Watts at full power. Its predecessor, released in 2000 with "only" 28,100,000 transistors and a 1GHz clock speed, had a maximum power consumption of 26 Watts.

In the course of just two years there has been a 319% increase in power consumption for this processor.

This trend is expected to continue into the foreseeable future. Intel has predicted the number of transistors on their processors "should pass 200 million by 2005, and reach well in excess of 1 billion by the end of the decade." On the track spacing front, Sandia Labs has

The International Technology Roadmap for Semiconductors predicts microprocessor maximum power will reach 170 Watts by 2005.

experienced success using Extreme Ultraviolet Lithography to achieve track spacing of 0.07 micron, creating the potential for greater increases in transistor density.

These developments have led The International Technology Roadmap for Semiconductors – a cooperative effort among semiconductor manufacturers and suppliers, government organizations and universities that provides an ongoing assessment of semiconductor technology requirements – to predict that microprocessor maximum power will reach 170 Watts by 2005.

And all of that power will inevitably be transformed into heat – heat that must be removed from the data center.

THE POINT OF NO RETURN

Increases in computer system power consumption – and the resulting increases in heat generation – are not new. They have been occurring for years and traditional approaches to cooling have consistently been able to scale to meet these increases. However, with their recent advances, manufacturers have achieved densities that test the limits of traditional approaches to cooling.

Computer systems being installed today have total power dissipation in excess of 10 kW per rack. That density appears to be the threshold at which current approaches to cooling reach their practical limits.

Of course, system manufacturers are expected to drive power densities even higher in the years ahead. According to The Uptime Institute, a research institute supported by companies seeking to achieve the highest levels of availability from their computer systems, power densities increased 300% from 1992 to 2002. The Institute's study found that, "in each subsequent year the annual change gets larger."

"As the projected trends occur over the next three to six years, air from under the floor by itself will not be sufficient to remove the heat being generated."

The Uptime Institute

In its White Paper *The Implications of Increased Heat Density for Technology Spaces and Data Centers,* The Uptime Institute states clearly where the biggest impact from this trend will be felt:

> While many existing technology spaces and data centers are likely to be able to provide sufficient electrical power, most will struggle or may not be able to provide sufficient air circulation and air cooling capacity if large numbers of future high-performance IT products are installed. As the projected trends occur over the next three to six years, air from under the floor by itself will not be sufficient to remove the heat being generated.

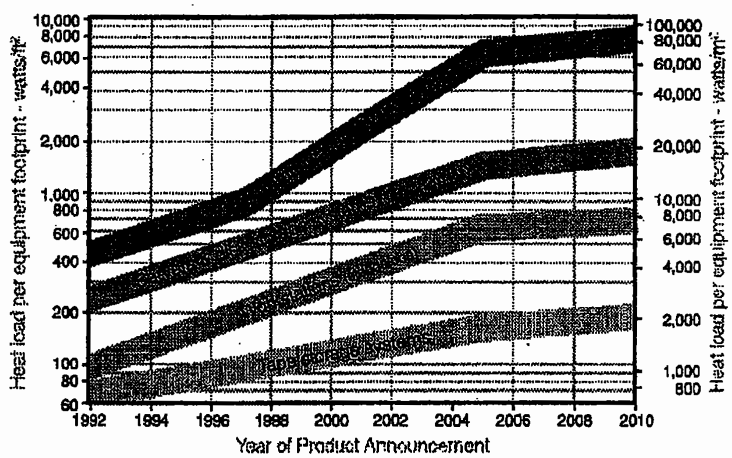


Figure 1. Projected Power Consumption. Heat density per square foot is continuing to rise for equipment typically found in today's data centers. (Source: Uptime Institute)

INCREASING

DATA CENTER COOLING

Data centers have traditionally been designed to handle densities up to 100 Watts per square foot using air distributed in a raised floor and exhausted upwards through perforated floor tiles.

With heat densities now exceeding 100 Watts per square foot, data center designers and managers are being forced to evaluate current cooling systems and identify ways to dissipate heat that meet the growing requirements of the new generation of high-density systems.

Solutions that have been considered to meet this challenge include increasing equipment spacing, adding more computer room air conditioners and making modifications to the floor that increase airflow. Each of these approaches offers some potential for incremental improvement. But when evaluated in light of the trends previously discussed, it becomes clear that none provides a practical, long-term solution to the growing heat density problem.

Increased Equipment Spacing
For the most part, focus on heat dissipation has been on the entire room, rather than any one zone. This being the case, it seems logical that the solution to more power per square foot is to increase the spacing between equipment. However, when you begin to calculate spacing using projected heat densities, the limitations of this solution become obvious.

Hardware manufacturers are on the verge of rolling out products that will consume over 16 kW of power per rack. Based on field measurements that have been conducted at various sites, the actual airflow in raised floor sites averages 250 cubic feet per minute or less through each perforated floor tile, creating the ability to dissipate about 2 kW of heat. Spacing the equipment in a way that will allow existing airflow to dissipate 16 kW would require aisle widths of 16 feet – hardly a practical solution considering the cost of data center space today.

Increase the Number of CRAC Units

Perhaps the most obvious solution to the problem posed by increased heat densities is to increase the number of computer room air conditioning units (CRACs) to accommodate the increased heat. This is often the first approach implemented, but at some point data centers will encounter one of three situations which limit their ability to add additional units:

- Space in the controlled environment is often scarce and expensive. Many data centers have simply run out of space for additional systems.
- As new high-density equipment is installed next to older equipment, hot spots occur where the new equipment is generating more heat than the equipment around it. The airflow will not be optimized for the high-density equipment.
- The mix of new and old equipment will create large variations in temperature across the space. Optimizing air conditioners to handle the hot spots results in all other equipment being cooled less efficiently, increasing costs.

Increasing Airflow

Inadequate air flow for raised floor air distribution systems has become one of the limiting factors for room cooling. A typical tile can provide a few hundred cubic feet of cold air for a rack that may need five to ten times as much cold air. New higher capacity floor tiles are being introduced, but these require careful balancing and still may not be able to scale to meet the demands of systems that will be introduced in three to five years. Plus, balancing the flows under the floor is nearly impossible with the high velocities that result in large rooms. In some cases, air is actually drawn into the tile due to Bernoulli effects. This significantly impacts the temperature distribution in a room.

Doubling the floor height has been shown to increase capacity by as much as 50%; however, it is a very disruptive solution that requires the removal of all existing equipment while the new floor is installed.

CREATING

NEW COOLING SOLUTIONS

With existing technologies unable to meet the expected increases in power densities, new approaches to cooling must be developed and implemented. These new approaches must have the following characteristics to meet the changing cooling requirements of data centers implementing high-density equipment:

- *Focused cooling*
  New solutions must have the ability to address specific zones within the data center, providing targeted cooling for high density systems as data centers experience increased heat diversity across the room as new and old systems share the space.

- *Scalability*
  Solutions implemented within the next five years must have the ability to scale significantly as heat densities of new systems continue to increase and more high density systems are installed.

- *Flexibility*
  No single solution will work for every data center. New approaches to cooling must have the flexibility to adapt to existing physical conditions within the data center. Rarely does a one-size-fits-all approach work in data center environments.

- *Minimal footprint*
  Significant increases in cooling cannot be achieved at the expense of large pieces of data center real estate.

- *Efficiency*
  The energy efficiency of cooling systems is increasingly being scrutinized. To be effective, new approaches to cooling must result in an increase in overall cooling efficiency.

- *Low Risk*
  New technologies must be able to handle high density cooling in a way that does not increase the perceived risk to data center equipment. Specifically, water-free approaches to cooling eliminate the risk associated with leaks in the data center.

THE LIEBERT XD™ FAMILY

Liebert has responded to the need for supplemental cooling within the data center with the Liebert XD family, a new generation cooling system that revolutionizes data center cooling. The Liebert XD family is a flexible, scalable and waterless solution to supplemental cooling that delivers sensible cooling of heat densities up to 500 Watts per square foot.

The Liebert XD family uses an environmentally friendly coolant to achieve high efficiencies and waterless cooling. The XD coolant is pumped as a liquid, converts to a gas within the heat exchangers, and then is returned to the pumping station where it is re-condensed to liquid. This is not a compression cycle so the concerns of refrigerant oil management have been eliminated. Also, because the Liebert coolant system takes advantage of the heat absorption capacity of a fluid changing state, the system is extremely efficient when compared to water-cooled systems.

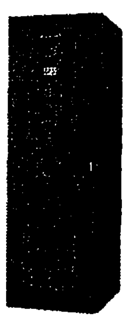


Figure 2. Liebert XDA Air Flow Enhancer.

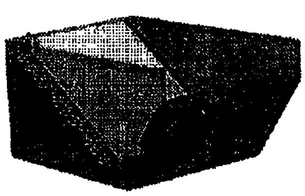


Figure 3. Liebert XDV Vertical Rack-Mounted Fan Coil

This has the additional benefit of not introducing any liquid into the controlled space. If a leak were to occur, the fluid would escape as a gas, causing no damage to the equipment within the controlled space. The Liebert XD family consists of the following products. Plus, the cooling capacity of the Liebert XD system is regulated to prevent any condensation from forming on the coil.

Liebert XD systems take advantage of the "hot aisle/cold aisle" practice in which cold aisles have perforated floor tiles that allow cooling air to come through the plenum under the floor and the hot aisle does not. Equipment racks are arranged face-to-face so the cooling air being pushed into the cold aisle is sucked into the face of the rack and exhausted out the back of the rack onto the adjacent hot aisle. This enables cooling systems to operate closer to their capacity.

Liebert XDA Air Flow Enhancer
A lightweight, rear-mounted fan for high-density racks. The Liebert XDA is designed to increase the air flow through densely populated enclosures, removing hot spots that threaten system availability. The Liebert VDA can move 2000 CFM of air through the enclosure.

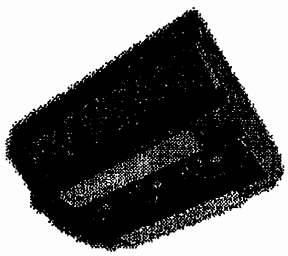


Figure 4. Liebert XDO Overhead Fan Coil

Liebert XDV Vertical Rack-Mounted Fan Coil
A vertical fan coil that mounts to the top of the rack. The Liebert XDV draws hot air directly out of the rack or from the hot aisle and exhausts cold air into the cold aisle where equipment air intakes are located.

Liebert XDO Overhead Fan Coil
An overhead, ceiling-mounted fan coil. The Liebert XDO delivers efficient cooling by drawing hot air into side-mounted coils and exhausting cool air vertically down into the cold aisle.

Liebert XDP Pumping Unit
Circulates and controls the XD coolant to Liebert XDO or XDV units in Indirect system configurations. The Liebert XDP serves as the intermediary between the building chilled water loop and the XD coolant loop.

The Liebert XDC Chiller
An indoor chiller that connects directly to the XDO or XDV units and provides coolant circulation and control. The XDC eliminates the need for a pumping unit in the Direct system configuration (see Application Guidelines for the Liebert XD family).

APPLICATION GUIDELINES FOR THE LIEBERT XD™ FAMILY

The Liebert XD family is designed to provide maximum flexibility and energy efficiency for data center designers and managers seeking to address hot zones in existing data centers or designing new spaces for high density systems.

The Liebert XD family delivers improved efficiency as compared to traditional solutions. This efficiency is achieved through the following:

- The Liebert XD system takes advantage of the energy absorption capacity of a fluid as it changes state. This reduces the pumping power by 85% compared to a water-based system.

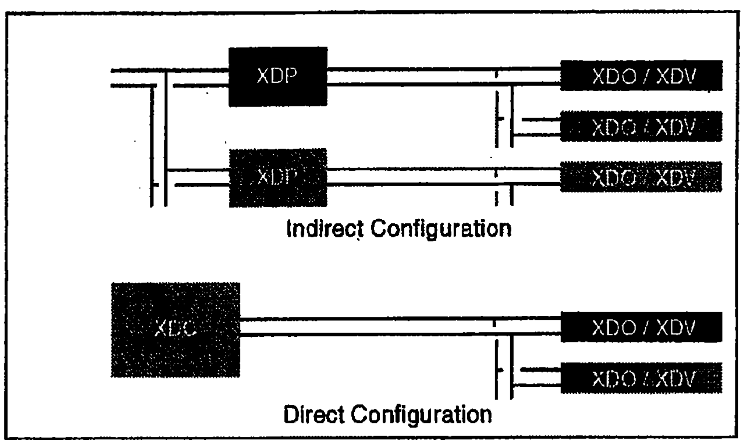
- Placing the cooling near the heat source shortens the distance which air must be moved (only 3 feet on a Liebert XDV) and the static pressure is negligible. This reduces the energy required for air movement by more than 70%.

- The chiller has an optional fluid economizer that allows for savings of over 25%, even in warmer climates such as Washington DC. Colder climates will realize greater savings.

The Liebert XDO and XDV units can be deployed in either Direct or Indirect system configurations. The main difference between the two is the location of the pumping unit (see Figure 5).

The Indirect system consists of a Liebert XDP pumping unit supporting several XDO or XDV units. The XDP works as an interface between building chilled water systems and the XD coolant system. It circulates and controls the coolant being used by either the XDV or XDO system, ensuring the coolant is always above the dewpoint in the room. This eliminates the possibility of condensation forming, potentially damaging equipment or creating an electrical hazard. This configuration provides added flexibility for existing data centers with physical space or other practical constraints that make the Direct configuration unfeasible. The building chiller must be suitable for precision cooling applications, which require year-round operation.

Figure 5. Direct and Indirect System Configurations. The Indirect XD system configuration uses the XD pumping unit to control and circulate the XD coolant. In the Direct configuration all functionality of the pumping unit is incorporated in the chiller.



The Direct configuration uses a Liebert XDC chiller in place of the XDP unit. All the functionality of the XDP is incorporated into the chiller. The XDC Chiller utilizes sophisticated technologies to transfer the heat from air- water- or glycol-cooled condensers and can be provided with liquid based economizers.

In each case, the Liebert XD family provides supplemental, sensible cooling only. The XD family is not designed to serve as the primary environmental control system within a controlled space. Floor-mounted Liebert units are used to handle the latent heat loads, filtration and the base air flow balance within the room.

Rack-Mounted Cooling

The XD family offers two solutions for rack-mounted cooling: the Liebert XDA and the Liebert XDV. The XDA is a rear-mounted air flow enhancer, while the XDV is a top-mounted system that uses the XD coolant to remove and cool hot air from the rack. The XDA has a capacity of 1000 CFM; the nominal capacity on the Liebert XDV unit is 8 kW.

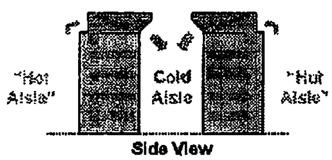


Figure 6. Liebert XDV Operation. The XDV mounts directly to the rack to provide zero footprint supplemental cooling.

The Liebert XDV is a fan coil that is mounted on top of the rack. It draws the hot air directly from the rack or from the hot aisles in the room. Inside the XDV the air is cooled as it passes through the coil and cold air is exhausted into the cold aisle where the computer air intakes are located.

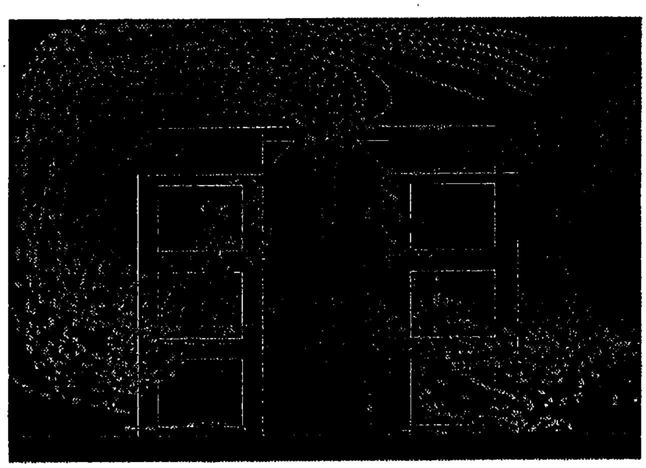


Figure 7. Liebert XDV Air flow. Computational fluid dynamic model showing air vectors and their temperatures for a data center using the Liebert XDV system. The raised floor cooling system cools the lower half of the room, while the XDV supplements this system by cooling the upper areas.

The Liebert XDA addresses one of the key challenges involved in cooling racks: overcoming the air restrictions, which are created by congested cabling. The fans in the servers inside the rack produce insufficient flow to push air through the masses of communication and power cabling at the rear of the rack

The Liebert XDA is designed as a rear-mounted system that is placed on the exhaust side of the rack. It moves heat out of the enclosure, across the entire height of the enclosure, where it can be removed by the room air conditioning unit. One or two units can be placed on the inside or outside of most racks. The Liebert XDA does not take up any server space within the rack and does not rely on airflow through the floor for cooling so it can be used in a wide range of environments.

Ceiling Mounted Cooling

The Liebert XDO is an overhead fan coil that is mounted above the cold aisle. It takes the hot air from the equipment in the room through the coils on the sides of the unit. The air is cooled in the coils and exhausted vertically down into the cold aisle where the equipment intake is located. Each Liebert XDO unit is capable of 32 kW of cooling and 8000 CFM.

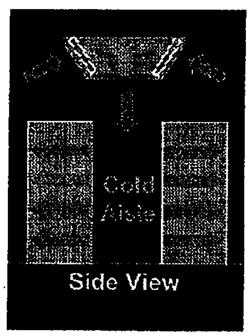


Figure 8. Liebert XDO Operation.
The XDO mounts above the cold aisle and pulls hot air from high density systems and exhausts cold air into the cold aisle.

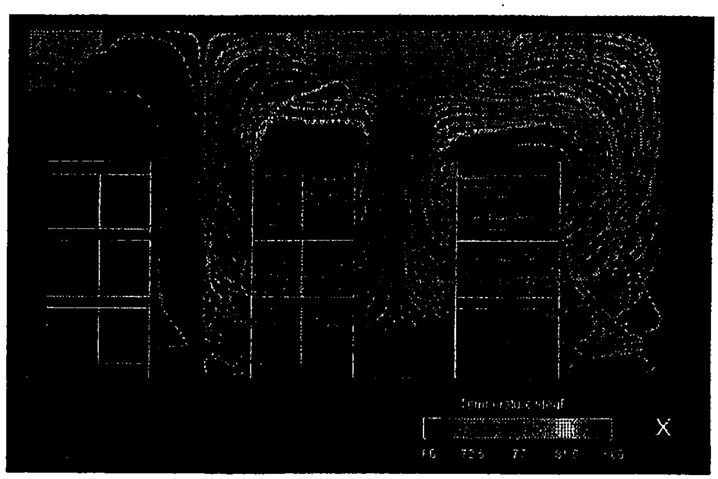


Figure 9. Liebert XDO Air Flow. A computational fluid dynamics model illustrating how the Liebert XDO system supplements the existing cooling system by eliminating hot spots in the upper parts of the room.

CONCLUSIONS

Power densities have exceeded the capacities of traditional approaches to cooling and are going even higher. Over the next five years, data centers installing new systems will be faced with two challenges:

- Managing hot spots within the data center caused by new systems that have much higher densities than the systems around them.
- Managing increasing overall temperatures as high-density systems displace older systems, creating higher heat across the room.

Forcing increased cooling through the raised floor is simply not a practical solution to this problem due to the limited capacity of floor tiles, physical space limitations within the data center for additional computer room air conditioners and the inefficiency of this approach.

The Liebert XD family is the first complete solution to extreme density cooling, providing a scalable, flexible solution that can be optimized for new and existing data centers. The Liebert XD family supplements the existing cooling system with 100% sensible cooling designed specifically to meet the needs of high-density systems. The Liebert XD family's range of solutions enables data center managers to address specific hot spots within the data center and increase cooling capacity as new systems are installed.

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Doppelt gemoppelt kühlt besser

Kombination spart Strom bei der Schaltschrank-Klimatisierung

p.138+140 = (2) p.d. 05-11-1998

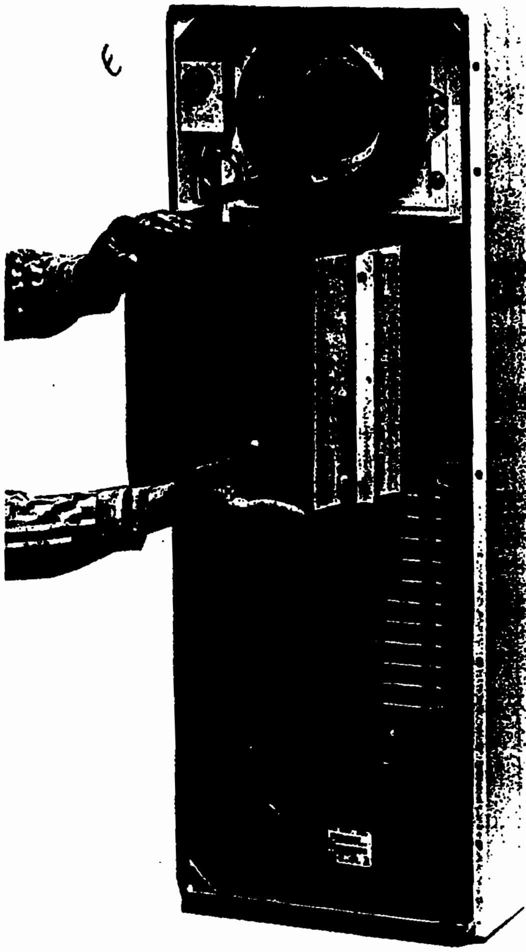

Bild 1: Wird's zu heiß, unterstützen sich Luft-/Luft-Wärmeaustauscher und das Luft-/Kältemittelsystem.

Bilder: Bader Engineering

Effektive Klimatisierung ist heute nicht mehr das alleinige Kriterium für die Auswahl eines Schaltschrankkühlgerätes. Umweltfreundliche und energiesparende Lösungen liegen im Trend.

G. Klingberg*

Zur Vermeidung von Schäden bei elektronischen Komponenten im Schaltschrank muß dafür gesorgt werden, daß der Innenraum des Schrankes eine definierte Höchsttemperatur nicht überschreitet. Darüber hinaus ist es häufig erforderlich, den Innenraum auf einem gleichmäßigen Temperaturniveau zu halten, ohne daß eine Verschmutzung der Elektrik durch die Umgebungsbedingungen auftritt. Zum Kühlen von Schaltschränken sind einerseits sogenannte passive oder indirekte Kühlgeräte mit einem Wärmeaustauscher im Einsatz, die nach dem Luft-/Luft-Prinzip funktionieren. Bei diesen Geräten wird die erwärmte Luft aus dem Innenraum des Schaltschrankes mittels eines Ventilators abgesaugt, durch einen Wärmeaustauscher befördert und schließlich kälter zurückgeführt. Solche Kühlgeräte arbeiten zufriedenstellend, wenn die Umgebungstemperatur und die Anforderungen an eine gleichmäßige Temperatur im Innenraum des Schrankes nicht allzu hoch sind.

Klimatisieren auch bei höheren Temperaturen

Das Problem der Schaltschrankklimatisierung wird gravierender, wenn die Umgebungstemperatur höher ist als die erforderliche Innentemperatur. In Industrieanlagen, in denen die Umgebungstemperatur ständig hoch ist, kann nicht mit dem bisher beschriebenen Kühlgerät gearbeitet werden. In vielen Fällen werden für die gestiegenen Anforderungen Luft-/Kältemittel-Kühlgeräte notwendig. Diese arbeiten mit einem Kältemittel (R 134 a) nach dem bekannten Carnot-Kreisprozeß: Dabei wird gasförmiges Kältemittel von einem Verdichter angesaugt, komprimiert und in einem Luft-/Kältemittel-Wärmeaustauscher verflüssigt. Es durchläuft einen Trockner/Sammler und das Expansionsventil. Schließlich wird die nunmehr sehr niedrige Temperatur des Kältemittels in einem zweiten Luft-/Kältemittel-Wärmeaustauscher dazu genutzt, dem Schaltschrank-Innenraum Wärme zu entziehen. Mit diesen Kühlgeräten lassen sich Schranktemperaturen erreichen, die unter den Umgebungstemperaturen liegen. Allerdings verbrauchen die Geräte etwa 10 mal mehr Strom als Luft-/Luft-Kühlgeräte.

Dies veranlaßte den Sendener Schaltschrankkühlgeräte-Hersteller Bader Engineering ein Kombigerät zu entwickeln, das umweltschonend, energiesparend und kostengünstig arbeitet. Das zum Patent angemeldete Kombinationsgerät verbindet die Vorzüge der beiden Kühlsysteme. Es ist bei verhältnismäßig niedrigen Betriebskosten für hohe Temperaturanforderungen genauso geeignet wie für wechselnde Umgebungstemperaturen.

Kombigeräte senken den Stromverbrauch

Dabei wird der passive, stromsparendere Teil der Kombination (fast) ohne Unterbrechung benutzt, während der aktive Teil des Kühlsystems nur bei Bedarf zugeschaltet wird, beispielsweise in der Mittagszeit oder im Sommer. Der Vorteil liegt unter anderem in der Ausnutzung des in

*Gottfried Klingberg ist Inhaber eines Beratungsbüros für Schaltschrank- und Gehäuseklimatisierung in Überherrn.

jedem Falle notwendigen Wärmeaustauschers durch die beiden Teile des Gesamtsystems. In jenen Phasen, in denen beide Systeme arbeiten, wird die Arbeit des kältemittelverflüssigenden Wärmeaustauschers bei relativ niedrigen Umgebungstemperaturen durch den vorgeschalteten Luft-/Luft-Wärmeaustauscher begünstigt, der ständig betrieben wird.

Bei dem Kombigerät werden beide Arten der beschriebenen Wärmeübertragung zur Schaltschrankkühlung genutzt. Radialventilatoren übernehmen sowohl für die Luft-/Luft als auch für den Kältemittelbetrieb den Transport der Luft zu den Wärmeaustauschern. Das Aluminium- Wärmeaustauscherpaket für die Luft-/Luft-Übertragung ist konstruktiv so angeordnet, daß die beiden Luftkreisläufe immer zuerst diesen Wärmeaustauscher durchströmen. Sind optimale Voraussetzungen vorhanden, reicht die Kühlleistung der Luft-/Luft- Wärmeübertragung aus, die produzierte Wärme aus dem Schaltschrank abzuführen. Steigende Umgebungstemperaturen vermindern die Leistung des Luft-/Luft-Austauschs und führen dazu, daß die Kühlleistung allein durch diese Komponente nicht mehr ausreicht.

Luftkühlung unterstützt den Kältemittelbetrieb

Jetzt wird der Kältemittelkreislauf aktiviert und erhöht die geforderte Nutzkühlleistung. Der Kältemittelkreis bleibt solange aktiv, bis eine bestimmte Solltemperatur im Schaltschrank erreicht wird. Sind die Umgebungstemperaturen so hoch, daß die Kühlleistung der Luft-/Luft-Wärmeübertragung gegen Null geht, übernimmt der Kältekreislauf die gesamte Nutzkühlleistung. Übersteigen die Umgebungstemperaturen die Solltemperatur im Schaltschrank, muß der Kältemittelkreislauf bei der Kompaktbauweise des Kombinationsgerätes zusätzlich die entgegenwirkende Leistung des Luft-/Luft-Wärmeaustauschers kompensieren.

Größere Ausführungen des Kombigerätes mit einer 3 bzw. 4-Ventilatorkonstruktion können bei diesen Temperaturbedingungen vollständig, auch mit der Luftführung, auf den Kältemittelbetrieb umschalten.

Bis zu 50 % weniger Energie in der Halle

Um die Energieeinsparung realistisch beziffern zu können, wurden Feldversuche in unterschiedlichen Einsatzgebieten durchgeführt, die gezeigt haben, daß sich durch die Anwendung des Kombinations- Kompaktgerätes gegenüber einem vergleichbaren, konventionellen Kältemittel- Kühlgerätes eine Energieeinsparung von über 50 % erzielen läßt, wenn sich der zu kühlende Schaltschrank im Indoor-Bereich befindet. Ist die Aufstellung im Outdoor-Bereich, scheinen sogar Energieeinsparungen von bis zu 80 % erreichbar.

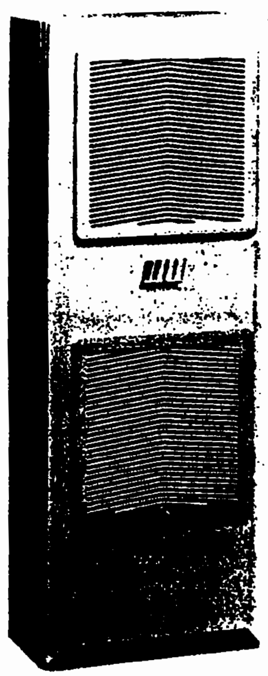

Bild 2: Das Schaltschrank-Kombinationskühlgerät ermöglicht Energieeinsparungen zwischen 50 und 80 % gegenüber herkömmlichen Varianten.

Beim Einsatz von Kompressor-Kühlgeräten ist darauf zu achten, daß die Umgebungstemperatur nicht zu niedrig ist, da sonst der Kältekreislauf zusammenbricht. Der Verflüssiger von Kältemittelkühlgeräten wird konstruktiv bedingt immer der Umgebungstemperatur ausgesetzt. Fällt diese unter 0 °C sammelt sich der größte Teil des Kältemittels im Verflüssiger und ist dem Kühlkreislauf entzogen. Um bei niedrigen Umgebungstemperaturen eine Nutzkühlleistung zu gewinnen, muß die Verflüssigungstemperatur durch eine spezielle Technik auf einer bestimmten Mindesttemperatur gehalten werden, da ein Vereisen des Verdampfers die Leistung deutlich mindert. Diese Tatsachen begrenzen den Einsatz von herkömmlichen Schaltschrank-Kältemittelkühlgeräten im Bereich niedriger Umgebungstemperaturen. Hingegen wird beim Kombigerät durch das Vorschalten des Luft-/Luft-Wärmeaustauschers die Umgebungsluft erwärmt und erst dann dem Verflüssiger zugeführt. Dadurch und durch eine Heißgasbeimischung mit einem Bypassventil ist die Kombination auch bei sehr tiefen Umgebungstemperaturen einsetzbar.

Ein weiteres Merkmal des Kombigerätes ist die Tatsache, daß bei Außenaufstellung (z.B. Mobilfunkstationen) eine Notlauf-Nutzkühlleistung beim Ausfall des Verdichters durch den Luft-/Luft-Wärmeaustauschers gegeben ist. Desweiteren können die zur Notkühlung relevanten Ventilatoren so ausgelegt werden, daß eine Batteriespannung von 48 Volt für den Betrieb bei Netzausfall ausreicht. Die 4-Ventilatorkonstruktion des Kombigerätes ist auch in der Lage einen Ventilatorausfall zu kompensieren.

Weitere Informationen zu diesem kombinierten Schaltschrankkühlgerät vermittelt der Leserdienst unter Kennziffer 429

PATENT COOPERATION TREATY

PCT

INTERNATIONAL SEARCH REPORT

(PCT Article 18 and Rules 43 and 44)

Applicant's or agent's file reference 021944-085WO	**FOR FURTHER ACTION** see Form PCT/ISA/220 as well as, where applicable, Item 5 below.	
International application No. PCT/US2005/040745	International filing date *(day/month/year)* 10/11/2005	(Earliest) Priority Date *(day/month/year)* 14/11/2004
Applicant LIEBERT CORPORATION		

This international search report has been prepared by this International Searching Authority and is transmitted to the applicant according to Article 18. A copy is being transmitted to the International Bureau.

This international search report consists of a total of 4 sheets.

[X] It is also accompanied by a copy of each prior art document cited in this report.

1. **Basis of the report**
   a. With regard to the **language**, the international search was carried out on the basis of:
      [X] the international application in the language in which it was filed
      [ ] a translation of the international application into ________, which is the language of a translation furnished for the purposes of international search (Rules 12.3(a) and 23.1(b))
   b. [ ] With regard to any **nucleotide and/or amino acid sequence** disclosed in the international application, see Box No. I.
2. [ ] **Certain claims were found unsearchable** (See Box No. II)
3. [ ] **Unity of invention is lacking** (see Box No III)
4. With regard to the **title**,
   [X] the text is approved as submitted by the applicant
   [ ] the text has been established by this Authority to read as follows:
5. With regard to the **abstract**,
   [ ] the text is approved as submitted by the applicant
   [X] the text has been established, according to Rule 38.2(b), by this Authority as it appears in Box No. IV. The applicant may, within one month from the date of mailing of this international search report, submit comments to this Authority
6. With regard to the **drawings**,
   a. the figure of the **drawings** to be published with the abstract is Figure No. 1
      [X] as suggested by the applicant
      [ ] as selected by this Authority, because the applicant failed to suggest a figure
      [ ] as selected by this Authority, because this figure better characterizes the invention
   b. [ ] none of the figures is to be published with the abstract

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Date Due/Done: ___
By: KA. Checked ___

Form PCT/ISA/210 (first sheet) (April 2005)

Box No. IV Text of the abstract (Continuation of item 5 of the first sheet)

A system for cooling heat-generating objects (12), such as computer boards situated in a rack, includes an enclosure (16) in which the heat generating objects are situated. The enclosure has an air inlet and an air outlet, an a fan (14) induces airflow into the air inlet, through the enclosure and out the air outlet. A heat exchanger (20) is situated in the enclosure such that the heat exchanger is in a spaced apart relationship with the heat-generating object. Air moving through or past the heat-generating object is warmed, and the heat exchanger removes the heat before the air exits the enclosure.

A. CLASSIFICATION OF SUBJECT MATTER
H05K7/20

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

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
H05K

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

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

EPO-Internal

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	US 2001/042616 A1 (BAER DANIEL B) 22 November 2001 (2001-11-22) paragraph [0002] paragraph [0021] - paragraph [0044] claim 16 figures 1-8	1,12,13
A	US 2002/039280 A1 (O'CONNOR STEPHEN D ET AL) 4 April 2002 (2002-04-04) column 1, line 16 - line 22 column 3, line 29 - column 5, line 34 figures 1-4	2-6, 14-18
A	US 6 557 624 B1 (STAHL LENNART ET AL) 6 May 2003 (2003-05-06) paragraph [0002] paragraph [0045] - paragraph [0065]; figures 1-5	9

☐ Further documents are listed in the continuation of Box C.

☒ See patent family annex.

* Special categories of cited documents :

"A" document defining the general state of the art which is not considered to be of particular relevance

"E" earlier document but published on or after the international filing date

"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)

"O" document referring to an oral disclosure, use, exhibition or other means

"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone

"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art.

"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
22 March 2006	30/03/2006

Name and mailing address of the ISA/	Authorized officer
European Patent Office, P.B. 5818 Patentlaan 2 NL – 2280 HV Rijswijk Tel. (+31-70) 340-2040, Tx. 31 651 epo nl, Fax: (+31-70) 340-3016	Miot, F

1

Patent document cited in search report		Publication date	Patent family member(s)		Publication date
US 2001042616	A1	22-11-2001	NONE		
US 2002039280	A1	04-04-2002	NONE		
US 6557624	B1	06-05-2003	AU	8314301 A	18-02-2002
			WO	0212797 A2	14-02-2002

From the
INTERNATIONAL SEARCHING AUTHORITY

To: see form PCT/ISA/220	# PCT WRITTEN OPINION OF THE INTERNATIONAL SEARCHING AUTHORITY (PCT Rule 43*bis*.1)
	Date of mailing *(day/month/year)* see form PCT/ISA/210 (second sheet)
Applicant's or agent's file reference see form PCT/ISA/220	**FOR FURTHER ACTION** See paragraph 2 below

International application No.	International filing date *(day/month/year)*	Priority date *(day/month/year)*
PCT/US2005/040745	10.11.2005	14.11.2004

International Patent Classification (IPC) or both national classification and IPC
INV. H05K7/20

Applicant
LIEBERT CORPORATION

1. This opinion contains indications relating to the following items:

- ☒ Box No. I Basis of the opinion
- ☐ Box No. II Priority
- ☒ Box No. III Non-establishment of opinion with regard to novelty, inventive step and industrial applicability
- ☐ Box No. IV Lack of unity of invention
- ☒ Box No. V Reasoned statement under Rule 43*bis*.1(a)(i) with regard to novelty, inventive step or industrial applicability; citations and explanations supporting such statement
- ☐ Box No. VI Certain documents cited
- ☐ Box No. VII Certain defects in the international application
- ☐ Box No. VIII Certain observations on the international application

2. **FURTHER ACTION**

If a demand for international preliminary examination is made, this opinion will usually be considered to be a written opinion of the International Preliminary Examining Authority ("IPEA"). However, this does not apply where the applicant chooses an Authority other than this one to be the IPEA and the chosen IPEA has notifed the International Bureau under Rule 66.1*bis*(b) that written opinions of this International Searching Authority will not be so considered.

If this opinion is, as provided above, considered to be a written opinion of the IPEA, the applicant is invited to submit to the IPEA a written reply together, where appropriate, with amendments, before the expiration of three months from the date of mailing of Form PCT/ISA/220 or before the expiration of 22 months from the priority date, whichever expires later.

For further options, see Form PCT/ISA/220.

3. For further details, see notes to Form PCT/ISA/220.

DOCKETED ☑ UPDATED ☐
Previously___ Not Required___
Appl. Info ______
Reg/Grant Info ______
Action Required: Written Opinion rcvd
Date Due/Done: ______
By: R.A. Checked ______

Name and mailing address of the ISA:	Authorized Officer
European Patent Office D-80298 Munich Tel. +49 89 2399 - 0 Tx: 523656 epmu d Fax: +49 89 2399 - 4465	Miot, F Telephone No. +49 89 2399-2714

Form (PCT/ISA/237) (Cover Sheet) (January 2004)

WRITTEN OPINION OF THE INTERNATIONAL SEARCHING AUTHORITY

International application No.
PCT/US2005/040745

Box No. I Basis of the opinion

1. With regard to the **language**, this opinion has been established on the basis of the international application in the language in which it was filed, unless otherwise indicated under this item.

   ☐ This opinion has been established on the basis of a translation from the original language into the following language , which is the language of a translation furnished for the purposes of international search (under Rules 12.3 and 23.1(b)).

2. With regard to any **nucleotide and/or amino acid sequence** disclosed in the international application and necessary to the claimed invention, this opinion has been established on the basis of:

   a. type of material:

      ☐ a sequence listing

      ☐ table(s) related to the sequence listing

   b. format of material:

      ☐ in written format

      ☐ in computer readable form

   c. time of filing/furnishing:

      ☐ contained in the international application as filed.

      ☐ filed together with the international application in computer readable form.

      ☐ furnished subsequently to this Authority for the purposes of search.

3. ☐ In addition, in the case that more than one version or copy of a sequence listing and/or table relating thereto has been filed or furnished, the required statements that the information in the subsequent or additional copies is identical to that in the application as filed or does not go beyond the application as filed, as appropriate, were furnished.

4. Additional comments:

Box No. III Non-establishment of opinion with regard to novelty, inventive step and industrial applicability

The questions whether the claimed invention appears to be novel, to involve an inventive step (to be non obvious), or to be industrially applicable have not been examined in respect of:

☐ the entire international application,

☒ claims Nos. 1-12

because:

☐ the said international application, or the said claims Nos. relate to the following subject matter which does not require an international preliminary examination *(specify)*:

☒ the description, claims or drawings *(indicate particular elements below)* or said claims Nos. claim 1 and claim 12 are so unclear that no meaningful opinion could be formed *(specify)*:

see separate sheet

☐ the claims, or said claims Nos. are so inadequately supported by the description that no meaningful opinion could be formed.

☐ no international search report has been established for the whole application or for said claims Nos.

☐ the nucleotide and/or amino acid sequence listing does not comply with the standard provided for in Annex C of the Administrative Instructions in that:

the written form	☐	has not been furnished
	☐	does not comply with the standard
the computer readable form	☐	has not been furnished
	☐	does not comply with the standard

☐ the tables related to the nucleotide and/or amino acid sequence listing, if in computer readable form only, do not comply with the technical requirements provided for in Annex C-*bis* of the Administrative Instructions.

☐ See separate sheet for further details

Box No. V Reasoned statement under Rule 43*bis*.1(a)(i) with regard to novelty, inventive step or industrial applicability; citations and explanations supporting such statement

1. Statement

Novelty (N)	Yes:	Claims	
	No:	Claims	13
Inventive step (IS)	Yes:	Claims	
	No:	Claims	13-18
Industrial applicability (IA)	Yes:	Claims	13-18
	No:	Claims	

2. Citations and explanations

see separate sheet

Re Item III.

1.1 Although claims 1 and 12 have been drafted as separate independent claims, they appear to relate effectively to the same subject-matter and to differ from each other only with regard to the definition of the subject-matter for which protection is sought and/or in respect of the terminology used for the features of that subject-matter. The aforementioned claims therefore lack conciseness and as such do not meet the requirements of Article 6 PCT.

1.2 Since claims 2-11 are dependent on claim 1, also these claims do not meet the requirements of Article 6 PCT.

Re Item V.

1 Reference is made to the following documents:
D1 : US 2001/042616 A1 (BAER DANIEL B) 22 November 2001 (2001-11-22)

INDEPENDENT CLAIM 13

1.1 The present application does not meet the criteria of Article 33(1) PCT, because the subject-matter of claim 13 is not new in the sense of Article 33(2) PCT.
Document D1 discloses (the references applying to this document):

a method for cooling heat generating objects 240 situated in an enclosure 210 (see claim 16, paragraph 43 and fig. 8), the method comprising:
situating a heat exchanger 250 in the enclosure 210 such that the heat exchanger is spaced apart from the heat generating object 240 (see paragraph 43; fig. 8 and claim 16); and
inducing air flow into the air inlet, through the enclosure and out the air outlet such that the heat exchanger 250 removes heat from the air flowing through the enclosure

(see paragraph 43; fig. 8 and claim 16).

1.2 DEPENDENT CLAIMS 14-18

Dependent claims 14-18. do not appear to contain any additional features which involve an inventive step when combined with the subject matter of any claim to which it refers.

PATENT COOPERATION TREATY

JUN 18 2007

DOCKETING DEPT.

From the INTERNATIONAL SEARCHING AUTHORITY

To:

LOCKE LIDDELL & SAPP LLP
Attn. Nelson, Brit D.
600 Travis Street, Suite 3400
Houston, TX 77002-3095
ETATS-UNIS D'AMERIQUE

PCT

NOTIFICATION OF TRANSMITTAL OF THE INTERNATIONAL SEARCH REPORT AND THE WRITTEN OPINION OF THE INTERNATIONAL SEARCHING AUTHORITY, OR THE DECLARATION

(PCT Rule 44.1)

	Date of mailing *(day/month/year)* 18/06/2007
Applicant's or agent's file reference 021944-093WO	**FOR FURTHER ACTION** See paragraphs 1 and 4 below
International application No. PCT/US2006/028088	International filing date *(day/month/year)* 20/07/2006
Applicant LIEBERT CORPORATION	

1. [x] The applicant is hereby notified that the international search report and the written opinion of the International Searching Authority have been established and are transmitted herewith.

Filing of amendments and statement under Article 19:
The applicant is entitled, if he so wishes, to amend the claims of the International Application (see Rule 46):

When? The time limit for filing such amendments is normally two months from the date of transmittal of the International Search Report.

Where? Directly to the International Bureau of WIPO, 34 chemin des Colombettes
1211 Geneva 20, Switzerland, Fascimile No.: (41-22) 338.82.70

For more detailed instructions, see the notes on the accompanying sheet.

2. [ ] The applicant is hereby notified that no international search report will be established and that the declaration under Article 17(2)(a) to that effect and the written opinion of the International Searching Authority are transmitted herewith.

3. [ ] **With regard to the protest** against payment of (an) additional fee(s) under Rule 40.2, the applicant is notified that:

[ ] the protest together with the decision thereon has been transmitted to the International Bureau together with the applicant's request to forward the texts of both the protest and the decision thereon to the designated Offices.

[ ] no decision has been made yet on the protest; the applicant will be notified as soon as a decision is made.

4. **Reminders**

Shortly after the expiration of **18 months** from the priority date, the international application will be published by the International Bureau. If the applicant wishes to avoid or postpone publication, a notice of withdrawal of the international application, or of the priority claim, must reach the International Bureau as provided in Rules 90*bis*.1 and 90*bis*.3, respectively, before the completion of the technical preparations for international publication.

The applicant may submit comments on an informal basis on the written opinion of the International Searching Authority to the International Bureau. The International Bureau will send a copy of such comments to all designated Offices unless an international preliminary examination report has been or is to be established. These comments would also be made available to the public but not before the expiration of 30 months from the priority date.

Within **19 months** from the priority date, but only in respect of some designated Offices, a demand for international preliminary examination must be filed if the applicant wishes to postpone the entry into the national phase **until 30 months** from the priority date (in some Offices even later); otherwise, the applicant must, **within 20 months** from the priority date, perform the prescribed acts for entry into the national phase before those designated Offices.

In respect of other designated Offices, the time limit of **30 months** (or later) will apply even if no demand is filed within 19 months.

See the Annex to Form PCT/IB/301 and, for details about the applicable time limits, Office by Office, see the *PCT Applicant's Guide*, Volume II, National Chapters and the WIPO Internet site.

DOCKETED [x] UPDATED [ ]
Previously ___ Not Required ___
Appl. Info ___
Reg/Grant Info ___
Action Required: PCT-Article 19 Amendment
~~Date Due~~ Done: 8-18-07
By: RA Checked ___

Name and mailing address of the International Searching Authority European Patent Office, P.B. 5818 Patentlaan 2 NL-2280 HV Rijswijk Tel. (+31-70) 340-2040, Tx. 31 651 epo nl, Fax: (+31-70) 340-3016	Authorized officer Iveta Bujanská

Form PCT/ISA/220 (October 2005)

(See notes on accompanying sheet)

NOTES TO FORM PCT/ISA/220

These Notes are intended to give the basic instructions concerning the filing of amendments under article 19. The Notes are based on the requirements of the Patent Cooperation Treaty, the Regulations and the Administrative Instructions under that Treaty. In case of discrepancy between these Notes and those requirements, the latter are applicable. For more detailed information, see also the *PCT Applicant's Guide*, a publication of WIPO.

In these Notes, "Article", "Rule", and "Section" refer to the provisions of the PCT, the PCT Regulations and the PCT Administrative Instructions, respectively.

INSTRUCTIONS CONCERNING AMENDMENTS UNDER ARTICLE 19

The applicant has, after having received the international search report and the written opinion of the International Searching Authority, one opportunity to amend the claims of the international application. It should however be emphasized that, since all parts of the international application (claims,description and drawings) may be amended during the international preliminary examination procedure, there is usually no need to file amendments of the claims under Article 19 except where, e.g. the applicant wants the latter to be published for the purposes of provisional protection or has another reason for amending the claims before international publication. Furthermore, it should be emphasized that provisional protection is available in some States only (see *PCT Applicant's Guide*, Volume I/A, Annexes B1 and B2).

The attention of the applicant is drawn to the fact that amendments to the claims under Article 19 are not allowed where the International Searching Authority has declared, under Article 17(2), that no international search report would be established (see *PCT Applicant's Guide*, Volume I/A, paragraph 296).

What parts of the international application may be amended?

Under Article 19, only the claims may be amended.

During the international phase, the claims may also be amended (or further amended) under Article 34 before the International Preliminary Examining Authority. The description and drawings may only be amended under Article 34 before the International Examining Authority.

Upon entry into the national phase, all parts of the international application may be amended under Article 28 or, where applicable, Article 41.

When? Within 2 months from the date of transmittal of the international search report or 16 months from the priority date, whichever time limit expires later. It should be noted, however, that the amendments will be considered as having been received on time if they are received by the International Bureau after the expiration of the applicable time limit but before the completion of the technical preparations for international publication (Rule 46.1).

Where not to file the amendments?

The amendments may only be filed with the International Bureau and not with the receiving Office or the International Searching Authority (Rule 46.2).

Where a demand for international preliminary examination has been/is filed, see below.

How? Either by cancelling one or more entire claims, by adding one or more new claims or by amending the text of one or more of the claims as filed.

A replacement sheet must be submitted for each sheet of the claims which, on account of an amendment or amendments, differs from the sheet originally filed.

All the claims appearing on a replacement sheet must be numbered in Arabic numerals. Where a claim is cancelled, no renumbering of the other claims is required. In all cases where claims are renumbered, they must be renumbered consecutively (Section 205(b)).

The amendments must be made in the language in which the international application is to be published.

What documents must/may accompany the amendments?

Letter (Section 205(b)):

The amendments must be submitted with a letter.

The letter will not be published with the international application and the amended claims. It should not be confused with the "Statement under Article 19(1)" (see below, under "Statement under Article 19(1)").

The letter must be in English or French, at the choice of the applicant. However, if the language of the international application is English, the letter must be in English; if the language of the international application is French, the letter must be in French.

PATENT COOPERATION TREATY

PCT

INTERNATIONAL SEARCH REPORT

(PCT Article 18 and Rules 43 and 44)

Applicant's or agent's file reference 021944-093WO	**FOR FURTHER ACTION** see Form PCT/ISA/220 as well as, where applicable, item 5 below.	
International application No. PCT/US2006/028088	International filing date *(day/month/year)* 20/07/2006	(Earliest) Priority Date *(day/month/year)* 04/08/2005
Applicant LIEBERT CORPORATION		

This international search report has been prepared by this International Searching Authority and is transmitted to the applicant according to Article 18. A copy is being transmitted to the International Bureau.

This international search report consists of a total of 3 sheets.

[X] It is also accompanied by a copy of each prior art document cited in this report.

1. **Basis of the report**
   a. With regard to the **language,** the international search was carried out on the basis of:
      [X] the international application in the language in which it was filed
      [ ] a translation of the international application into ____________, which is the language of a translation furnished for the purposes of international search (Rules 12.3(a) and 23.1(b))
   b. [ ] With regard to any **nucleotide and/or amino acid sequence** disclosed in the international application, see Box No. I.
2. [ ] **Certain claims were found unsearchable** (See Box No. II)
3. [ ] **Unity of invention is lacking** (see Box No III)
4. With regard to the **title,**
   [X] the text is approved as submitted by the applicant
   [ ] the text has been established by this Authority to read as follows:
5. With regard to the **abstract,**
   [X] the text is approved as submitted by the applicant
   [ ] the text has been established, according to Rule 38.2(b), by this Authority as it appears in Box No. IV. The applicant may, within one month from the date of mailing of this international search report, submit comments to this Authority
6. With regard to the **drawings,**
   a. the figure of the **drawings** to be published with the abstract is Figure No. 3
      [ ] as suggested by the applicant
      [X] as selected by this Authority, because the applicant failed to suggest a figure
      [ ] as selected by this Authority, because this figure better characterizes the invention
   b. [ ] none of the figures is to be published with the abstract

DOCKETED [X] UPDATED [ ]
Previously ___ Not Required ___
Appl. Info ___
Reg. Grant Info ___
Action Required: Search Report
Date Due/Done: ___
By: DV Checked

Form PCT/ISA/210 (first sheet) (April 2005)

PCT/US2006/028088

A. CLASSIFICATION OF SUBJECT MATTER
INV. H05K7/20

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

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
H05K

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

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

EPO-Internal

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	US 5 414 591 A (KIMURA HIDEYUKI [JP] ET AL) 9 May 1995 (1995-05-09) column 1, line 6 - line 13 column 4, line 8 - column 6, line 16 column 13, line 22 - column 15, line 53 figures 1-25 -----	1-3,19, 22
A	DE 299 08 370 U1 (BADER ENGINEERING GMBH [DE]) 30 September 1999 (1999-09-30) page 1, line 1 - line 9 page 4, line 3 - page 9, line 8 figures 1-5 -----	1,19
A	EP 1 357 778 A2 (MITSUBISHI ELECTRIC CORP [JP]) 29 October 2003 (2003-10-29) paragraph [0001] - paragraph [0004] paragraph [0026] - paragraph [0028] figures 1-32 -----	1,19

☐ Further documents are listed in the continuation of Box C.

☒ See patent family annex.

* Special categories of cited documents :

"A" document defining the general state of the art which is not considered to be of particular relevance

"E" earlier document but published on or after the international filing date

"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)

"O" document referring to an oral disclosure, use, exhibition or other means

"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone

"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art.

"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
11 June 2007	18/06/2007

Name and mailing address of the ISA/	Authorized officer
European Patent Office, P.B. 5818 Patentlaan 2 NL – 2280 HV Rijswijk Tel. (+31–70) 340–2040, Tx. 31 651 epo nl, Fax: (+31–70) 340–3016	Miot, Francesco

2

Information on patent family members

International application No
PCT/US2006/028088

Patent document cited in search report		Publication date	Patent family member(s)		Publication date
US 5414591	A	09-05-1995	NONE		
DE 29908370	U1	30-09-1999	DE	19921554 A1	25-11-1999
			DE	19921674 A1	18-11-1999
EP 1357778	A2	29-10-2003	NONE		

Form PCT/ISA/210 (patent family annex) (April 2005)

From the
INTERNATIONAL SEARCHING AUTHORITY

To:

see form PCT/ISA/220

PCT

WRITTEN OPINION OF THE INTERNATIONAL SEARCHING AUTHORITY

(PCT Rule 43*bis*.1)

	Date of mailing *(day/month/year)* see form PCT/ISA/210 (second sheet)
Applicant's or agent's file reference see form PCT/ISA/220	**FOR FURTHER ACTION** See paragraph 2 below

International application No.	International filing date *(day/month/year)*	Priority date *(day/month/year)*
PCT/US2006/028088	20.07.2006	04.08.2005

International Patent Classification (IPC) or both national classification and IPC
INV. H05K7/20

Applicant
LIEBERT CORPORATION

1. This opinion contains indications relating to the following items:

   - ☒ Box No. I Basis of the opinion
   - ☐ Box No. II Priority
   - ☐ Box No. III Non-establishment of opinion with regard to novelty, inventive step and industrial applicability
   - ☐ Box No. IV Lack of unity of invention
   - ☒ Box No. V Reasoned statement under Rule 43*bis*.1(a)(i) with regard to novelty, inventive step or industrial applicability; citations and explanations supporting such statement
   - ☐ Box No. VI Certain documents cited
   - ☐ Box No. VII Certain defects in the international application
   - ☐ Box No. VIII Certain observations on the international application

2. **FURTHER ACTION**

   If a demand for international preliminary examination is made, this opinion will usually be considered to be a written opinion of the International Preliminary Examining Authority ("IPEA") except that this does not apply where the applicant chooses an Authority other than this one to be the IPEA and the chosen IPEA has notifed the International Bureau under Rule 66.1*bis*(b) that written opinions of this International Searching Authority will not be so considered.

   If this opinion is, as provided above, considered to be a written opinion of the IPEA, the applicant is invited to submit to the IPEA a written reply together, where appropriate, with amendments, before the expiration of 3 months from the date of mailing of Form PCT/ISA/220 or before the expiration of 22 months from the priority date, whichever expires later.

   For further options, see Form PCT/ISA/220.

3. For further details, see notes to Form PCT/ISA/220.

☑ UPDATED ☐
Not Required
Written Opinion of the Int'l Searching Authority
Checked

Name and mailing address of the ISA:	Date of completion of this opinion	Authorized Officer
European Patent Office D-80298 Munich Tel. +49 89 2399 - 0 Tx: 523656 epmu d Fax: +49 89 2399 - 4465	see form PCT/ISA/210	Miot, Francesco Telephone No. +49 89 2399-2714

Form (PCT/ISA/237) (Cover Sheet) (April 2005)

Box No. I Basis of the opinion

1. With regard to the **language**, this opinion has been established on the basis of:

   ☒ the international application in the language in which it was filed

   ☐ a translation of the international application into , which is the language of a translation furnished for the purposes of international search (Rules 12.3(a) and 23.1 (b)).

2. With regard to any **nucleotide and/or amino acid sequence** disclosed in the international application and necessary to the claimed invention, this opinion has been established on the basis of:

   a. type of material:

      ☐ a sequence listing

      ☐ table(s) related to the sequence listing

   b. format of material:

      ☐ on paper

      ☐ in electronic form

   c. time of filing/furnishing:

      ☐ contained in the international application as filed.

      ☐ filed together with the international application in electronic form.

      ☐ furnished subsequently to this Authority for the purposes of search.

3. ☐ In addition, in the case that more than one version or copy of a sequence listing and/or table relating thereto has been filed or furnished, the required statements that the information in the subsequent or additional copies is identical to that in the application as filed or does not go beyond the application as filed, as appropriate, were furnished.

4. Additional comments:

Box No. V Reasoned statement under Rule 43*bis*.1(a)(i) with regard to novelty, inventive step or industrial applicability; citations and explanations supporting such statement

1. Statement

Novelty (N)	Yes:	Claims	
	No:	Claims	1,19
Inventive step (IS)	Yes:	Claims	4-18, 20-21, 23-25
	No:	Claims	2, 3, 22
Industrial applicability (IA)	Yes:	Claims	1-25
	No:	Claims	

2. Citations and explanations

see separate sheet

Re Item V.

1 Reference is made to the following document:
D1 : US 5 414 591 A (KIMURA HIDEYUKI [JP] ET AL) 9 May 1995 (1995-05-09)

2 INDEPENDENT CLAIM 1

2.1 The present application does not meet the criteria of Article 33(1) PCT, because the subject-matter of claim 1 is not new in the sense of Article 33(2) PCT.
Document D1 discloses (the references without parentheses applying to this document):
A cooling system for electronic equipment (see col. 1, lines 6-13 and figs. 1-2), comprising:
a) a closed loop refrigeration system comprising a compressor, a condenser coupled to the compressor, an expansion device coupled to the condenser, an evaporator coupled to the expansion device and to the compressor (see col. 14, lines 7-16), and a cooling system controller adapted to control operation of the refrigeration system (see col. 14, lines 43-54 and fig. 26), the refrigeration system being coupled to the cabinet (see fig. 1, 23, 26);
b) a cabinet 9 having a width, depth, and height (see fig. 1), the cabinet comprising:
(1) a first portion having a plurality of horizontal spaces adapted to contain electronic equipment 31a (see col. 4, lines 8-21); and
(2) a second portion extending at least partially along the height of the cabinet and adapted to house the evaporator, wherein the evaporator extends at least partially along the height (see col. 4, lines 20 - 25 and fig. 1-2,23); and
c) at least one fan 13 adapted to flow air through a first flow path through the evaporator in a circumferential horizontal direction around at least a partial periphery of the first portion and through the second portion (see fig. 23).

3 INDEPENDENT CLAIM 19

3.1 The present application does not meet the criteria of Article 33(1) PCT, because the subject-matter of claim 19 is not new in the sense of Article 33(2) PCT.

Document D1 discloses (the references without parentheses applying to this document):
a method of controlling temperature in an electronic equipment cabinet (see col. 4, lines 8-24; col. 13, lines 46-54 and fig. 1-2, 23), having a closed loop refrigeration system coupled thereto (col. 13, lines 64-68), comprising:
a) compressing and heating a refrigerant in the closed loop refrigeration system coupled to the cabinet (see col. 14, lines 7-16);
b) flowing the refrigerant into a condenser of the refrigeration system (see col. 14, lines 7-11);
c) flowing the refrigerant through an expansion device to cool the refrigerant (see col. 14, lines 7-11);
d) flowing the refrigerant through an evaporator and flowing air across surfaces of the evaporator to cool the air and heat the refrigerant (see col. 14, lines 7-11);
e) flowing the cooled air into the cabinet through a first flow path having a plurality of horizontal flow streams at different elevations in the cabinet across heated surfaces of the electronic equipment, the cooled air flowing in a circumferential horizontal direction around a periphery of the cabinet (see col. 12, lines 37-45 and fig. 23); and
f) returning at least a portion of the refrigerant for compressing (see col. 13, lines 67-68).

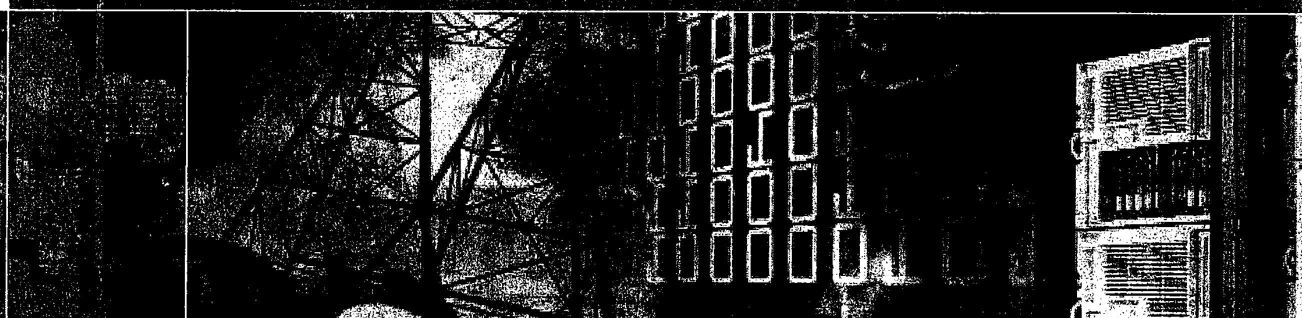
4 DEPENDENT CLAIMS 2, 3, 22

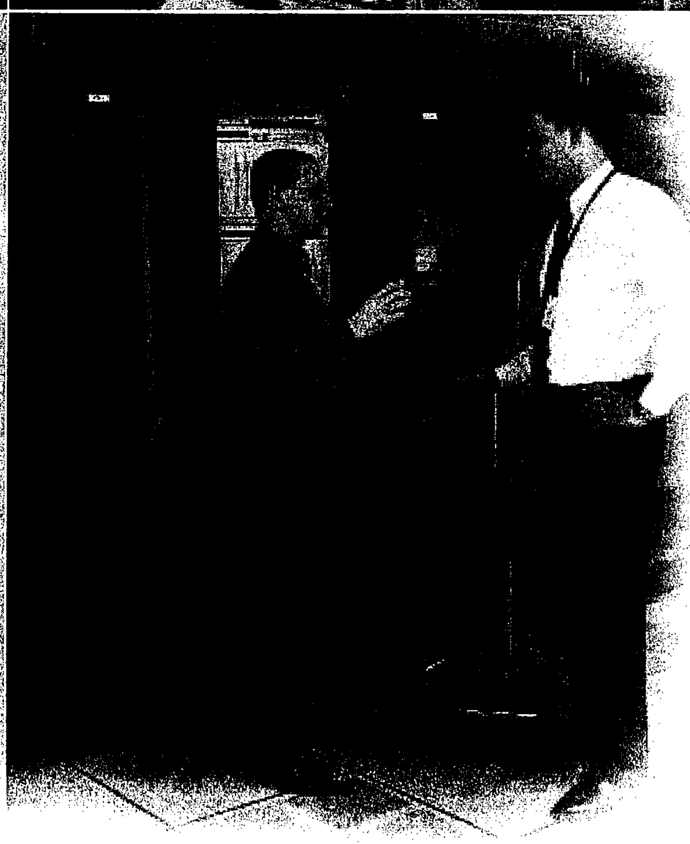
Dependent claims 2, 3 and 22 do not contain any features which, in combination with the features of any claim to which they refer, meet the requirements of the PCT in respect of novelty and/or inventive step (Article 33(2) and (3) PCT).

Liebert

Foundation™ Enclosure Systems

PROTECTION FOR TODAY'S NEEDS... AND TOMORROW'S OPPORTUNITIES





An Adaptive, Comprehensive and

Scalable Network Protection

System To Meet Ever-Changing

Computer Support Needs

EMERSON.
Network Power

TOO MUCH EQUIPMENT, TOO LITTLE PROTECTION

While sensitive electronics have changed over the past years, one thing remains constant – excessive heat, poor power quality and unauthorized access can damage or impair the operation of vital systems and peripherals.

The answer lies in finding the correct enclosure system to house critical components for both small and large facilities. There are, however, several key issues to consider before specifying one:

Evolving Technology

Computer equipment can change size and shape very rapidly. This means an enclosure you buy today must be suited for the equipment you buy a year from now.

Changes in Usage And Growth

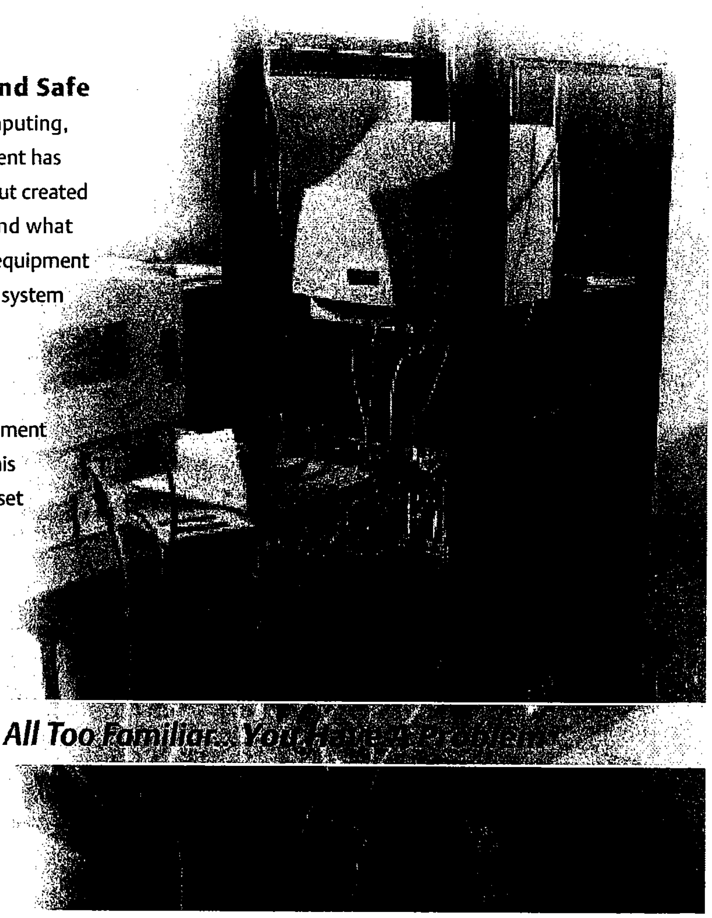
The need to quickly change equipment locations or configurations can be a real nightmare if a rack is not flexible or if there is no enclosure at all. Cabling presents still another challenge. It is important to keep wiring orderly and easy to trace. In the event of a malfunction of crossed wire, you could spend half your time just trying to determine which cable is at fault.

The Need To Keep It Organized And Safe

The downsizing and decentralization of computing, networking and telecommunications equipment has reduced the space required for these systems but created another dilemma...where do we put it all? And what about security? At best, an important piece of equipment could be accidentally disconnected. At worst, a system could be deliberately compromised.

Conditions Must Be Optimal

Often very critical pieces of networking equipment are exposed and unprotected. In some cases, this equipment may be housed in an equipment closet or other small room. But what about proper ventilation and cooling for that equipment? Overheated components may malfunction or shutdown completely.



If This Looks All Too Familiar... You Have A Problem!

Foundation Integration System

The Solution That Gives You Exactly The Right Amount Of Protection And Support

Adaptive

Foundation's highly flexible design accommodates varied support installation and application requirements. It is designed to provide easy access for equipment changes and relocation.

Comprehensive

Foundation features Liebert's full line of computer-grade support systems, including environmental, power, monitoring and security. Only Liebert has the capability to integrate all this support. Foundation provides "one stop shopping" for protection with single-source responsibility and complete solutions — now and in the future.

Scalable

Foundation has the flexibility to accommodate both changes in equipment and future growth. It allows you to start with a basic rack/enclosure configuration and scale up in features and level of protection – all the way to a Mini Computer Room. The system also provides significant cost savings by permitting you to maintain and build upon your original investment.

THE RIGHT ANSWER — NO MATTER WHAT YOUR NEEDS

The Foundation integration system is more than "just a rack"– it is the "foundation" of a mini-computer room. It is designed to deliver maximum flexibility today and to comprehensively accommodate future support needs in the rapidly changing network equipment environment.

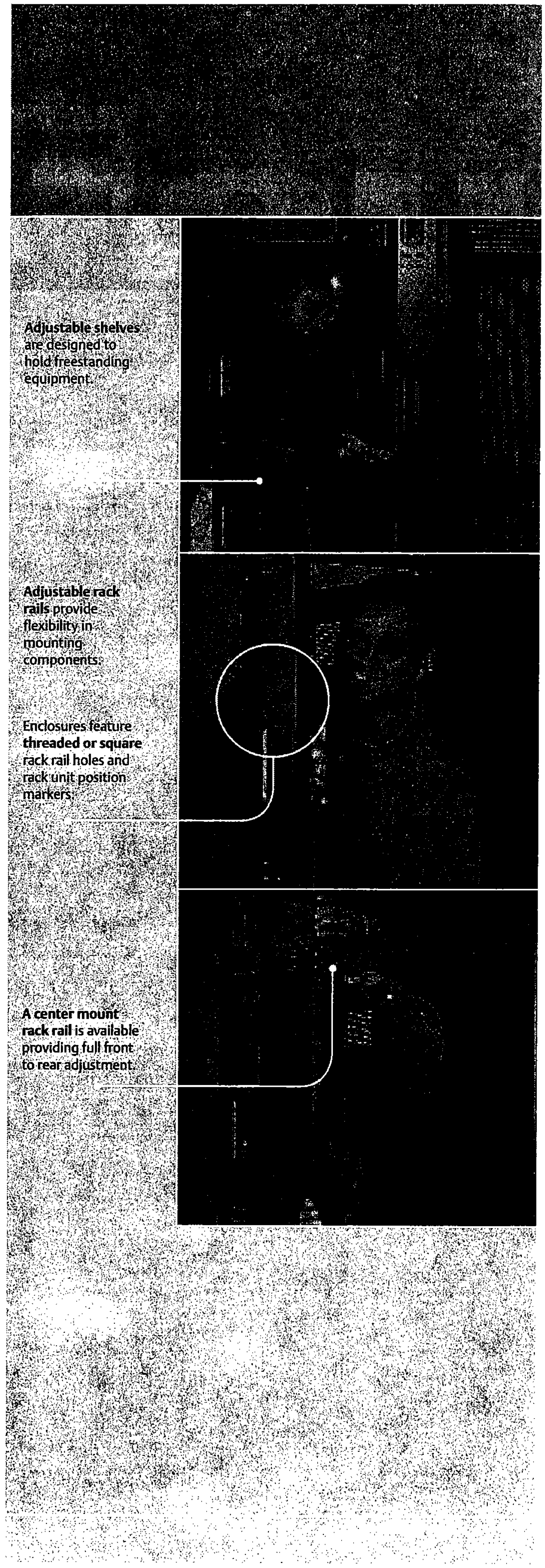
It can start simply as a basic integration system to house and organize network components. Or you can specify it at any level of protection up to the self-contained MCR (Mini Computer Room) by adding support systems including computer-grade air conditioning, UPS, monitoring capabilities and security features.

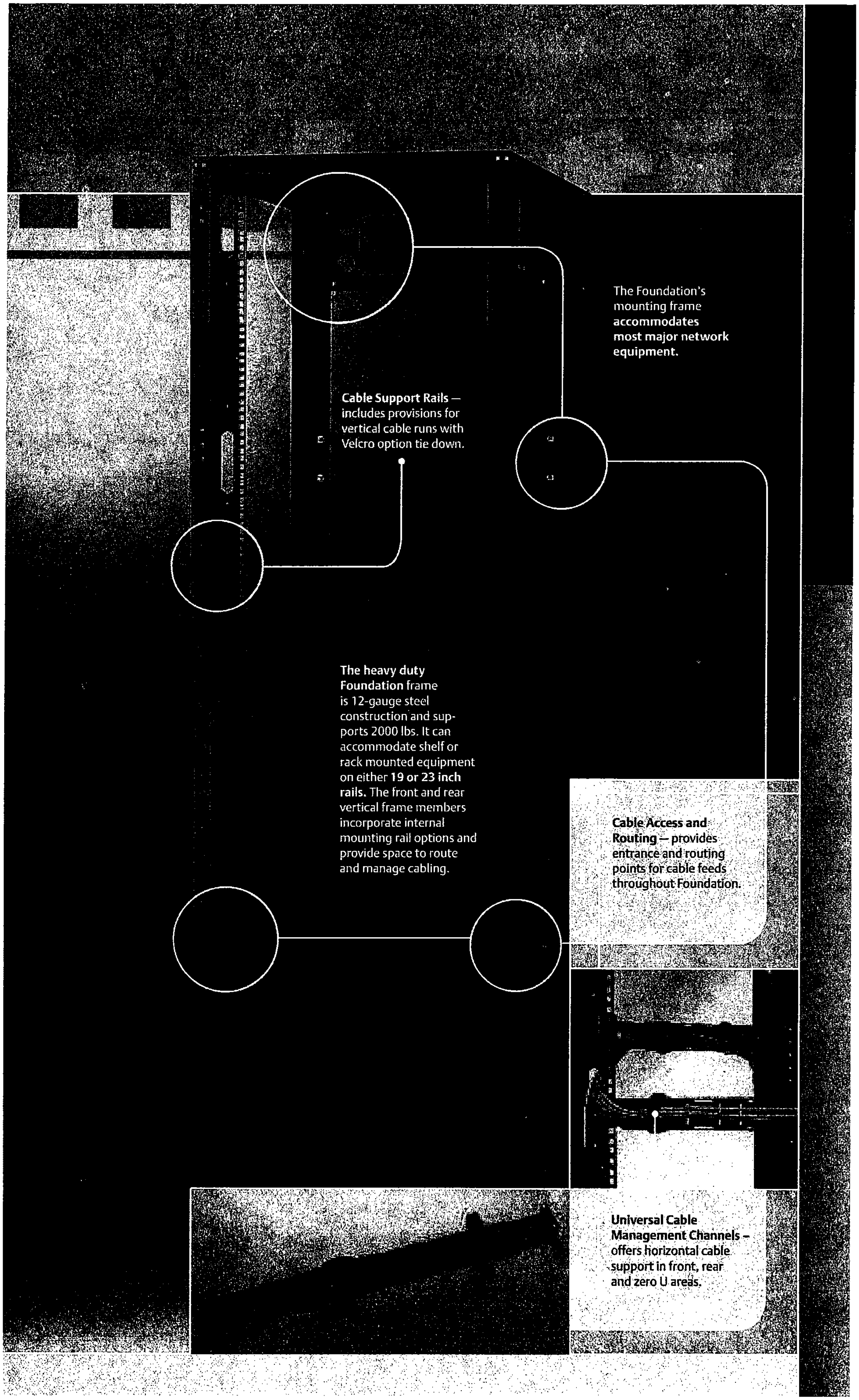
The beauty of the system is that you don't have to do it all at once. These features can be incorporated from the beginning or added at a later time to help meet both your current and future requirements.

Evolving Design

The Foundation system provides an organized, secure, controlled environment for your sensitive electronic equipment. A few of the major improvements to the Foundation over previous enclosure designs are:

- Adjustable Rack Rails
- Reversible Hinged Door
- Improved Cable Management
- Easy Access Side Panels
- Multiple Door Options
- Complete Upgradability
- Top Or Bottom Mount ECM Unit
- Lower Noise Level
- Energy Saving Features

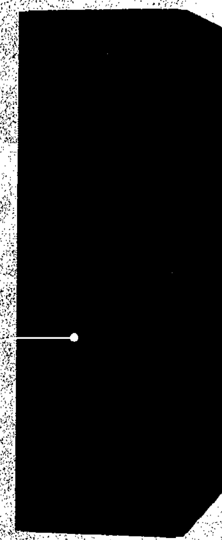



The Foundation's mounting frame accommodates most major network equipment.
Cable Support Rails — includes provisions for vertical cable runs with Velcro option tie down.
The heavy duty Foundation frame is 12-gauge steel construction and supports 2000 lbs. It can accommodate shelf or rack mounted equipment on either 19 or 23 inch rails. The front and rear vertical frame members incorporate internal mounting rail options and provide space to route and manage cabling.
Cable Access and Routing – provides entrance and routing points for cable feeds throughout Foundation.
Universal Cable Management Channels – offers horizontal cable support in front, rear and zero U areas.

6

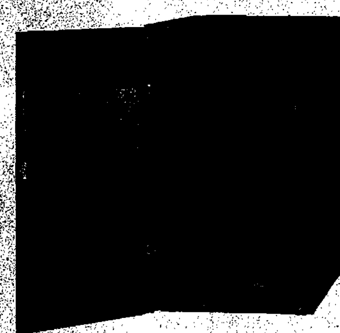
The Foundation Integration System features a robust, flexible design. Systems are selected as either "non-sealed" (open frame or enclosure) or as "sealed" (basis for NEMA 12 enclosure). Internal mounting rails are full height, adjustable position, sheetmetal mounting rails with EIA hole/spacing for equipment and/or mounting options. Front/rear rails are offered, as well as a center mount rail option. A choice of fixed and pullout shelves and keyboard tray is also available.

Various cutouts are located at the top, bottom and rear for cable entry. Leveling feet that can support 2,000 lbs. of total weight are provided, while casters are optional.

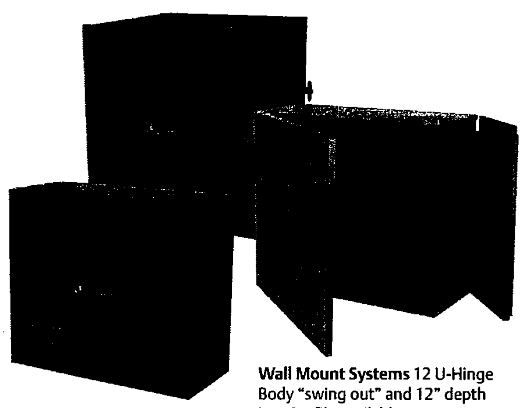


Doors with **perforated sheet metal** ventilation are available.

Where a minimal amount of equipment is needed, the Foundation 44" tall enclosure provides compact protection.

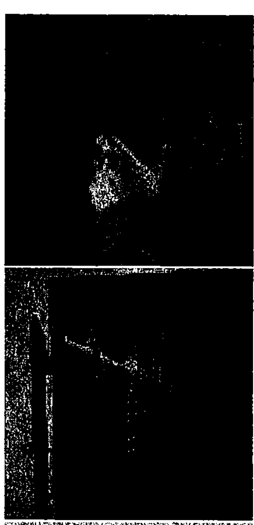


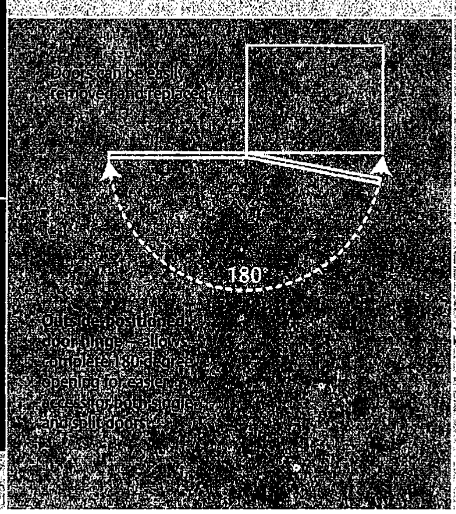
Wall Mount Systems 12 U-Hinge Body "swing out" and 12" depth Low Profile available.



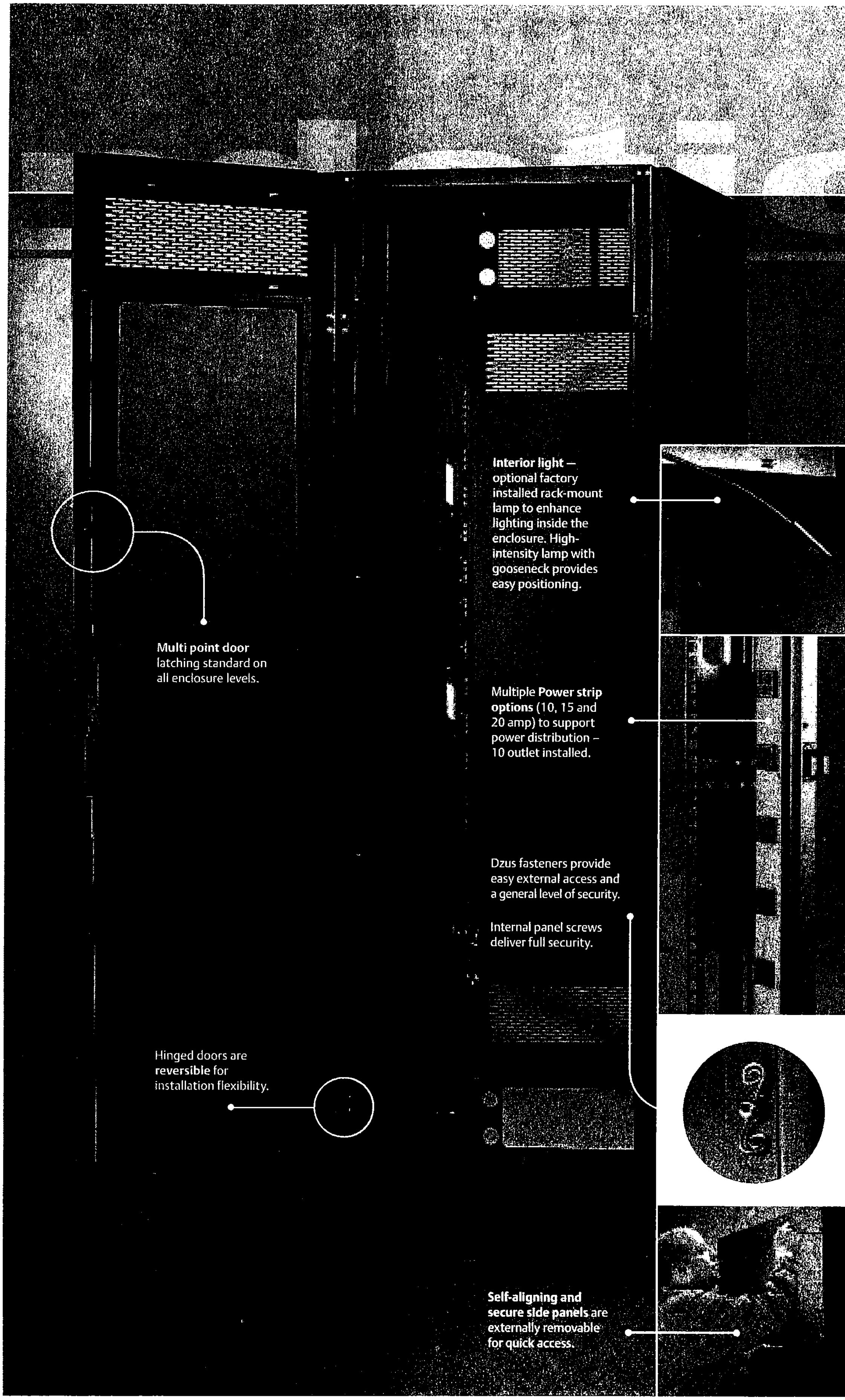
Bundled UPS systems available

Easy Access To Equipment

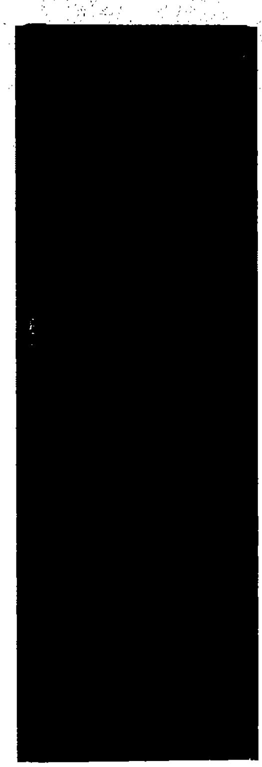




Dual-Split Rear Door available


Interior light – optional factory installed rack-mount lamp to enhance lighting inside the enclosure. High-intensity lamp with gooseneck provides easy positioning.
Multi point door latching standard on all enclosure levels.
Multiple **Power strip options** (10, 15 and 20 amp) to support power distribution – 10 outlet installed.
Dzus fasteners provide easy external access and a general level of security.
Internal panel screws deliver full security.
Hinged doors are **reversible** for installation flexibility.
Self-aligning and secure side panels are externally removable for quick access.

TOTAL PROTECTION UNDER ONE ROOF



The Foundation MCR (Mini Computer Room) provides the peace-of-mind that comes from knowing that Liebert's expertise and experience are protecting your vital network equipment and the integrity of your data.

Based on the industry award winning Little Glass House® design, the Foundation MCR self-contained integration system brings together all of the elements necessary to ensure the long-term viability of network components or other critical electronics.

A load-sized, computer-grade air conditioner can be located at either the top or bottom of the enclosure, driving cool air through sensitive equipment on all levels. A back-up cooling system ensures environmental security.

Power is supplied and protected through either a Liebert GXT 2U on-line UPS or a PowerSure® Interactive UPS.

And with a built-in Liebert SiteNet® Integrator, the Foundation MCR becomes an intelligent network peripheral, capable of monitoring conditions and initiating pre-set actions when problems arise.

Critical Features Designed To Protect Critical Equipment

Capabilities

The capabilities and features of the Foundation MCR are designed to provide maximum protection for systems housed within the enclosure:

Comprehensive, integrated Liebert design — combines computer-grade support systems, including cooling, power, monitoring and security into a single, pre-tested system.

Mobile design for quick deployment — let's you put a self-contained mini-computer room right where you need it, today or tomorrow.

Agency approved as a system — pre-qualified and ready for installation.

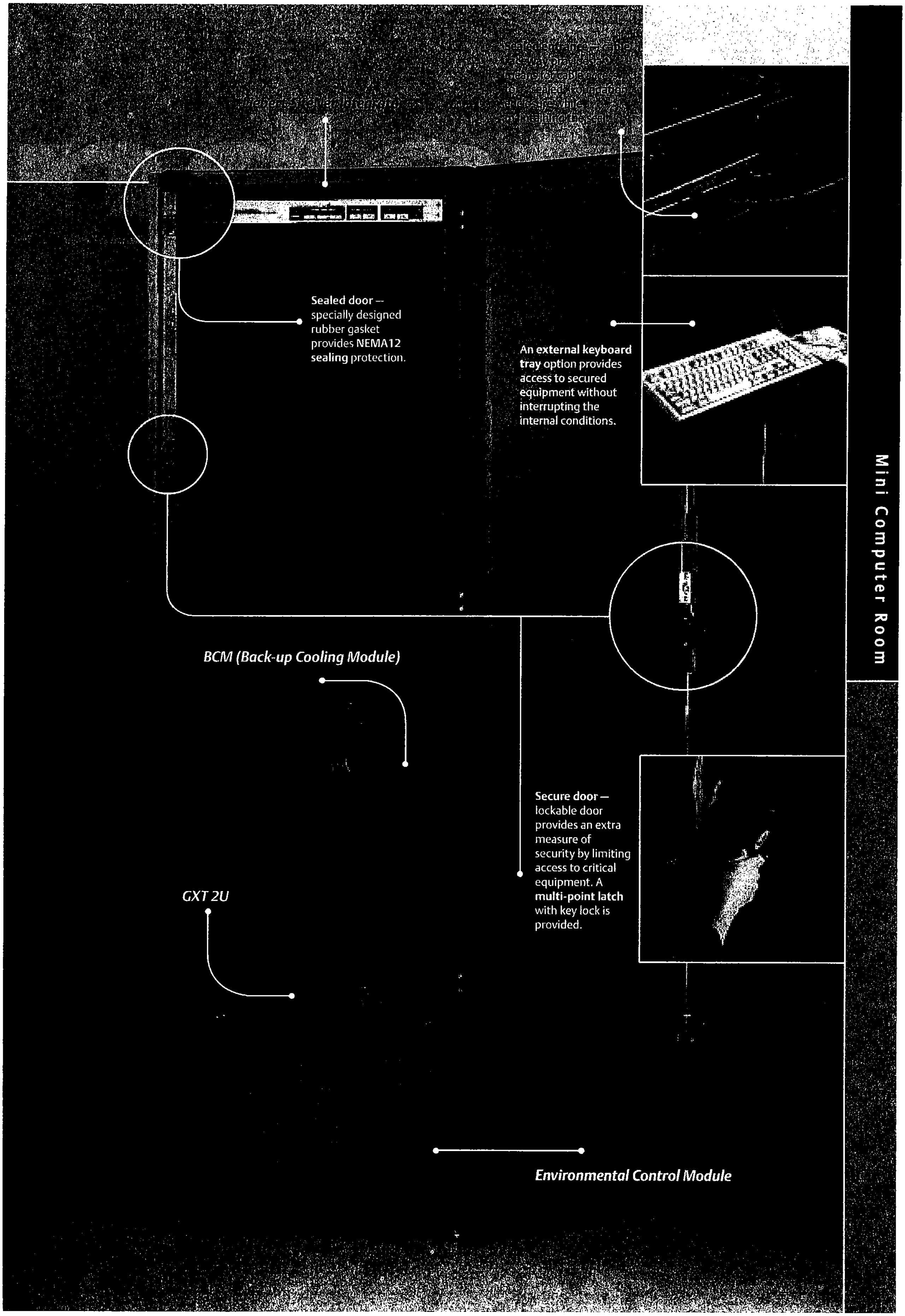
Features

ECM (Environmental Control Module) — computer-grade air conditioning load matched to UPS.

BCM (Back-up Cooling Module) — provides cooling in the event of a power loss or can be utilized to reduce energy consumption with the **BCM Energy Saver Control.**

Liebert On-Line or Line Interactive UPS — provides back-up power protection.

SiteNet Integrator — alarms and status monitoring.



Sealed door — specially designed rubber gasket provides NEMA12 sealing protection.
An external keyboard tray option provides access to secured equipment without interrupting the internal conditions.
Mini Computer Room
BCM (Back-up Cooling Module)
Secure door — lockable door provides an extra measure of security by limiting access to critical equipment. A multi-point latch with key lock is provided.
GXT 2U
Environmental Control Module

CREATING THE PERFECT ENVIRONMENT FOR YOUR CRITICAL SYSTEMS

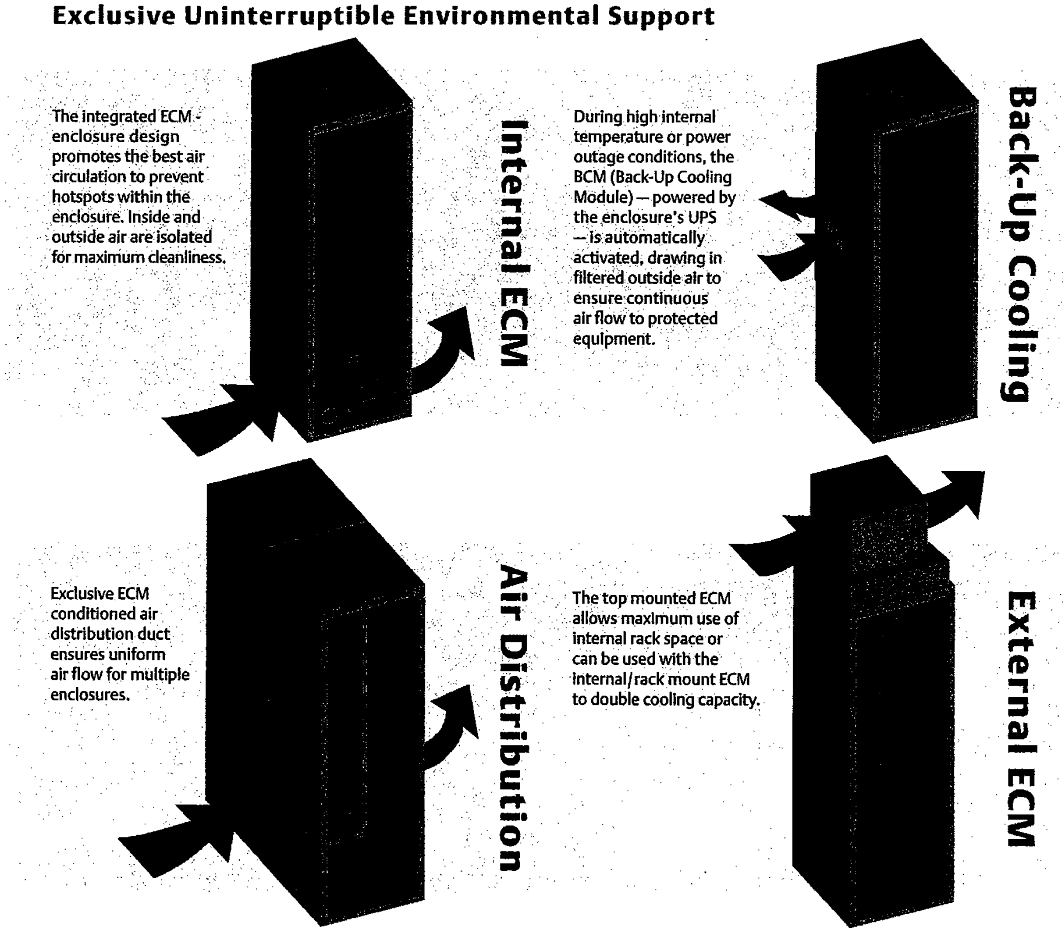
Only Liebert can offers a real choice of computer-grade cooling options designed to meet your most demanding protection needs.

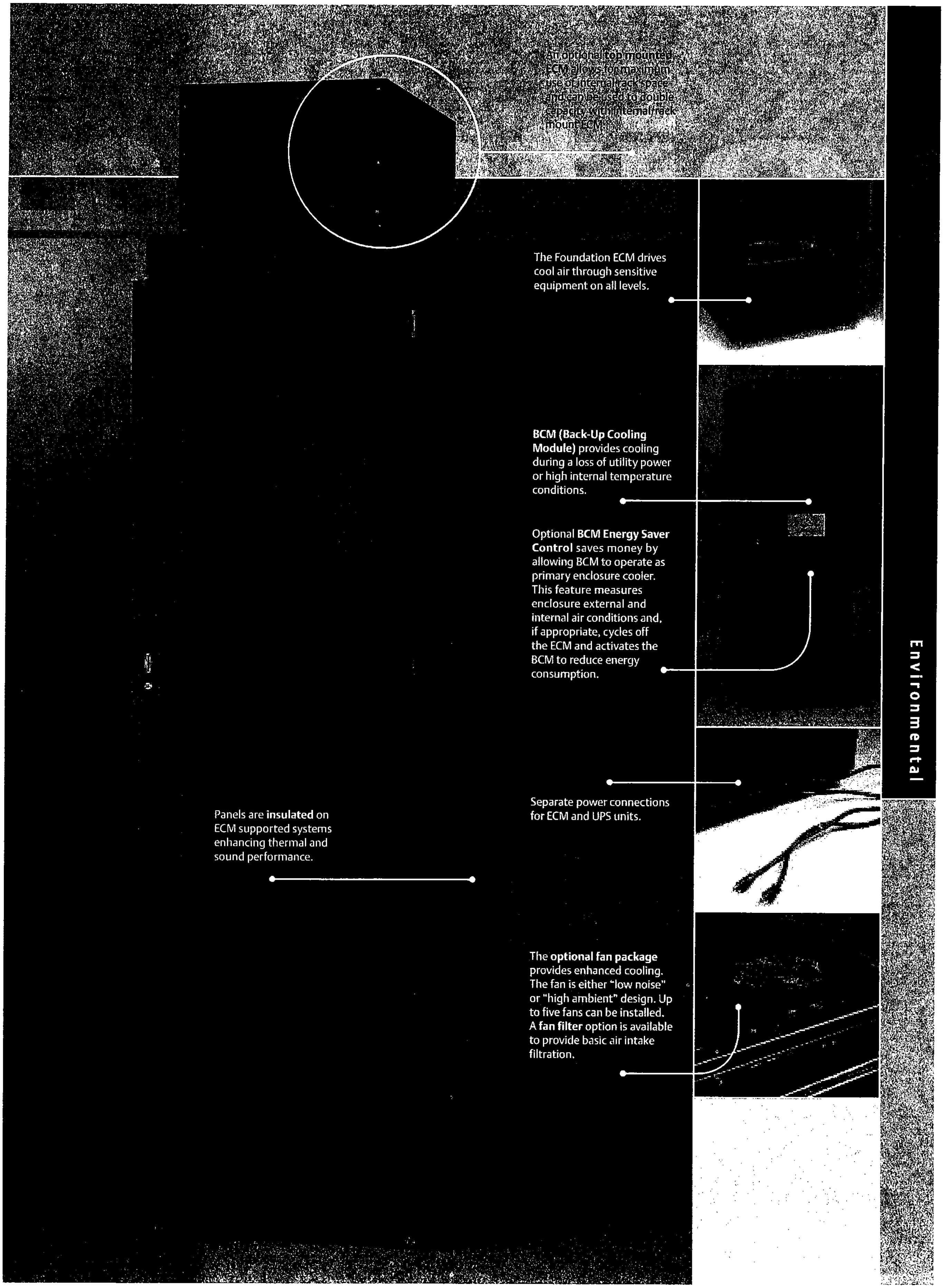
For systems without ECM air conditioning, **fan cooling** provides enhanced heat removal. Either 1 or 2 "low noise" or "high ambient" fans can be factory installed – up to 5 fans can be installed in the field. Fans can be located either on top or back panel of the Foundation.

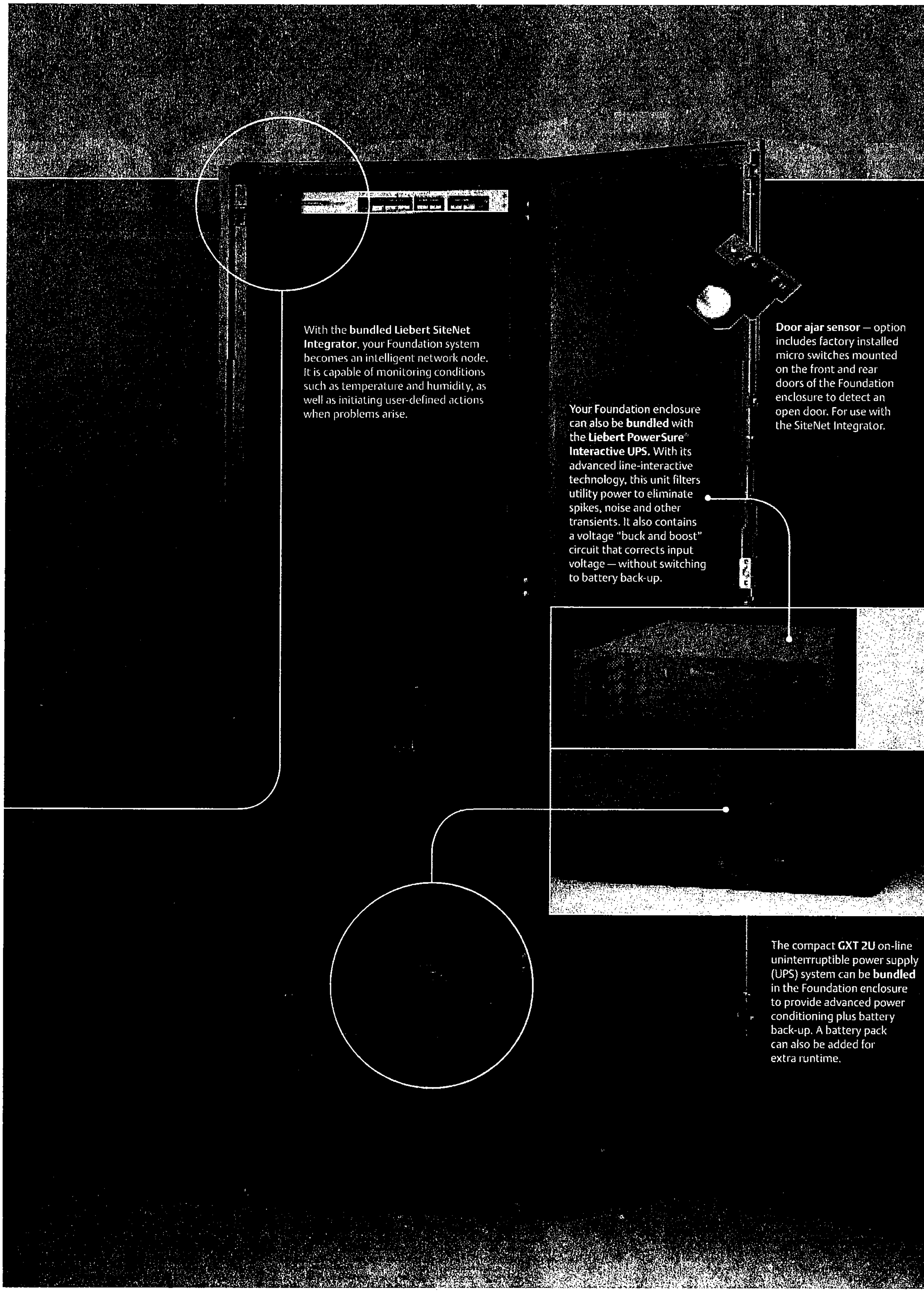
Your Foundation Integration System is available with a Liebert **ECM (Environmental Control Module)**. The ECM is a self-contained, air conditioning system that provides round-the-clock cooling to components housed within a Foundation enclosure. The ECM features low-noise operation, suitable for use in occupied spaces and includes an automatic condensate reevaporation system. A hot gas by-pass provides automatic load matching and enhanced ECM life. All systems utilize "green" R-407C refrigerant.

A **BCM (Back-up Cooling Module)** can be added to the ECM to insure cooling in emergency conditions. With the addition of the Energy Saver Control the BCM can operate in "economizer" mode to save energy when conditions are right.

Exclusive Uninterruptible Environmental Support




An optional top mounted ECM allows for maximum use of internal rack space and can be used to double capacity with internal rack mount ECM.
The Foundation ECM drives cool air through sensitive equipment on all levels.
BCM (Back-Up Cooling Module) provides cooling during a loss of utility power or high internal temperature conditions.
Optional **BCM Energy Saver Control** saves money by allowing BCM to operate as primary enclosure cooler. This feature measures enclosure external and internal air conditions and, if appropriate, cycles off the ECM and activates the BCM to reduce energy consumption.
Separate power connections for ECM and UPS units.
Panels are **insulated** on ECM supported systems enhancing thermal and sound performance.
The **optional fan package** provides enhanced cooling. The fan is either "low noise" or "high ambient" design. Up to five fans can be installed. A **fan filter** option is available to provide basic air intake filtration.
Environmental



With the **bundled Liebert SiteNet Integrator**, your Foundation system becomes an intelligent network node. It is capable of monitoring conditions such as temperature and humidity, as well as initiating user-defined actions when problems arise.

Your Foundation enclosure can also be **bundled** with the **Liebert PowerSure® Interactive UPS.** With its advanced line-interactive technology, this unit filters utility power to eliminate spikes, noise and other transients. It also contains a voltage "buck and boost" circuit that corrects input voltage — without switching to battery back-up.

Door ajar sensor — option includes factory installed micro switches mounted on the front and rear doors of the Foundation enclosure to detect an open door. For use with the SiteNet Integrator.

The compact **GXT 2U** on-line uninterruptible power supply (UPS) system can be **bundled** in the Foundation enclosure to provide advanced power conditioning plus battery back-up. A battery pack can also be added for extra runtime.

14

Specifications

ENCLOSURE DIMENSIONS

Model	Overall Frame Dimensions			Rack		Adjustable Rack Depth		Internal Rack Height	
	Height*	Width**	Depth***	Width	Available Width	B****			
_ = D or K	In. (mm)			In. (mm)	In. (mm)	Max In.(mm)	Min In.(mm)	RACK U	In. (mm)
H_440	42 (1067)	23.5 (597)	30 (762)	19 (483)	17.8 (450)	22.5 (571.5)	18.5 (470)	22	38.5 (978)
H_448	42 (1067)	23.5 (597)	38 (965)	19 (483)	17.8 (450)	30.5 (775)	26.5 (673)	22	38.5 (978)
R_440	42 (1067)	27.5 (699)	30 (762)	23 (584)	22.8 (580)	22.5 (571.5)	18.5 (470)	22	38.5 (978)
R_448	42 (1067)	27.5 (699)	38 (965)	23 (584)	22.8 (580)	30.5 (775)	26.5 (673)	22	38.5 (978)
H_780	77 (1956)	23.5 (597)	30 (762)	19 (483)	17.8 (450)	22.5 (571.5)	18.5 (470)	42	73.5 (1867)
H_788	77 (1956)	23.5 (597)	38 (965)	19 (483)	17.8 (450)	30.5 (775)	26.5 (673)	42	73.5 (1867)
R_780	77 (1956)	27.5 (699)	30 (762)	23 (584)	22.8 (580)	22.5 (571.5)	18.5 (470)	42	73.5 (1867)
R_788	77 (1956)	27.5 (699)	38 (965)	23 (584)	22.8 (580)	30.5 (775)	26.5 (673)	42	73.5 (1867)
H_840	84 (2134)	23.5 (597)	30 (762)	19 (483)	17.8 (450)	22.5 (571.5)	18.5 (470)	46	80.5 (2045)
H_848	84 (2134)	23.5 (597)	38 (965)	19 (483)	17.8 (450)	30.5 (775)	26.5 (673)	46	80.5 (2045)
R_840	84 (2134)	27.5 (699)	30 (762)	23 (584)	22.8 (580)	22.5 (571.5)	18.5 (470)	46	80.5 (2045)
R_848	84 (2134)	27.5 (699)	38 (965)	23 (584)	22.8 (580)	30.5 (775)	26.5 (673)	46	80.5 (2045)

* CASTERS ADD 1.5" TO OVERALL HEIGHT OF FRAME = 2000mm.
** SIDE PANELS ADD 0.75" EACH TO OVERALL WIDTH OF FRAME.
*** DOORS ADD 0.1" EACH TO OVERALL DEPTH OF FRAME, BCM OPTION ADDS AN ADDITIONAL 3.00" TO OVERALL DEPTH OF FRAME.
**** MAX DIMENSION IS FOR EX-FACTORY CONFIGURATION. RAILS CAN BE INVERTED TO PROVIDE AN ADDITIONAL 4.00" OF ADJUSTMENT.

FAN PERFORMANCE DATA

Model Number	Fans	Airflow	Sound		Input Power (1PH)			
		CFM	dBa	Volts	Hertz	FLA	WSA	OPD
FAN1000L-60	1	114	47	120	60	0.2	0.3	15
FAN2000L-60	2	228	49	120	60	0.4	0.5	15
FAN1000L-50	1	94	45	230	50	0.1	0.2	10
FAN2000L-50	2	188	47	230	50	0.2	0.4	10
FAN1000H-60	1	235	59	120	60	0.3	0.4	15
FAN2000H-60	2	470	61	120	60	0.6	0.8	15
FAN1000H-50	1	200	57	230	50	0.1	0.1	10
FAN2000H-50	2	400	59	230	50	0.2	0.3	10

ECM PERFORMANCE DATA

Model Number	Rated Capacity	Supported Load	Max Ambient	Height	Width	Depth	Total Heat Rej.	Input Power (1PH)					Plug	Sound
	BTUH (Watts)	BTUH (Watts)		In (mm) - U	In (mm)	In (mm)	BTUH (Watts)	Volts	Hertz	FLA	WSA	OPD		Lpa (1.5 m)
ECM1000L*-C60	5315 (1557)	2811 (824)	105°F / 41°C	12.25 (311)-7	17.43 (443)	29 (737)	7146 (2094)	120	60	7.3	8.6	15	NEMA 5-15P	52
ECM1000L*-C50	5306 (1555)	2811 (824)	100°F / 38°C	12.25 (311)-7	17.43 (443)	29 (737)	7698 (2255)	230	50	3.5	4.1	10	IEC320-10A	52
ECM2000L*-C60	6897 (2021)	5621 (1647)	105°F / 41°C	12.25 (311)-7	17.43 (443)	29 (737)	10935 (3204)	120	60	9.8	11.7	15	NEMA5-15P	52
ECM2000L*-C50	6708 (1965)	5621 (1647)	100°F / 38°C	12.25 (311)-7	17.43 (443)	29 (737)	10375 (3040)	230	50	4.8	5.7	10	IEC320-10A	52

"* T (top mount) and R (rack mount). Top mount weight does not include interface plenum. The interface plenum for a 19"" rack x 30 "" deep cabinet is 38 lbs."
Sound data based on sound pressure A- weighted scale for free field spherical radiation at 1.5 meters from cabinet.
Sound data reflects only rack mount design. Consult factory for top mount data.

BCM PERFORMANCE DATA

Model Number	Rated Capacity	Supported Load	Max Ambient	Height	Width	Depth	Weight	Total Heat Rej.	Input Power (1PH)					Sound
	BTUH (Watts)	BTUH (Watts)		In (mm) - U	In (mm)	In (mm)	lbs (kg)	BTUH (Watts)	Volts	Hertz	FLA	WSA	OPD	Lpa (1.5 m)
BCM 1000L-60	N/A	2811 (824)	105°F / 41°C	35.0 (889)	15.5 (393.7)	3.75 (95.2)	47 (21.3)	3038 (890)	120	60	1.0	1.3	15	57
BCM 2000L-60	N/A	2811 (824)	105°F / 41°C	35.0 (889)	15.5 (393.7)	3.75 (95.2)	47 (21.3)	3038 (890)	120	60	2.0	2.5	15	59
BCM 1000L-50	N/A	5621 (1647)	105°F / 41°C	35.0 (889)	15.5 (393.7)	3.75 (95.2)	47 (21.3)	5918 (1734)	230	50	0.5	0.6	10	55
BCM 2000L-50	N/A	5621 (1647)	105°F / 41°C	35.0 (889)	15.5 (393.7)	3.75 (95.2)	47 (21.3)	5918 (1734)	230	50	1.0	1.2	10	57

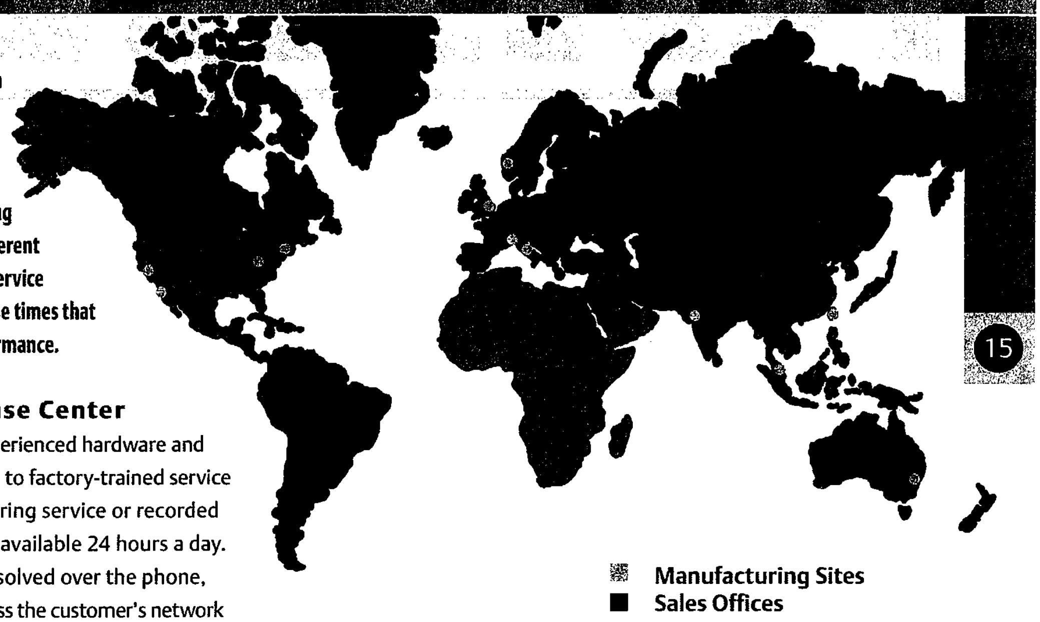
Above BCM weight includes rear door weight of 17 Lbs.
Sound data based on sound pressure A- weighted scale for free field spherical radiation at 1.5 meters from cabinet.

UPS PERFORMANCE DATA

Model Number	VA / Watts	Input Power (1PH)					Model Number	VA / Watts	Input Power (1PH)				
		Volts	Hertz	WSA	OPD	Plug			Volts	Hertz	WSA	OPD	Plug
GXT2-1000RT120	1000 / 700	120	60	15	15	NEMA 5-15P	PS700RM-120	700 / 450	120	60	15	15	NEMA 5-15P
GXT2-1500RT120	1500 / 1050	120	60	15	15	NEMA 5-15P	PS1000RM-120	1000 / 670	120	60	15	15	NEMA 5-15P
GXT2-2000RT120	2000 / 1400	120	60	20	20	NEMA 5-20P	PS1400RM-120	1400 / 950	120	60	15	15	NEMA 5-15P
GXT2-3000RT120	3000 / 2100	120	60	30	30	NEMA L5-30P	PS2200RM-120	2200 / 1600	120	60	30	30	NEMA L5-30P
GXT2-1000RT230	1000 / 700	230	50	10	10	IEC320-10A	PS700RM-230	700 / 450	230	50	10	10	IEC320-10A
GXT2-1500RT230	1500 / 1050	230	50	10	10	IEC320-10A	PS1000RM-230	1000 / 670	230	50	10	10	IEC320-10A
GXT2-2000RT230	2000 / 1400	230	50	10	10	IEC320-10A	PS1400RM-230	1400 / 950	230	50	10	10	IEC320-10A
GXT2-3000RT230	3000 / 2100	230	50	16	16	IEC320-16A	PS2200RM-230	2200 / 1600	230	50	16	16	IEC320-16A

Service and Support Whenever You Need It

Liebert has the largest service team in the power protection industry: more than 100 service centers around the world. Customers get professional support without having to hire and train staff, or retain different vendors at different sites. Liebert service customers get service with response times that set the industry standard for performance.



The Customer Response Center

The Center is staffed with experienced hardware and software specialists with access to factory-trained service technicians. There is no answering service or recorded message...representatives are available 24 hours a day. Many times, problems can be solved over the phone, especially when Liebert can access the customer's network via modem to download an event log for analysis and response. Liebert can also monitor battery status and other factors that affect the reliability of your power protection, then conduct regular maintenance and appropriate service to ensure maximum readiness.

Field Automation System

This proprietary database puts a complete set of customer information in the hands of Liebert Customer Engineers in the field. Stored in laptops and accessible via wireless modem, the Field Automation capability includes a complete service history of all installed equipment, and a list of current system parameters, including traps and alarms.

Remote Monitoring

For organizations that want a 24-hour watch over their critical systems, Liebert can provide automated diagnostic software that connects vital network nodes to the Customer Response Center. The Remote Monitoring software instantly locates alarms, measures criticality, and identifies appropriate responses. Within seconds, Liebert is implementing a pre-set response plan. And this happens around the clock...some Remote Monitoring customers arrive at work to find that an overnight problem has already been detected and solved.

TOTAL CONFIDENCE IN THE DECISION YOU'VE MADE.

Foundation™ Enclosure Systems

No organization in the world today has a better understanding of exactly what it takes to keep critical information and industrial processes operating continuously than Liebert.

We are the only company in this business that maintains a strong local presence of Representatives, Distributors and Resellers. This resource, coupled with our broad product line, gives Liebert the ability to create a "tailored solution" that will meet your protection needs precisely and efficiently.

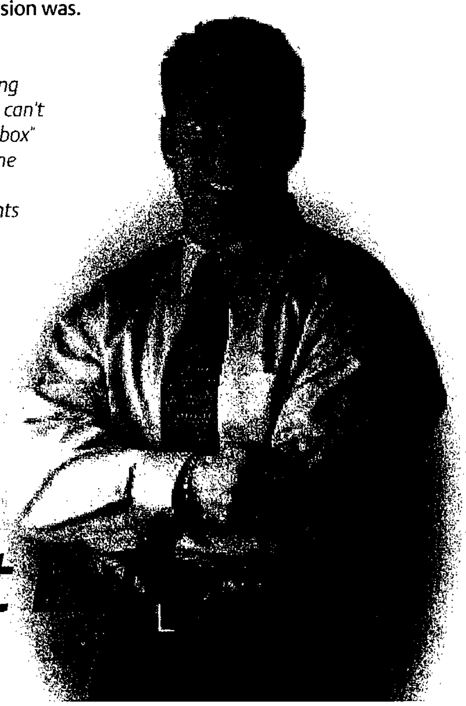
There are Liebert systems designed for nearly every application — from basic protection for network PCs, servers or point-of-sale terminals...to highly engineered systems for computer rooms, telecommunications centers, Internet hosting sites, colocation facilities and industrial control rooms. But no matter what the size or complexity, the availability of these critical electronic systems is Liebert's primary focus.

With your purchase of a Liebert product, you are buying into a company that stands behind its products. You are also aligning yourself with an organization that has a reputation for quality and reliability that is second to none.

After the sale, Liebert provides comprehensive support wherever and whenever it's needed, with the largest service organization in the industry.

In the systems protection business it's when you need someone to count on that you find out whether you've made the right choice. Liebert customers — many of them with us for over three decades — already know how good their decision was.

Truly protecting complex computing and communications systems just can't be done with a simple "out-of-the-box" solution. Liebert helps you make the ***smart decision*** *— one that will consistently meet your requirements for availability and reliability.*



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35028 Piove Di Sacco (PD)
Italy
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39 049 5841 257 FAX

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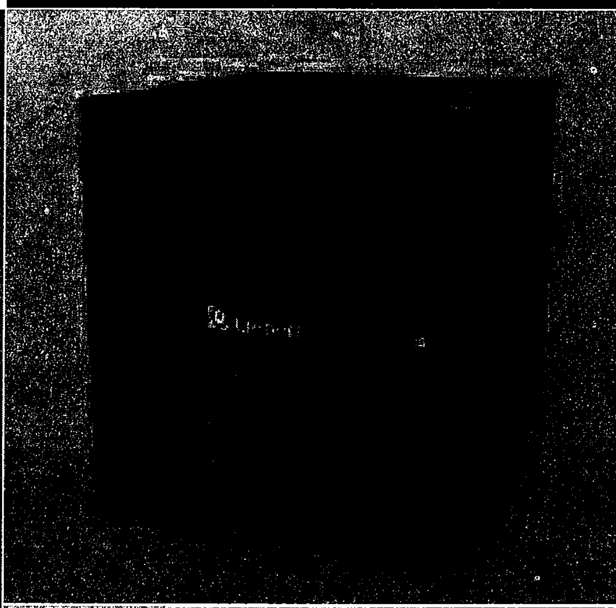
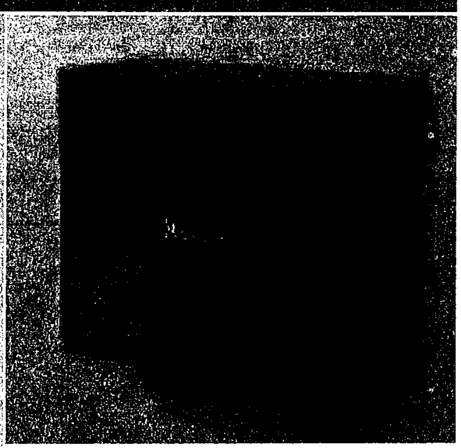
EMERSON. Network Power

Liebert Foundation™ Enclosure Systems

WALL MOUNT EQUIPMENT ENCLOSURES DESIGNED FOR TIGHT SPACES





When floor space to secure electronic equipment is not available inside equipment closets or small rooms, often space on the wall can be found.

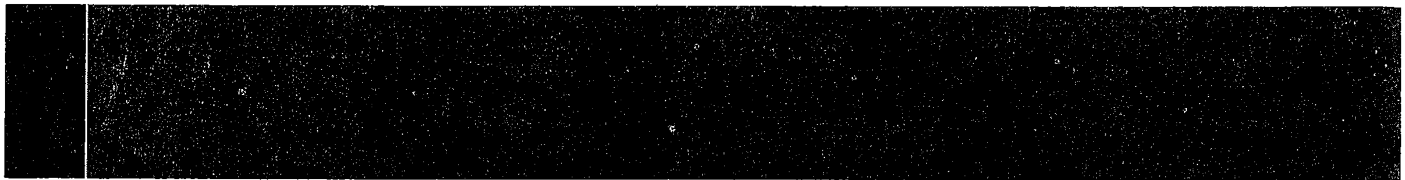
Liebert Foundation Wall Mount Enclosure systems provide an effective solution for these limited space equipment mounting applications – where both protection and security are required.

Two standard designs are offered: a Hinged Body model (with a nominal height, depth, width of 24" x 24" x 24") and a Low Profile unit (nominal 24" x 12" x 24").

All Liebert Foundation Wall Mount systems include:

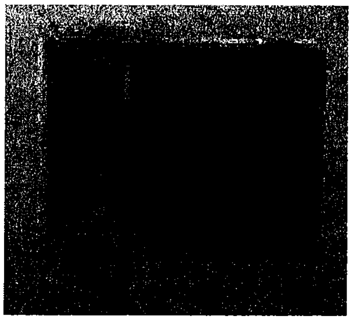
- Vented front and sides for enhanced heat rejection.
- Secure latching doors with key lock.
- Knockouts for top, bottom and side cable access.
- Reversible positioning for left/right access door hinging.
- Optional feet or casters for floor mounting.
- Grounding lug.
- Durable powder paint coated black.

EMERSON
Network Power



Liebert Foundation™

Enclosure Systems



Liebert Foundation Wall Mount 24" x 12" Low Profile Model

The 24" x 12" Low Profile model includes:

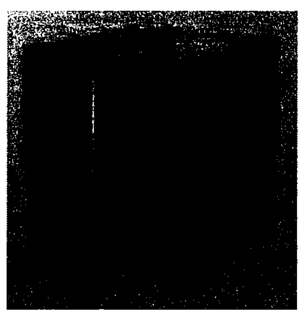
- Reduced depth for tight spaces.
- 12U front and 6U side adjustable position 19" rack mounting rails.
- Unique side access rack mounting section for deep components.
- Supports 100 pounds of user equipment.

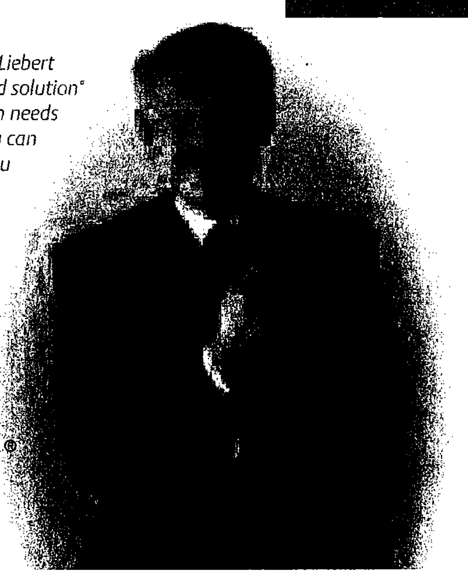
Liebert Foundation Wall Mount 24" x 24" Hinged Body Model

The 24" x 24" Hinged Body model features:

- 12U front and rear adjustable position 19" rack rails.
- Swing-out body design to allow for easy rear equipment access.
- Provisions for field addition of optional top and/or bottom cooling fan.
- Supports up to 175 pounds of user equipment.
- Provisions to support mounting options such as fixed or pullout shelf.

Wall mounting hardware field supplied per local codes and requirements.





Our broad product line gives Liebert the ability to create a "tailored solution" that will meet your protection needs precisely and efficiently — you can always count on us to give you the best answer every time.

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PATENT COOPERATION TREATY

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INTERNATIONAL SEARCH REPORT

(PCT Article 18 and Rules 43 and 44)

Applicant's or agent's file reference 021944-080WO	**FOR FURTHER ACTION** see Form PCT/ISA/220 as well as, where applicable, item 5 below.	
International application No. PCT/US2004/040407	International filing date *(day/month/year)* 02/12/2004	(Earliest) Priority Date *(day/month/year)* 05/12/2003
Applicant LIEBERT CORPORATION		

This International Search Report has been prepared by this International Searching Authority and is transmitted to the applicant according to Article 18. A copy is being transmitted to the International Bureau.

This International Search Report consists of a total of 6 sheets.

☐ It is also accompanied by a copy of each prior art document cited in this report.

1. **Basis of the report**
   a. With regard to the **language,** the international search was carried out on the basis of the international application in the language in which it was filed, unless otherwise indicated under this item.
      ☐ The international search was carried out on the basis of a translation of the international application furnished to this Authority (Rule 23.1(b)).
   b. ☐ With regard to any **nucleotide and/or amino acid sequence** disclosed in the international application, see Box No. I.

2. ☐ **Certain claims were found unsearchable** (See Box II).

3. ☒ **Unity of invention is lacking** (see Box III).

4. With regard to the **title,**
   ☒ the text is approved as submitted by the applicant.
   ☐ the text has been established by this Authority to read as follows:

5. With regard to the **abstract,**
   ☐ the text is approved as submitted by the applicant.
   ☒ the text has been established, according to Rule 38.2(b), by this Authority as it appears in Box No. IV. The applicant may, within one month from the date of mailing of this international search report, submit comments to this Authority.

6. With regard to the **drawings,**
   a. the figure of the **drawings** to be published with the abstract is Figure No. 1
      ☒ as suggested by the applicant.
      ☐ as selected by this Authority, because the applicant failed to suggest a figure.
      ☐ as selected by this Authority, because this figure better characterizes the invention.
   b. ☐ none of the figures is to be published with the abstract.

INTERNATIONAL SEARCH REPORT

International application No.
PCT/US2004/040407

Box II Observations where certain claims were found unsearchable (Continuation of item 2 of first sheet)

This International Search Report has not been established in respect of certain claims under Article 17(2)(a) for the following reasons:

1. ☐ Claims Nos.:
because they relate to subject matter not required to be searched by this Authority, namely:

2. ☐ Claims Nos.:
because they relate to parts of the International Application that do not comply with the prescribed requirements to such an extent that no meaningful International Search can be carried out, specifically:

3. ☐ Claims Nos.:
because they are dependent claims and are not drafted in accordance with the second and third sentences of Rule 6.4(a).

Box III Observations where unity of invention is lacking (Continuation of item 3 of first sheet)

This International Searching Authority found multiple inventions in this international application, as follows:

see additional sheet

1. ☐ As all required additional search fees were timely paid by the applicant, this International Search Report covers all searchable claims.

2. ☐ As all searchable claims could be searched without effort justifying an additional fee, this Authority did not invite payment of any additional fee.

3. ☐ As only some of the required additional search fees were timely paid by the applicant, this International Search Report covers only those claims for which fees were paid, specifically claims Nos.:

4. ☒ No required additional search fees were timely paid by the applicant. Consequently, this International Search Report is restricted to the invention first mentioned in the claims; it is covered by claims Nos.:

1-7, 9

Remark on Protest

☐ The additional search fees were accompanied by the applicant's protest.

☐ No protest accompanied the payment of additional search fees.

Form PCT/ISA/210 (continuation of first sheet (2)) (January 2004)

INTERNATIONAL SEARCH REPORT

Box No. IV Text of the abstract (Continuation of item 5 of the first sheet)

A cooling system (10) for transferring heat from a heat load to an environment has a volatile working fluid. The cooling system includes first (12) and second (14) cooling cycles that are thermally connected to each other. The first cooling cycle is not a vapor compression cycle and includes a pump (20), an air-to-fluid heat exchanger (30), and a fluid-to-fluid heat exchanger (40). The second cooling cycle can include a chilled water system for transferring heat from the fluid-to-fluid heat exchanger to the environment. Alternatively, the second cooling cycle can include a vapor compression system for transferring heat from the fluid-to-fluid heat exchanger to the environment.

FURTHER INFORMATION CONTINUED FROM PCT/ISA/ 210

This International Searching Authority found multiple (groups of) inventions in this international application, as follows:

1. claims: 1-7,9

   Cooling system with a first cooling cycle and a second cooling cycle, wherein the second cooling cycle comprises a refrigeration system

   ---

2. claim: 8

   Cooling system with a first cooling cycle and a second cooling cycle, wherein the second cooling cycle comprises a chilled water system

   ---

A. CLASSIFICATION OF SUBJECT MATTER
IPC 7 F25B25/00

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

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
IPC 7 F25B

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

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

EPO-Internal

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category °	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	US 5 400 615 A (PEARSON ET AL) 28 March 1995 (1995-03-28) column 3, line 3 - column 4, line 24; figure 1 -----	1-7,9
X	US 2003/182949 A1 (CARR RICHARD P) 2 October 2003 (2003-10-02) paragraph [0007] - paragraph [0022]; figures 1,2 -----	1-5,7,9
X	US 2002/184908 A1 (BROTZ FRIEDRICH ET AL) 12 December 2002 (2002-12-12) paragraph [0023] - paragraph [0053]; figures 1-12 -----	1-4,7,9
X	US 2003/126872 A1 (HARANO HIDEKI ET AL) 10 July 2003 (2003-07-10) paragraph [0017] - paragraph [0052]; figures 1-3 -----	1-4,6,7, 9

☐ Further documents are listed in the continuation of box C.

☒ Patent family members are listed in annex.

° Special categories of cited documents :

"A" document defining the general state of the art which is not considered to be of particular relevance

"E" earlier document but published on or after the international filing date

"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)

"O" document referring to an oral disclosure, use, exhibition or other means

"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone

"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art.

"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
23 March 2005	30. 06. 2005

Name and mailing address of the ISA	Authorized officer
European Patent Office, P.B. 5818 Patentlaan 2 NL - 2280 HV Rijswijk Tel. (+31-70) 340-2040, Tx. 31 651 epo nl, Fax: (+31-70) 340-3016	Szilagyi, B

2

Form PCT/ISA/210 (second sheet) (January 2004)

Patent document cited in search report		Publication date	Patent family member(s)		Publication date
US 5400615	A	28-03-1995	DE	4224896 A1	04-02-1993
			FR	2679985 A1	05-02-1993
			GB	2258298 A ,B	03-02-1993
US 2003182949	A1	02-10-2003	AU	2003223371 A1	13-10-2003
			WO	03083385 A1	09-10-2003
US 2002184908	A1	12-12-2002	DE	10128164 A1	12-12-2002
			EP	1264715 A2	11-12-2002
			JP	2002370527 A	24-12-2002
US 2003126872	A1	10-07-2003	JP	3594583 B2	02-12-2004
			JP	2003203905 A	18-07-2003
			TW	221644 B	01-10-2004

From the
INTERNATIONAL SEARCHING AUTHORITY

To:

see form PCT/ISA/220

PCT

WRITTEN OPINION OF THE INTERNATIONAL SEARCHING AUTHORITY

(PCT Rule 43*bis*.1)

Date of mailing (*day/month/year*) see form PCT/ISA/210 (second sheet)	
Applicant's or agent's file reference see form PCT/ISA/220	**FOR FURTHER ACTION** See paragraph 2 below

International application No.	International filing date (*day/month/year*)	Priority date (*day/month/year*)
PCT/US2004/040407	02.12.2004	05.12.2003

International Patent Classification (IPC) or both national classification and IPC
F25B25/00

DOCKETED

Applicant
LIEBERT CORPORATION

☐Non-Final+ ☐Final Action*
A WRITTEN OPINION OF THE ISA - 10/5/05
+Statutory Resp. Due ____
*Notice of Appeal Due ____
Dkt'd By CN. Ck'd By ____

1. This opinion contains indications relating to the following items:

- ☒ Box No. I Basis of the opinion
- ☒ Box No. II Priority
- ☐ Box No. III Non-establishment of opinion with regard to novelty, inventive step and industrial applicability
- ☒ Box No. IV Lack of unity of invention
- ☒ Box No. V Reasoned statement under Rule 43*bis*.1(a)(i) with regard to novelty, inventive step or industrial applicability; citations and explanations supporting such statement
- ☐ Box No. VI Certain documents cited
- ☐ Box No. VII Certain defects in the international application
- ☐ Box No. VIII Certain observations on the international application

2. **FURTHER ACTION**

If a demand for international preliminary examination is made, this opinion will usually be considered to be a written opinion of the International Preliminary Examining Authority ("IPEA"). However, this does not apply where the applicant chooses an Authority other than this one to be the IPEA and the chosen IPEA has notifed the International Bureau under Rule 66.1*bis*(b) that written opinions of this International Searching Authority will not be so considered.

If this opinion is, as provided above, considered to be a written opinion of the IPEA, the applicant is invited to submit to the IPEA a written reply together, where appropriate, with amendments, before the expiration of three months from the date of mailing of Form PCT/ISA/220 or before the expiration of 22 months from the priority date, whichever expires later.

For further options, see Form PCT/ISA/220.

3. For further details, see notes to Form PCT/ISA/220.

Name and mailing address of the ISA:	Authorized Officer
European Patent Office D-80298 Munich Tel. +49 89 2399 - 0 Tx: 523656 epmu d Fax: +49 89 2399 - 4465	Szilagyi, B Telephone No. +49 89 2399-7157

Box No. I Basis of the opinion

1. With regard to the **language**, this opinion has been established on the basis of the international application in the language in which it was filed, unless otherwise indicated under this item.

   ☐ This opinion has been established on the basis of a translation from the original language into the following language , which is the language of a translation furnished for the purposes of international search (under Rules 12.3 and 23.1(b)).

2. With regard to any **nucleotide and/or amino acid sequence** disclosed in the international application and necessary to the claimed invention, this opinion has been established on the basis of:

   a. type of material:

   ☐ a sequence listing

   ☐ table(s) related to the sequence listing

   b. format of material:

   ☐ in written format

   ☐ in computer readable form

   c. time of filing/furnishing:

   ☐ contained in the international application as filed.

   ☐ filed together with the international application in computer readable form.

   ☐ furnished subsequently to this Authority for the purposes of search.

3. ☐ In addition, in the case that more than one version or copy of a sequence listing and/or table relating thereto has been filed or furnished, the required statements that the information in the subsequent or additional copies is identical to that in the application as filed or does not go beyond the application as filed, as appropriate, were furnished.

4. Additional comments:

Box No. II Priority

1. ☒ The validity of the priority claim has not been considered because the International Searching Authority does not have in its possession a copy of the earlier application whose priority has been claimed or, where required, a translation of that earlier application. This opinion has nevertheless been established on the assumption that the relevant date (Rules 43*bis*.1 and 64.1) is the claimed priority date.

2. ☐ This opinion has been established as if no priority had been claimed due to the fact that the priority claim has been found invalid (Rules 43*bis*.1 and 64.1). Thus for the purposes of this opinion, the international filing date indicated above is considered to be the relevant date.

3. Additional observations, if necessary:

Box No. IV Lack of unity of invention

1. ☒ In response to the invitation (Form PCT/ISA/206) to pay additional fees, the applicant has:

   ☐ paid additional fees.

   ☐ paid additional fees under protest.

   ☒ not paid additional fees.

2. ☐ This Authority found that the requirement of unity of invention is not complied with and chose not to invite the applicant to pay additional fees.

3. This Authority considers that the requirement of unity of invention in accordance with Rule 13.1, 13.2 and 13.3 is

   ☐ complied with

   ☒ not complied with for the following reasons:

   see separate sheet

4. Consequently, this report has been established in respect of the following parts of the international application:

   ☐ all parts.

   ☒ the parts relating to claims Nos. 1-7,9

Box No. V Reasoned statement under Rule 43*bis*.1(a)(i) with regard to novelty, inventive step or industrial applicability; citations and explanations supporting such statement

1. Statement

Novelty (N)	Yes:	Claims	
	No:	Claims	1-7,9
Inventive step (IS)	Yes:	Claims	
	No:	Claims	1-7,9
Industrial applicability (IA)	Yes:	Claims	1-7,9
	No:	Claims	

2. Citations and explanations

   see separate sheet

WRITTEN OPINION OF THE INTERNATIONAL SEARCHING AUTHORITY (SEPARATE SHEET)

International application No.

PCT/US2004/040407

Re Item IV.

This Authority considers that there are 2 inventions covered by the claims indicated as follows:

I: Claims 1-7,9 directed to a cooling system with a first cooling cycle and a second cooling cycle, wherein the second cooling cycle comprises a refrigeration system.
II: Claim 8 directed to a cooling system with a first cooling cycle and a second cooling cycle, wherein the second cooling cycle comprises a chilled water system.

The reasons for which the inventions are not so linked as to form a single general inventive concept, as required by Rule 13.1 PCT, has been summarized in the extra sheet of form PCT/ISA/206.

Re Item V.

Objections concerning the first invention (claims 1-7,9)

1 Reference is made to the following documents:
D1 : US 5 400 615 A
D2 : US 2003/182949 A1
D3 : US 2002/184908 A1
D4 : US 2003/126872 A1

2.1 The present application does not meet the criteria of Article 33(1) PCT, because the subject-matter of claim 1 is not new in the sense of Article 33(2) PCT.
Document D1 discloses (the references in parentheses applying to this document) a cooling system for transferring heat from a heat load to a heat exchange system, the cooing system comprising:
- a volatile working fluid (see column 3,lines 52-55);
- a pump (28);
- a first heat exchanger (30) in fluid communication with the pump (28) and in thermal communication with the heat load; and
- a second heat exchanger (26) having a first fluid path in fluid communication with

the first heat exchanger (30) and the pump (28), and a second fluid path connected to the heat exchange system (14), the first and second fluid paths being in thermal communication with one other.

2.2 For the sake of completeness, it is pointed out that the subject-matter of claim 1 is also not new over the disclosure of at least one of the documents D2-D4.

2.3 Claims 6 and 9 comprise all the features of claim 1 and are therefore not appropriately formulated as claims dependent on the latter (Rule 6.4 PCT).

The additional features of claims 6 and 9 are disclosed both in document D1 and D4 and therefore, said claims do not meet the criteria of Article 33(1) PCT for lack of novelty of their subject-matter.

2.4 The features of dependent claims 2-5 and 7 can be found for example in documents D1 or D2 and thus, said claims are not new in the sense of Article 33(2) PCT.

3 Contrary to the requirements of Rule 5.1(a)(ii) PCT, the relevant background art disclosed in the documents D1 and D2 is not mentioned in the description, nor are these document identified therein.

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Bryn-Jacobsen, Caelia
Kilburn & Strode LLP
20 Red Lion Street
London WC1R 4PJ
GRANDE BRETAGNE

Date
26.01.11

Reference	Application No./Patent No.
P40722EP5	10183253.3 - 2301
Applicant/Proprietor	
Liebert Corporation	

Communication

The extended European search report is enclosed.

The extended European search report includes, pursuant to Rule 62 EPC, the European search report (R. 61 EPC) or the partial European search report/ declaration of no search (R. 63 EPC) and the European search opinion.

Copies of documents cited in the European search report are attached.

☑ 1 additional set(s) of copies of such documents is (are) enclosed as well.

The following have been approved:

☑ Abstract ☑ Title

☐ The Abstract was modified and the definitive text is attached to this communication.

The following figure(s) will be published together with the abstract: none

Refund of search fee

If applicable under Article 9 Rules relating to fees, a separate communication from the Receiving Section on the refund of the search fee will be sent later.

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EPO Form 1507N 08.10

Europäisches Patentamt
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Office européen des brevets

EUROPEAN SEARCH REPORT

Application Number
EP 10 18 3253

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (IPC)
X	S.F.PEARSON: "Development of Improved Secondary Refrigerants", THE PROCEEDINGS OF THE INSTITUTE OF REFRIGERATION, vol. 89, 4 March 1993 (1993-03-04), pages 65-80, XP002615227, * page 65 - page 80 *	1-17	INV. F25B25/00
A	1996 ASHRAE HANDBOOK: "Heating, Ventilating and Air-Conditioning SYSTEMS AND EQUIPMENT", 1996, ASHRAE, Atlanta, XP002615228, * page 12.11 - page 12.12 *	1-17	
X	US 2003/182949 A1 (CARR RICHARD P) 2 October 2003 (2003-10-02) * paragraph [0007] - paragraph [0022]; figures 1,2 *	1-17	
X	US 5 400 615 A (PEARSON ET AL) 28 March 1995 (1995-03-28) * column 3, line 3 - column 4, line 24; figure 1 *	1-17	TECHNICAL FIELDS SEARCHED (IPC)
			F25B
A	EP 0 281 762 A2 (TAKENAKA KOMUTEN CO [JP]; SHINKO KOGYO KK [JP]) 14 September 1988 (1988-09-14) * column 2, line 47 - column 3, line 36 * * column 5, line 3 - column 10, line 45; figure 1 *	1	
A	US 4 344 296 A (STAPLES JACK W ET AL) 17 August 1982 (1982-08-17) * column 2, line 33 - column 3, line 40; figures 1,2 *	1-17	

The present search report has been drawn up for all claims

1

Place of search	Date of completion of the search	Examiner
Munich	11 January 2011	Szilagyi, Barnabas

CATEGORY OF CITED DOCUMENTS

X : particularly relevant if taken alone
Y : particularly relevant if combined with another document of the same category
A : technological background
O : non-written disclosure
P : intermediate document

T : theory or principle underlying the invention
E : earlier patent document, but published on, or after the filing date
D : document cited in the application
L : document cited for other reasons

& : member of the same patent family, corresponding document

EPO FORM 1503 03.82 (P04C01)

ANNEX TO THE EUROPEAN SEARCH REPORT ON EUROPEAN PATENT APPLICATION NO. EP 10 18 3253

This annex lists the patent family membersrelating to the patent documents cited in the above-mentioned European search report.
The members are as contained in the European Patent Office EDP file on
The European Patent Office is in no way liable for these particulars which are merely given for the purpose of information.

11-01-2011

Patent document cited in search report	Publication date	Patent family member(s)	Publication date
US 2003182949 A1	02-10-2003	AU 2003223371 A1	13-10-2003
		WO 03083385 A1	09-10-2003
US 5400615 A	28-03-1995	DE 4224896 A1	04-02-1993
		FR 2679985 A1	05-02-1993
		GB 2258298 A	03-02-1993
EP 0281762 A2	14-09-1988	AU 599760 B2	26-07-1990
		AU 1274688 A	15-09-1988
		CA 1295129 C	04-02-1992
		CN 88101287 A	21-09-1988
		DE 3871995 D1	23-07-1992
		DE 3871995 T2	28-01-1993
		ES 2033348 T3	16-03-1993
		MX 167565 B	30-03-1993
		US 4843832 A	04-07-1989
US 4344296 A	17-08-1982	NONE	

EPO FORM P0459

For more details about this annex : see Official Journal of the European Patent Office, No. 12/82

The examination is being carried out on the application documents as originally filed.

1 The applicant is herewith invited to indicate the passages of the parent application as filed (04812840.9) on which present claims 1 to 17 are based.

2 In order to speed up the procedure, however, the applicant is informed that the application documents as originally filed introduce subject-matter which extends beyond the content of the parent application 04812840.9 as filed, contrary to Article 76(1) EPC. The subject-matter concerned is the following:

2.1 Claim 1 claims a "control system (100) controlling the cooling system". As the passage on page 2, last line merely states that "the disclosed cooling system also includes a control system 100" and no further basis can be found in the originally-filed application documents of the parent application for said feature, claim 1 contravenes Article 76(1) EPC.
It is clear from the application documents of the parent application that the control system is merely used for controlling the pump (cf. Figs. 1, 2 and the description page 3, lines 9 to 11)

2.2 Claim 1 also claims a cooling system comprising a first cooling cycle and a second cooling cycle in combination with features relating to controlling the cooling system. As said features relating to controlling the cooling system were originally disclosed in combination with a chilled water system or a vapour compression system (cf. page 7, lines 20 to 24) the present broader disclosure also covers embodiments wich have not been disclosed in the original application documents of the parent application, contrary to Article 76(1) EPC.

2.3 Moreover, the features of dependent claims 2-17 represent a mere summary of certain combinations of features disclosed in the the parent application as originally filed. Said claims claim features picked out from the description without maintaining the disclosed combination of features and thus, infringe Article 76(1) EPC.

2.4 For the time being the substantive examination is suspended.

2.5 When filing amended claims to overcome the above objections, care must be taken not to add subject-matter which extends beyond the content of the application as originally filed (Article 123(2) EPC).

File: 04 812-840.9

Annex

1996 ASHRAE HANDBOOK

Heating, Ventilating, and Air-Conditioning SYSTEMS AND EQUIPMENT

SI Edition

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American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.
1791 Tullie Circle, N.E., Atlanta, GA 30329
(404) 636-8400

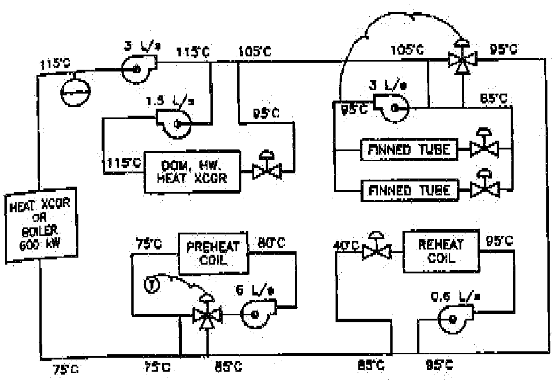

Fig. 20 Example of Series-Connected Loading

low flow rate and high temperature drop, while a lower temperature and conventional temperature drop can be used in the secondary circuit(s).

For example, a system could be designed with primary-secondary pumping in which the supply temperature from the boiler was 115°C, the supply temperature in the secondary was 95°C, and the return temperature was 85°C. This design results in a conventional 10 K Δt in the secondary zones, but permits the primary circuit to be sized on the basis of a 30 K drop. This primary-secondary pumping arrangement is most advantageous with terminal units such as convectors and finned radiation, which are generally unsuited for small flow rate design.

Many types of terminal heat transfer units are being designed to use smaller flow rates with temperature drops up to 55 K in low-temperature systems and up to 85 K in medium-temperature systems. Fan apparatus, the heat transfer surface used for air heating in fan systems, and water-to-water heat exchangers are most adaptable to such design.

A fourth technique is to put certain loads in series utilizing a combination of control valves and compound pumping (Figure 20). In the system illustrated, the capacity of the boiler or heat exchanger is 600 kW, and each of the four loads is 150 kW. Under design conditions, the system is designed for a 40 K water temperature drop, and the loads each provide 10 K of the total Δt. The loads in these systems, as well as the smaller or simpler systems in residential or commercial applications, can be connected in a direct-return or a reverse-return piping system. The different features of each load are as follows:

1. The domestic hot water heat exchanger has a two-way valve and is thus arranged for variable flow (while the main distribution circuit provides constant flow for the boiler circuit).
2. The finned-tube radiation circuit is a 10 K Δt circuit with the design entering water temperature reduced to and controlled at 95°C.
3. The reheat coil circuit takes a 55 K temperature drop for a very low flow rate.
4. The preheat coil circuit provides constant flow through the coil to keep it from freezing.

When loads such as water-to-air heating coils in LTW systems are valve controlled (flow varies), they have a heating characteristic of flow versus capacity as shown in Figure 21 for 10 K and 30 K temperature drops. For a 10 K Δt coil, 50% flow provides approximately 90% capacity; valve control will tend to be unstable. For this reason, proportional temperature control is required, and equal percentage characteristic two-way valves should be selected such that 10% flow is achieved with 50% valve lift. This combination of the valve characteristic and the heat transfer characteristic of the coil

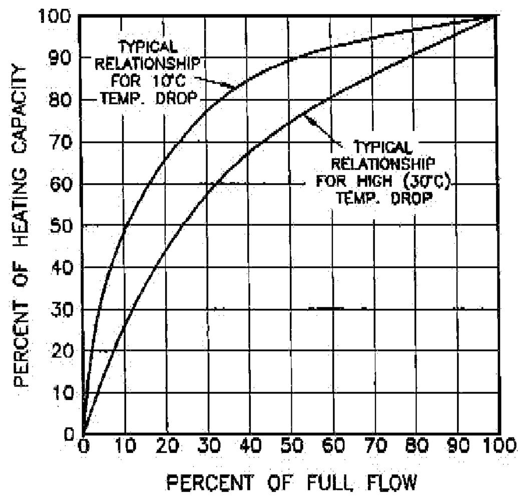

Fig. 21 Heat Emission Versus Flow Characteristic of Typical Hot Water Heating Coil

makes the control linear with respect to the control signal. This type of control can be obtained only with equal percentage two-way valves and can be further enhanced if piped with a secondary pump arrangement as shown in Figure 19A. See Chapter 42 of the 1995 *ASHRAE Handbook—Applications* for further information on automatic controls.

CHILLED WATER SYSTEMS

Designers have less latitude in selecting supply water temperatures for cooling applications because there is only a narrow range of water temperatures low enough to provide adequate dehumidification and high enough to avoid chiller freeze-up. Circulated water quantities can be reduced by selecting proper air quantities and heat transfer surface at the terminals. Terminals suited for a 6 K rise rather than a 4 K rise reduce circulated water quantity and pump power by one-third and increase chiller efficiency.

A proposed system should be evaluated for the desired balance between installation cost and operating cost. Table 1 shows the effect of coil circuiting and chilled water temperature on water flow and temperature rise. The coil rows, fin spacing, air-side performance, and cost are identical for all selections. Morabito (1960) showed how such changes in coil circuiting affect the overall system. Considering the investment cost of piping and insulation versus the operating cost of refrigeration and pumping motors, higher temperature rises, (i.e., 9 to 13 K temperature rise

Table 1 Chilled Water Coil Performance

Coil Circuiting	Chilled Water Inlet Temp., °C	Coil Pressure Drop, kPa	Chilled Water Flow, mL/(s·kW)	Chilled Water Temp. Rise,
Full[a]	7.2	6.9	39	6.1
Half[b]	7.2	37.9	30	8.3
Full[a]	4.5	3.4	25	9.5
Half[b]	4.5	17.2	20	12.1

Note. Table is based on cooling air from 27°C dry bulb, 19°C wet bulb to 14°C dry bulb, 13°C wet bulb

[a]Full circuiting (also called single circuit). Water at the inlet temperature flows simultaneously through all tubes in a plane transverse to airflow: it then flows simultaneously through all tubes, in unison, in successive planes (i.e., rows) of the coil.

[b]Half circuiting. Tube connections are arranged so there are half as many circuits as there are tubes in each plane (row) thereby using higher water velocities through the tubes. This circuiting is used with small water quantities.

at about 18 to 27 mL/s per kilowatt of cooling capacity) can be applied on chilled water systems with long distribution piping runs; larger flow rates should be used only where reasonable in close-coupled systems.

For the most economical design, the minimum flow rate to each terminal heat exchanger is calculated. For example, if one terminal can be designed for a 10 K rise, another for 8 K, and others for 7 K, the highest rise to each terminal should be used, rather than designing the system for an overall temperature rise based on the smallest capability.

The control system selected also influences the design water flow. For systems using multiple terminal units, diversity factors can be applied to flow quantities before sizing pump and piping mains if exposure or use prevents the unit design loads from occurring simultaneously and if two-way valves are used for water flow control. If air-side control (e.g, face-and-bypass or fan cycling) or three-way valves on the water side are used, diversity should not be a consideration in pump and piping design, although it should be considered in the chiller selection.

A primary consideration with chilled water system design is the control of the source systems at reduced loads. The constraints on the temperature parameters are (1) a water freezing temperature of 0°C, (2) economics of the refrigeration system in generating chilled water, and (3) the dew-point temperature of the air at nominal indoor comfort conditions (13°C dew point at 24°C and 50% rh). These parameters have led to the common practice of designing for a supply chilled water temperature of about 7°C and a return water temperature between 13 and 18°C.

Historically, most chilled water systems have used three-way control valves to achieve constant water flow through the chillers. However, as systems have become larger, as designers have turned to multiple chillers for reliability and controllability, and as energy economics have become an increasing concern, the use of two-way valves and source pumps for the chillers has greatly increased.

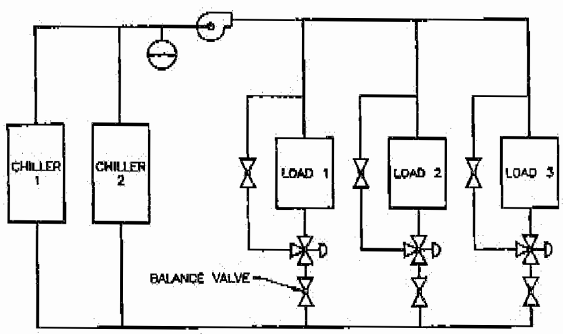

Fig. 22 Constant Flow Chilled Water System

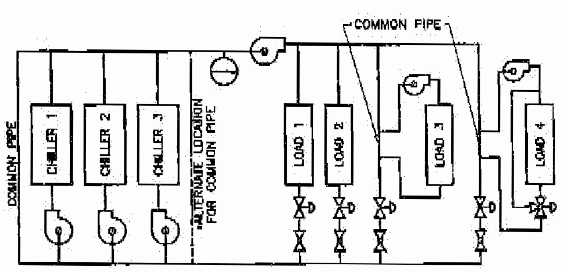

Fig. 23 Variable Flow Chilled Water System

A typical configuration of a small chilled water system using two parallel chillers and loads with three-way valves is illustrated in Figure 22. Note that the flow is essentially constant. A simple energy balance [Equation (9)] dictates that with a constant flow rate, at one-half of design load, the water temperature differential drops to one-half of design. At this load, if one of the chillers is turned off, the return water circulating through the off chiller mixes with the supply water. This mixing raises the temperature of the supply chilled water and can cause a loss of control if the designer does not consider this operating mode.

A typical configuration of a large chilled water system with multiple chillers and loads and compound piping is shown in Figure 23. This system provides variable flow, essentially constant supply temperature chilled water, multiple chillers, more stable two-way control valves, and the advantage of adding chilled water storage with little additional complexity.

One design issue illustrated in Figure 23 is the placement of the common pipe for the chillers. With the common pipe as shown, the chillers will unload from left to right. With the common pipe in the alternate location shown, the chillers will unload equally in proportion to their capacity (i.e., equal percentage).

DUAL-TEMPERATURE SYSTEMS

Dual-temperature systems are used when the same load devices and distribution systems are used for both heating and cooling (e.g., fan-coil units and central station air-handling unit coils). In the design of dual-temperature systems, the cooling cycle design usually dictates the requirements of the load heat exchangers and distribution systems. Dual-temperature systems are basically of three different configurations, each requiring different design techniques:

1. Two-pipe systems
2. Four-pipe common load systems
3. Four-pipe independent load systems

Two-Pipe Systems

In a two-pipe system, the load devices and the distribution system circulate chilled water when cooling is required and hot water when heating is required (Figure 24). Design considerations for these systems include the following:

- Loads must all require cooling or heating coincidentally; that is, if cooling is required for some loads and heating for other loads at a given time, this type of system should not be used.
- When designing the system, the flow and temperature requirements for both the cooling and the heating media must be calculated first. The load and distribution system should be designed for the more stringent, and the water temperatures and temperature differential should be dictated by the other mode.

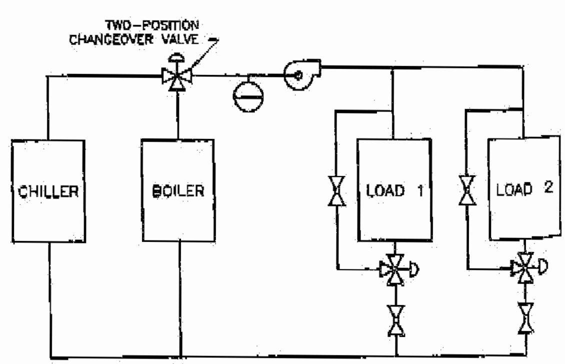

Fig. 24 Simplified Diagram of Two-Pipe System

Annex to 04 812 840

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Development of Improved Secondary Refrigerants

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(Session 1992–1993)

Presented before the Institute of Refrigeration at The Institute of Marine Engineers, 76 Mark Lane, London, E.C.3, on 4 March 1993, at 5.45 p.m.

Introduction

The replacement of environmentally damaging halocarbon refrigerants by less harmful, but more expensive, substitutes and the return to toxic but equally effective refrigerants such as ammonia makes it relevant to examine the use of what are commonly called secondary refrigerants. The official term for such substances is "heat transferring liquids" but that term is not in common use.

In the early days of refrigeration, secondary refrigerants were more commonly used than at present. The advent of non-toxic halocarbon refrigerants in the 1930s resulted in a move away from the use of secondary refrigerants. In many cases it became convenient to distribute small refrigerating systems to locations where refrigeration was required rather than to install a central refrigerating plant and distribute the refrigeration by circulation of a secondary refrigerant. The disadvantage of secondary refrigerating systems is that an additional temperature difference is required thus reducing the efficiency of the process and that, the transmission of heat being by sensible heat only, large masses of secondary refrigerant have to be circulated. This in turn increases the pumping power required and needs a large and expensive piping system to distribute the secondary refrigerant in the required quantities.

Few refrigerating systems extract heat entirely by the evaporation of a volatile refrigerant. Most use air as the medium to extract heat from the product and transfer it to the refrigerant evaporator. Water is also commonly used as a secondary refrigerant in large air conditioning systems because it is cheap, non-toxic and easily pumped.

All secondary refrigerants have their advantages and disadvantages. The object of this paper is to indicate ways in which the performance of secondary refrigerants can be enhanced.

Desirable Properties

The ideal secondary refrigerant would be a dense, non-toxic liquid having a high thermal conductivity, a high specific heat and a low viscosity. It is also important that the secondary refrigerant should not freeze or evaporate within its range of use and that it should be compatible with common engineering materials.

For the efficient transfer of heat from a circulating liquid it is desirable that the flow should be turbulent. Fully turbulent flow can be expected in tubes of circular cross-section at Reynolds numbers exceeding 10,000. At Reynolds numbers below about 2,100, the flow can be expected to be streamline. Under streamline flow conditions there is no mixing of the liquid with the result that heat transfer from tube wall to the bulk liquid is by conduction only. This makes for very poor heat transfer. In the transition region between Reynolds numbers of 2,100 and 10,000 some mixing of the liquid will take place with consequent improved heat transfer. Despite the very poor heat transfer characteristics associated with streamline flow it is by no means uncommon to encounter streamline flow when using certain antifreeze solutions at low temperatures. This is a measure of the need to find improved secondary refrigerants.

Comparison of Conventional Liquid Secondary Refrigerants

Nomenclature

j = Product of dimensionless groups	$\frac{h}{C_p \rho V} \cdot \left(\frac{C_p \mu}{k}\right)^{2/3} \cdot \left(\frac{\mu_w}{\mu}\right)^{0.14}$
h = Coefficient of heat transfer	W/m² K
h_B = Base value of h	W/m² K
C_p = Specific heat at constant pressure	kJ/kg K
ρ = Density	kg/m³
V = Velocity	m/hour
μ = Viscosity	kg/hr.m
μ_w = Viscosity of fluid at tube wall temp.	kg/hr.m
k = Thermal conductivity	W/m.K

The mechanism of fluid flow within tubes was first explained by Osborne Reynolds[1] who observed that at a characteristic dimensionless ratio, now called the Reynolds number, the dynamic forces due to the motion of the fluid overcame viscous forces thus causing the intense mixing which is known as turbulence. The presence or absence of turbulence has a marked effect on the transmission of heat from tube wall to fluid. The rules governing heat transfer in completely streamline flow and in turbulent flow were relatively easy to work out. However, there remained a region between Reynolds numbers of 2,100 and 10,000 where the flow was in a state of transition between streamline and turbulent. It was also observed that heat transfer at Reynolds numbers below 2,100 depended to a considerable extent on the length to diameter ratio of the tube in which flow was taking place. This was because of the time taken to establish completely streamline flow after the disturbing effects at entry to the tube. A comprehensive relationship of heat transfer coefficient to Reynolds number taking account of the effects of the tube length to diameter ratio in the streamline and transitional regions was published by Sieder and Tate of California in 1936[2]. Curves showing the Seider and Tate relationships are given in Fig. 1. From such curves it is possible to calculate an expected coefficient of heat transfer provided the properties of the fluid, the velocity and the tube dimensions are known.

The Seider and Tate relationship can be expressed in the form;

$$j = \frac{h}{C_p \rho V} \cdot \left(\frac{C_p \mu}{k}\right)^{2/3} \cdot \left(\frac{\mu_w}{\mu}\right)^{0.14}$$

The inclusion of a term consisting of the ratio of the fluid viscosity at wall temperature and the fluid viscosity at mean temperature allows the relationship to be used both for heating and cooling. Because of the relatively low power to which the viscosity ratio is raised, it has only a minor effect in turbulent flow. In streamline and transitional flow, the viscosity ratio is not negligible but depends on the heat flux for each particular heat transfer process being studied. It is possible to produce general curves for heat transfer coefficients associated with fluids at various Reynolds numbers, but the effect of the viscosity ratio must be added as a correction factor when the approximate viscosity ratio has been established. Other, and more complex correlations of heat transfer coefficient with Reynolds number have been proposed subsequent to Seider and Tate, but the Seider and Tate correlation is adequate for the purposes of this paper. It should be remembered that experimentally determined heat transfer coefficients, even under carefully controlled conditions show a considerable degree of scatter from the mean. It follows that coefficients of heat transfer obtained in practice will display a similar scatter about the calculated values.

There are many liquids which could be used as secondary refrigerants. A comparison has been made between liquids ranging in cost from water to silicone fluid. In general the substances selected are of low toxicity and are not environmentally damaging. Substances such as R11 (CCl_3F) and methylene chloride (CH_2Cl_2), which are perfectly adequate secondary refrigerants, have not been included because of their environmentally damaging effect. Pressurised liquid CO_2 has been included because of its low toxicity and superior heat transferring properties. It is also included for comparison because there is a possibility that liquid CO_2 can be used as a volatile secondary refrigerant resulting in greatly reduced flow rates per kW of heat transfer[3].

Tables 1, 2 and 3 show some of the relevant properties of the 11 selected secondary refrigerants at temperatures of +5 °C, −10 °C and −40 °C respectively.

Some of the secondary refrigerants are solutions in water. The strength of the solution has been selected to give a margin of safety against freezing. Apart from that, it is desirable to use as weak a solution as possible because water is cheap and a very good secondary refrigerant.

In general, boiling points are given for the pure substance though in practice, particularly with the glycols, it is normal to add inhibitors in sufficient quantities to change the boiling points slightly. Diethylbenzene is a mixture of diethylbenzene isomers and therefore does not have a sharply defined freezing point.

It should be noted from the tables that water is eliminated from tables two and three and that sodium chloride solution is eliminated from table three for the

TABLE 1
(5 °C)

No.	Substance	Solution by wt. %	Viscosity Centipoise	Freezing point °C	Boiling point °C	l/sec per 100 kW for 5K temp. rise	Relative cost per unit vol.
1	Water	100	1.55	0.0	100.0	4.78	0.017
2	NaCl	12	1.75	−8.0	101.7	5.05	1.00
3	$CaCl_2$	12	2.01	−7.2	100.6	5.23	0.98
4	Methanol	15	2.5	−10.3	86.1	4.89	0.69
5	Ethanol	20	4.2	−11.1	87.2	4.66	4.47
6	Ethylene glycol	25	2.7	−10.6	102.8	5.07	3.10
7	Propylene glycol	30	5.0	−10.6	102.2	4.94	7.26
8	Trichloroethylene	100	0.7	−87.0	86.7	13.30	29.43
9	Diethylbenzene	100	1.2	−73.0	181.0	13.30	154.73
10	Liquid CO_2	100	0.09	−57.0	−78.45*	8.23	20.00
11	Polydimethylsiloxane	100	1.91	−111.0	175.0	14.50	300.00

*CO_2 sublimes from the solid at −78.45 °C at atmospheric pressure.

TABLE 2
(−10 °C)

No.	Substance	Solution by wt. %	Viscosity Centipoise	Freezing point °C	Boiling point °C	l/sec per 100 kW for 5K temp. rise	Relative cost per unit vol.
2	NaCl	21	4.12	−17.2	102.2	4.1	1.87
3	$CaCl_2$	20	4.80	−17.2	101.1	5.5	1.77
4	Methanol	22	5.40	−15.3	83.3	5.1	1.00
5	Ethanol	25	8.40	−15.3	86.1	4.7	5.63
6	Ethylene glycol	35	7.00	−18.0	116.0	5.4	4.42
7	Propylene glycol	40	23.00	−20.0	115.3	5.2	9.83
8	Trichloroethylene	100	0.80	−87.0	86.7	13.6	29.90
9	Diethylbenzene	100	1.50	−73.0	181.0	13.1	153.00
10	Liquid CO_2	100	0.12	−57.0	−78.45*	9.0	22.50
11	Polydimethylsiloxane	100	2.50	−111.0	175.0	14.6	300.00

*CO_2 sublimes from the solid at −78.45 °C at atmospheric pressure.

TABLE 3
(−40 °C)

No.	Substance	Solution by wt. %	Viscosity Centipoise	Freezing point °C	Boiling point °C	l/sec per 100 kW for 5K temp. rise	Relative cost per unit vol.
3	$CaCl_2$	30	36.00	−50.0	102.2	5.6	2.92
4	Methanol	50	22.50	−54.0	76.0	6.7	2.20
5	Ethanol	55	23.50	−51.0	79.0	6.4	12.00
6	Ethylene glycol	60	350.00	−45.5	120.0	6.8	7.90
7	Propylene glycol	60	1500.00	−47.0	118.0	5.8	15.00
8	Trichloroethylene	100	1.10	−87.0	86.7	14.8	30.50
9	Diethylbenzene	100	2.80	−73.0	181.0	13.2	157.00
10	Liquid CO_2	100	0.21	−57.0	−78.45*	9.3	25.60
11	Polydimethylsiloxane	100	4.80	−111.0	175.0	14.9	300.00

*CO_2 sublimes from the solid at −78.45 °C at atmospheric pressure.

same reason, namely that it is not a liquid at the specified temperature.

Figures 2, 3 and 4 are plots of heat transfer coefficients in W/m^2 K against velocity in metres/second for the selected secondary refrigerants at the chosen temperatures of +5 °C, −10 °C and −40 °C. The plots are for the flow of liquid in a tube of 25 mm internal diameter and having a length to diameter ratio of 100. The effect of other length to diameter ratios between the streamline and transitional regions can be estimated from Fig. 1. Diameter correction values for turbulent flow and for streamline flow are given in Tables 4 and 5. The plots for individual refrigerants are identified by number. The numbers are given in Tables 1, 2 and 3.

This method of displaying the relationship between heat transfer coefficient and velocity is based on the presentation of Stoever[4]. It is a simple and convenient method of obtaining approximate heat transfer coefficients for those who have neither time nor inclination to work them out from first principles. As previously explained, the plots require to be corrected for viscosity ratio in the transitional and streamline flow regions. This cannot be done till an approximate heat transfer coefficient has been worked out. In practice it will be found that constant heat transfer coefficients can be assumed in the turbulent region but that in the streamline flow region the heat transfer coefficient will vary along the tube as the bulk temperature of the liquid changes. Solution of heat transfer problems involving streamline flow is therefore an iterative process both at individual points to establish the viscosity ratio and at intervals along the tube to establish the fluid temperature and the associated viscosity. The disadvantages of the streamline flow regime include poor heat transfer coefficients, high pressure drops and complex calculations!

Individual Performance

Water (1)

Figure 2 shows water to be the most effective of the selected refrigerants at 5 °C. Water of course is not a contender at −10 °C and −40 °C.

It will be noted from Fig. 2 that water enters the transitional region at velocities below about 0.6 m/sec. Water is cheap, non toxic and has very good heat transfer and transport properties. In the absence of air, water is not corrosive although corrosion inhibitors are sometimes added, particularly if the water is being used at higher temperatures.

Sodium Chloride Solution (2)

Figures 2 and 3 show that sodium chloride solution gives relatively high coefficients of heat transfer though it enters the transitional region at low velocities. Sodium chloride is rarely used in UK because it is highly corrosive. It should be employed in a sealed system with inhibitors and should be checked for pH on a regular basis. Sodium chloride solutions are occasionally used because they are cheap and non toxic, but the relatively high eutectic point of −21 °C is also a disadvantage.

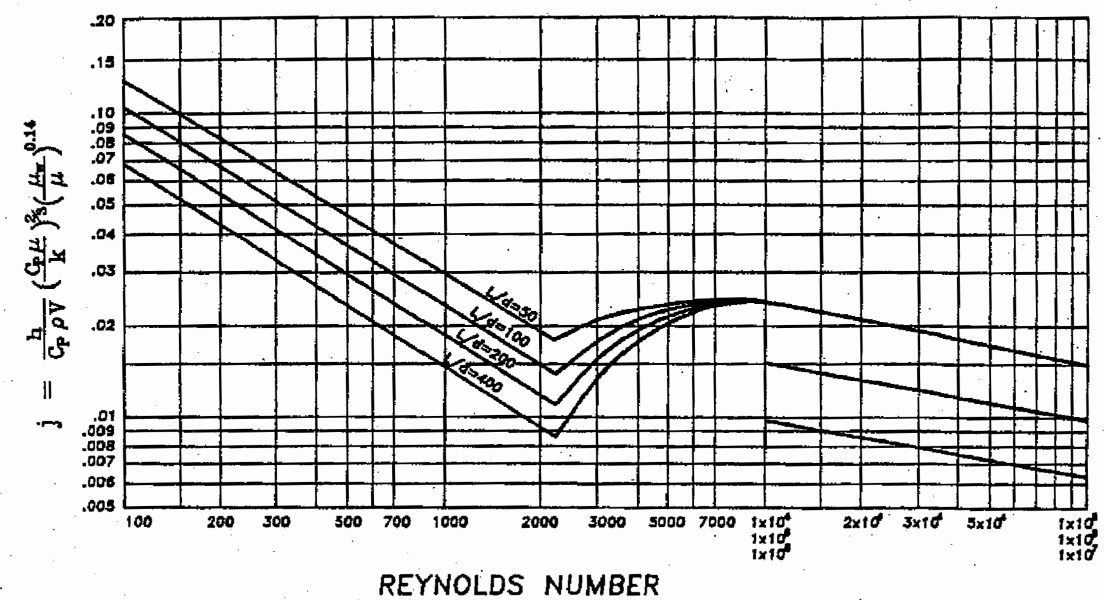


Fig. 1.

Calcium Chloride Solution (3)

Calcium chloride solution is less corrosive than sodium chloride solution though those who have worked with it might find that difficult to believe. Air in the vicinity of exposed calcium chloride is highly corrosive to untreated steel. At + 5 °C and − 10 °C calcium chloride solutions give marginally higher coefficients of heat transfer than are obtained with sodium chloride solution, but both enter the transitional region at low velocities. At − 40 °C, the use of calcium chloride solution tends to result in streamline flow. Although the eutectic temperature of calcium chloride solutions is very much lower than − 40 °C, it is difficult to apply such solutions, even at − 40 °C because the freezing point is very markedly dependent on concentration in this region. Calcium chloride is deliquescent and strong solutions will tend to absorb water, thus diluting themselves. Calcium chloride brine is unpleasant, toxic and corrosive to skin, leather and natural fabrics. It should be used in systems where its exposure to air is at a minimum. Systems which are self venting to a relatively small header tank with a cover are generally satisfactory though fully sealed and pressurised systems can be used. Potassium dichromate inhibitor is usually added and the pH of the solution is kept slightly alkaline by the addition of sodium hydroxide. Sealed pressurised systems are not necessarily a good idea because they increase the pressure difference at pump and valve seals and they make it more difficult to vent the system and control the pH. Sealed systems are very beneficial for pressurised hot water systems where they reduce the risk of cavitation at the pump. They provide no such benefits when using low temperature secondary refrigerants.

Methanol (4)

Methanol solutions can be used for temperatures down to − 40 °C. They give reasonably good heat transfer coefficients and are not highly viscous. Methanol is both toxic and flammable. It should thus be used in sealed systems with caution. The use of methanol is mainly confined to industrial applications. Because of its toxicity it should not be used in the presence of food stuffs. Over exposure to methanol can permanently damage the optic nerve leading to blindness.

Ethanol (5)

Ethanol is a slightly poorer secondary refrigerant than methanol but it is less toxic. Ethanol is industrial ethyl alcohol and as such as is liable both to misuse and to excise duty unless it is carefully controlled in defined areas. When used as a secondary refrigerant ethanol is usually in the form of methylated spirit containing some methyl alcohol and other substances to make it unpleasant. Ethanol vapour is narcotic, dense and flammable. In general these disadvantages are sufficient to make ethanol unattractive as a secondary refrigerant.

Ethylene Glycol (6)

In general glycols are less toxic but more viscous than the alcohols used as secondary refrigerants. Ethylene glycol has properties very similar to methanol at + 5 °C and gives heat transfer coefficients similar but slightly lower than methanol at − 10 °C. At − 40 °C there is little to chose between them and they are generally unsatisfactory. Ethylene glycol is a sweet tasting liquid which is mildly toxic, hygroscopic and corro-

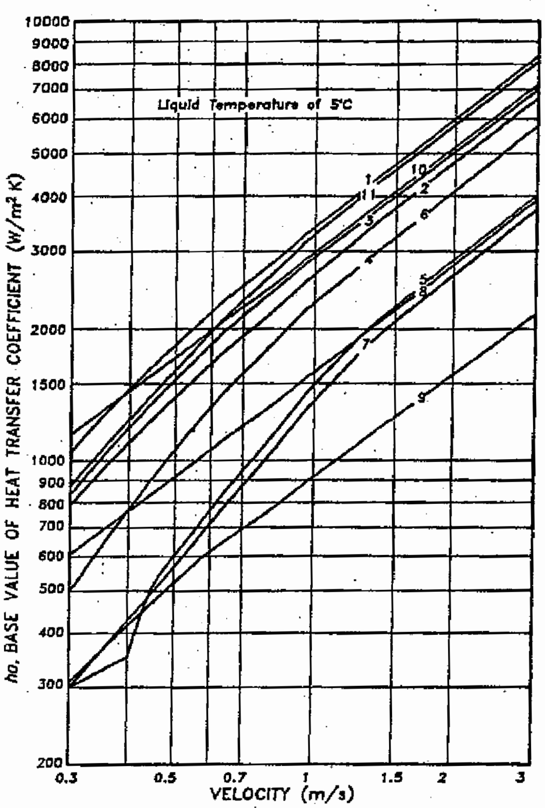


Fig. 2.

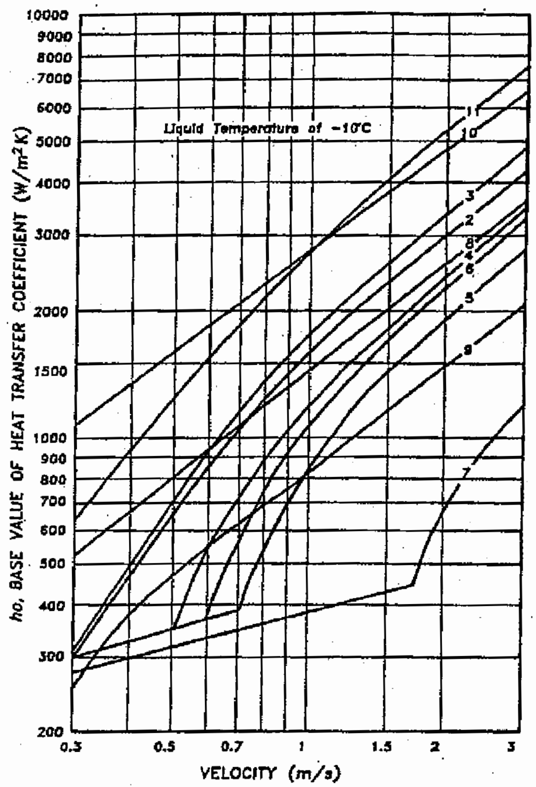


Fig. 3.

TABLE 4
Diameter correction values (*Fd*) for the turbulent flow of liquids inside circular tubes

	$h = ho \times Fd$							
Inside tube diameter (mm)	6	12	18	25	38	50	75	100
Fd	1.34	1.16	1.06	1.00	0.92	0.87	0.80	0.76

TABLE 5
Diameter correction values (*Fd*) for the streamline flow of liquids inside circular tubes

	$h = ho \times Fd$							
Inside tube diameter (mm)	6	12	18	25	38	50	75	100
Fd	1.6	1.30	1.10	1.00	0.87	0.79	0.69	0.63

sive. It is sufficiently non-toxic to have been dishonestly added to certain categories of wine to improve apparent quality. Ethylene glycol is usually employed with corrosion inhibitors and is not generally used by the food industry despite its low toxicity.

Propylene Glycol (7)

Propylene glycol is substantially non-toxic and in the appropriate grade can be a registered food additive. This makes it very popular with consultants having a background in food science, especially if they don't know much about heat transfer or refrigeration. Propylene glycol is also generally used with anti corrosion additives but these can also be non-toxic. The major problem with propylene glycol arises from its very high viscosity. This gives rise to high pumping powers and the onset of laminar flow at relatively low velocities. Even at −10 °C, propylene glycol solutions are in the streamline flow region for most of the velocities covered by figures 2 to 4. At −40 °C the viscosity of propylene glycol is so high that pumping losses would be unacceptable.

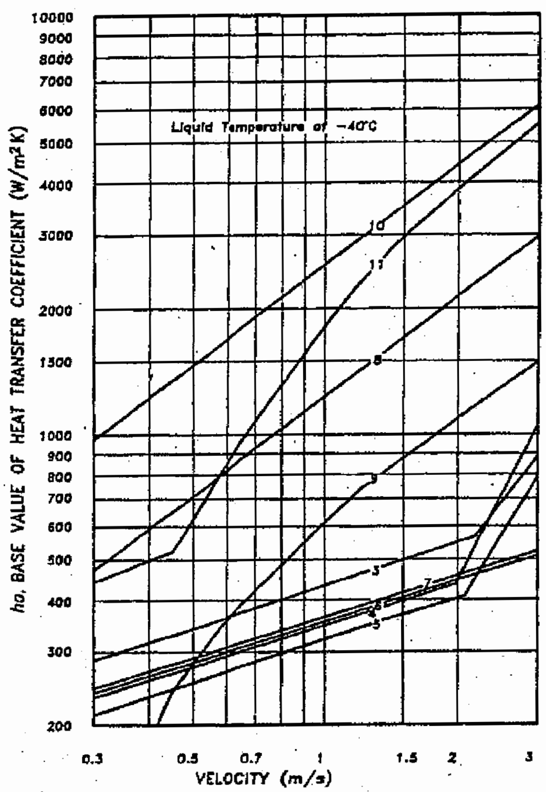


Fig. 4.

Trichloroethylene (8)

Trichloro-ethylene is a dense colourless liquid of very low viscosity giving very good heat transfer coefficients at low temperatures. Trichloroethylene has a low vapour pressure and is only slightly toxic. Trichloroethylene has anaesthetic properties and at one stage was in common use as a self administered anaesthetic and analgesic during child birth. Trichloroethylene has a strong smell and has narcotic effects. It was formerly used as a solvent but its use was discontinued in favour of substances which are even less toxic. However, there may be a return to the use of trichloro-ethylene both as a solvent and as a vapour degreaser because the double bond between the carbon atoms in the trichloroethylene molecule makes it sufficiently unstable to have a zero ozone depleting potential. Trichloroethylene is relatively safe to use in closed systems and is perhaps the best available secondary refrigerant for low temperature industrial uses at present. Its use cannot be recommended in conjunction with foodstuffs because of its strong smell and the possibility of contamination. At 5 °C and −10 °C there are other secondary refrigerants giving better heat transfer coefficients than trichloroethylene. At −40 °C trichloroethylene is perhaps the most suitable conventional low temperature secondary refrigerant. The main disadvantages of trichloroethylene are its smell, its low specific heat and its low thermal conductivity. These disadvantages are to some extent offset by its high density and very low viscosity.

Diethyl Benzene (9)

Diethyl benzene is inferior to trichloroethylene as a secondary refrigerant under practically all conditions and it has an even more offensive smell. The main advantage of diethyl benzene is that it can be used from a range of about −70 °C to about +175 °C. It is thus particularly suitable for use in industrial and chemical processes where heat exchange takes place at both high and low temperatures. As can be seen from Figs. 2, 3 and 4, diethyl benzene is inferior to most of the other secondary refrigerants at +5 °C and −10 °C but becomes significantly better than them for turbulent flow at −40 °C.

Liquid CO_2 (10)

Liquid CO_2 is not at present used as a secondary refrigerant because of the need to pressurise it to keep it in the liquid phase. It is here included because there is now a need for a safe non-toxic secondary refrigerant giving high heat transfer coefficients throughout a wide range of operating temperatures. CO_2 is non-toxic in food or drink and small leakages would evaporate without contaminating product or putting operatives at risk. CO_2 is characterised by good specific heat and extremely low viscosity. Its worst feature from the heat transfer point of view is its low thermal conductivity although this is less important in turbulent flow. At +5 °C the heat transfer coefficients experienced with liquid CO_2 are similar to those associated with water and with sodium chloride and calcium chloride brines. At −10 °C the performance of liquid CO_2 is significantly better than all the other secondary refrigerants while at −40 °C it is about twice as good as its nearest rival, trichloroethylene, which in turn is about twice as good as diethyl benzene. The only disadvantage of liquid carbon dioxide is the pressures at which it must operate. These could be in the region of 40 bar which is outside the range of pressures experienced even in the high pressure sides of conventional refrigerating systems. However, modern hydraulic systems work at even greater pressures with the result that piping, fittings and flexible hoses are available to operate at CO_2 pressures with a satisfactory margin of safety (see Appendix 1). The main reason why liquid CO_2 is unlikely to be accepted as a conventional secondary refrigerant is that its use has been proposed as a volatile secondary refrigerant in a recent patent application[3], and this application of CO_2 would be even more effective.

Polymethylsiloxane (11)

Polydimethylsiloxane is a transparent, non-toxic silicone liquid with a very low freezing point. In common with other silicone fluids, it is very stable and of very low toxicity. It is therefore suitable for use in medical applications and for the cooling of food, though it is not a registered food additive. The main disadvantage of polydimethylsiloxane is its high price.

Corrugated Plate Heat Exchangers

The main concern of this paper is with the development of improved secondary refrigerants but methods of getting improved performance from existing secondary refrigerants should not be neglected. One of the most significant advances in the past decade has been the advent of plate type heat exchangers which are suitable for primary refrigerant pressures. Such heat exchangers were the subject of a recent paper to The Institute of Refrigeration[5]. The heat exchangers may be brazed, gasketed, welded in pairs with primary refrigerant gaskets at inlet and outlet ports or totally welded but contained in a pressure vessel. Apart from its ability to cram a vast amount of heat exchange surface within a very small volume, the major advantage of the plate type heat exchanger is that the critical Reynolds number below which liquid flow is streamline can be in the region of 100 to 400. This should be compared with the conventional critical Reynolds number of 2,100 for flow within circular tubes. Plate type heat exchangers are now coming into common use both as refrigerant evaporators and as refrigerant condensers. Where relatively viscous secondary refrigerants are used, the plate type heat exchanger has enormous advantages over shell and tube heat exchangers.

Performance of plate type heat exchangers as condensers or as evaporators with pumped refrigerant flow appears to be reasonably well established. Manufacturer's predictions of performance of plate type heat exchangers operating as natural circulation boilers should be treated with considerably more caution. For pump circulation plate type evaporators the best performance will be associated with plate designs having low chevron angles to the horizontal. Such plates may not be suitable for natural circulation because the static head required to overcome the pressure drop within the plates may require an impracticably high accumulator producing undesirable static head effects at plate inlet and undesirable pressure drops in the wet return line.

The high efficiency of plate type heat exchangers makes them vulnerable to the effects of surface fouling. This can be of particular importance in ammonia refrigerating systems where the oil is not miscible with the refrigerant. The choice of lubricant and the arrangement of the system are of considerable importance if heat transfer is to be maximised.

Improvement of Apparent Specific Heat

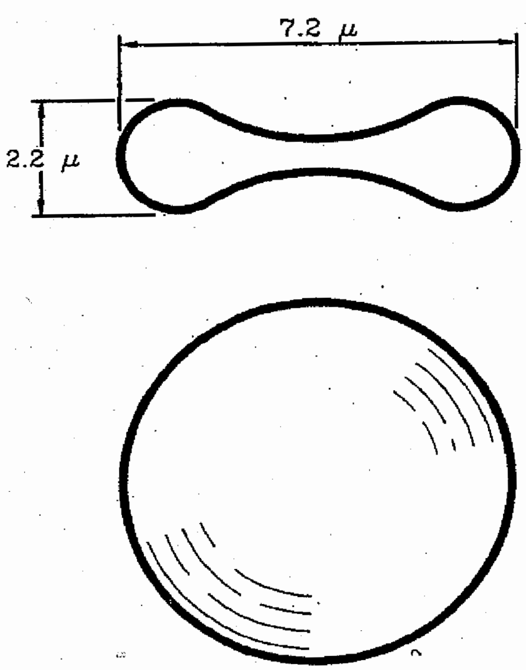
One of the disadvantages of the use of liquid secondary refrigerants is that the transmission of heat is by sensible heat alone. This predicates the circulation of large quantities of liquid which must be heated and cooled through significant temperature ranges. If it were possible to circulate frozen particles with the secondary refrigerant, the refrigerant mass flow could be much reduced if the particles could be induced to absorb heat by melting at the appropriate point of the circuit. A very recent paper[6] by Dr Ing Joachim Paul discusses the use of pumpable ice slurries in this fashion. If one assumes a 5 °C temperature raise in the circulating water and a circulation which contains 50% by weight of ice crystals, it can be calculated that the mass flow can be reduced by a factor of 7 or 8 times provided all the ice crystals are melted during the circulation. This is a major advantage in terms both of pumping power required and of pipeline diameter. The savings obtained by the circulation of a pumpable ice slurry are probably sufficient to justify the cost and complexity associated with producing the slurry and circulating it. In cases where the heat load is at a considerable distance from the refrigerating system as in long tunnels or deep shafts, the circulation of particulate frozen material may be the only safe and economic way of transferring the heat.

It remains to be seen whether the circulation of ice slurries will be generally accepted despite the complexities and costs involved in their production and circulation. The use of ice slurry is restricted to heat exchange at temperatures around 0 °C.

Circulation of Frozen Particles

In 1965 a patent was granted to Bronsverk[7] for the circulation of brine containing small spherical vessels of liquid which it was intended to freeze and thaw during the process of circulation. As far as I know, nothing came of this patent, possibly because the spheres envisaged were of a size which was not conducive to rapid freezing and thawing. A sphere is not the ideal shape for rapid freezing and thawing if the complete sphere is to be frozen and thawed. Spherical objects contain the maximum amount of material within the minimum possible surface. They are also unlikely to promote good transfer of heat when carried in a stream of liquid in which they have more or less neutral buoyancy. Rapid freezing and thawing requires a shape which will "tumble" to promote good heat transfer while at the same time being robust while having a large surface area. The problem is similar to that which was solved by the evolution of the red blood corpuscle. The function of the red blood corpuscle is to absorb and transport oxygen and carbon dioxide molecules. The oxygen must be absorbed rapidly through as large a surface as possible, but the corpuscle must be of a robust form to withstand repeated pumping through the heart. Figure 5 shows the shape of a typical mammalian red blood corpuscle. It can be assumed that this is the best shape to combine robustness with surface area and well distributed volume. The genetically transmitted disorder known as sickle-cell anaemia causes the production of red blood corpuscles in the form of curved cylinders which appear under the microscope to be sickle shaped. The cylinder or rod shape is at least superior to the sphere in terms of surface to volume ratio but the lack of surface compared to the normal corpuscle gives rise to severe anaemia in those afflicted by this disorder. The only reason for the survival of populations with sickle-cell anaemia is that the condition provides a sufficient degree of immunity to malarial infection to allow those suffering from it to reach reproductive age in parts of the world where malarial infection is a major threat.

In March 1990 a patent[8] was applied for, covering a secondary refrigerating system designed to circulate small frozen particles having the shape but being much larger than red blood corpuscles. Studies were undertaken at Strathclyde University to determine the freezing times and volumes of such corpuscles. Figures 6 and 7 show the temperature distribution within a partially frozen and an almost completely frozen 3.5 mm diameter corpuscle. Calculations indicate that the freezing time of such a corpuscle in turbulent flow within a liquid of temperature 10 K colder than the freezing temperature of the corpuscle would be about three seconds. Further work needs to be done on the hydraulic transport properties of corpuscles and on the economics of their manufacture. The use of a sealed corpuscle removes the restriction to temperatures in the region of 0 °C. Corpuscles containing 100% acetic or lactic acid would freeze and thaw at about 15 °C.



RED BLOOD CORPUSCLE

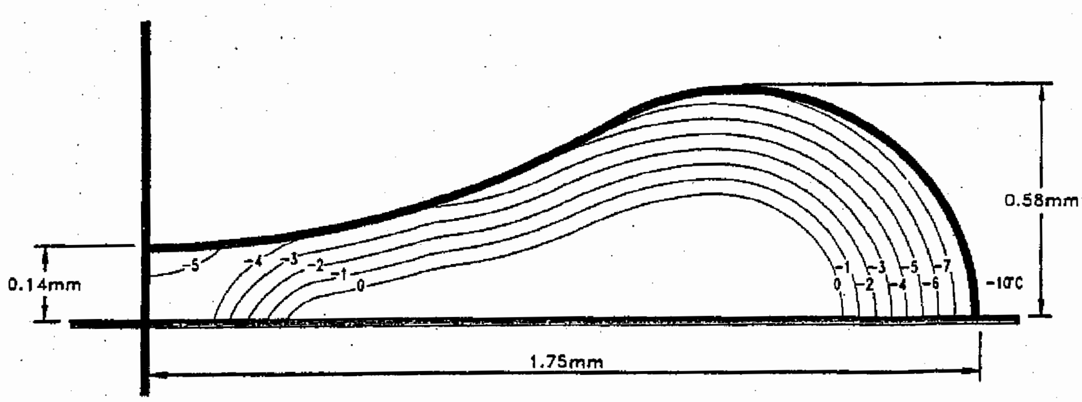
Fig. 5.

Corpuscle circulation can be used to reduce refrigerant mass flow and pumping power or it can be used as a form of thermal storage. A particular benefit would be in its use to allow capacity increases in systems which were limited by the dimensions of interconnecting pipework. This is a problem which must arise frequently in chemical plants.

Much work remains to be done on the corpuscle project which has been neglected of late because a better way presented itself, namely the use of CO_2 as a volatile secondary refrigerant.

Volatile Secondary Refrigerants

Carbon dioxide has many advantages as a refrigerant. It is non toxic compared even to the best of the halocarbons, it has excellent heat transfer properties, it has zero ozone depleting potential and a global warming potential lower than that of halocarbons by a factor of several thousand. The major practical disadvantage of carbon dioxide is its low critical temperature of 31.1 °C. Such a low critical temperature means that carbon dioxide cannot be used efficiently at normal condensing temperatures. The concept of a cascade refrigerating system using two refrigerants where the evaporator of the high temperature refrigerant serves as the condenser for the low temperature refrigerant is well known. Cascade refrigerating systems are in common use for the achievement of low temperatures using refrigerant pairs like R13 and R22. The R13 refrigerant remains at sufficiently high pressure in its evaporator to be handled by conventional reciprocating compressors, but it does not reach very high pressures in its condenser because the condenser is refrigerated to say −20 °C by the high stage R22 system. Cascade refrigerating systems allow relatively conventional equipment to be used to obtain temperatures which could not otherwise be achieved using a single refrigerant. It would be possible, though rather pointless, to overcome the low critical pressure of carbon dioxide by using it as the low temperature refrigerant in a cascade system using ammonia or R22



CORPUSCLE ISOTHERMS DURING FREEZING
ELAPSED TIME: 1.5 SEC.

Fig. 6.

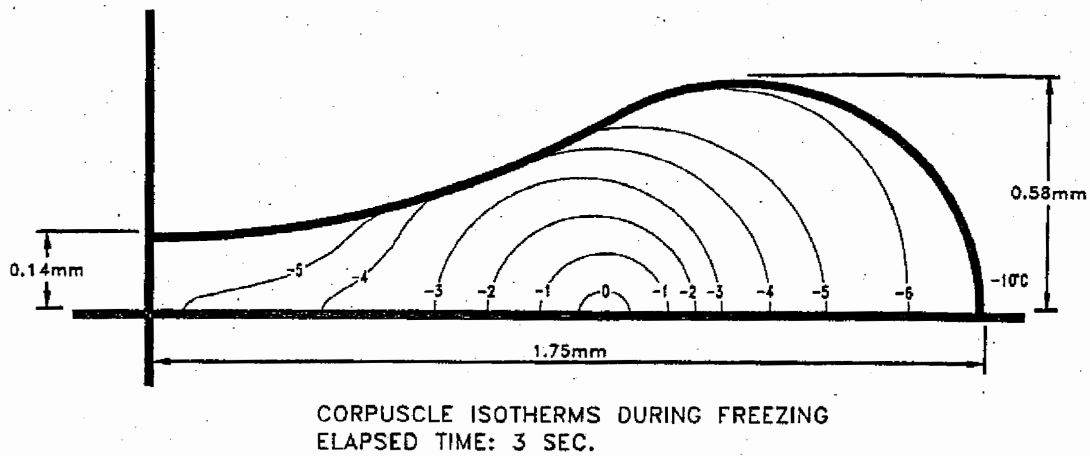


Fig. 7.

as the high temperature refrigerant. Such a system would be practical but would involve the use of a positive displacement compressor for the carbon dioxide. Such compressors are available but are not attractive for refrigeration because of the high pressures and costs associated with them. Carbon dioxide is not miscible with normal lubricants. This is not an insuperable obstacle, but it makes the idea of redeveloping refrigeration compressors for use on CO_2 rather unattractive.

It is not necessary to compress carbon dioxide vapour to use it in the low temperature stage of a cascade system. It is perfectly feasible to circulate liquefied carbon dioxide to an evaporator where it changes to vapour which is drawn, still at high pressure, to an appropriately designed condenser which is the evaporator of the high temperature refrigerating system. From the condenser the reliquefied CO_2 is circulated by pump or gravity back to the evaporator. Figure 8 indicates a simplified version of such a circuit. An additional temperature difference is required to liquefy the carbon dioxide vapour, but this temperature difference need be no greater than the temperature difference required to cool a liquid secondary refrigerant. An obvious advantage of the use of liquefied carbon dioxide as a volatile secondary refrigerant compared to its use in the low temperature stage of a cascade system is that it need not be contaminated with compressor lubricating oil.

The allowable exposure limit of carbon dioxide is 5,000 ppm which is five times the allowable exposure limit of R12. However, carbon dioxide under certain circumstances can be dangerous because at high concentrations it can interfere with the breathing reflex and cause death. Carbon dioxide detectors should be fitted in any space where there is a danger of a significant concentration of carbon dioxide arising. It would also be prudent to make provision for forced air ventilation in the event of excess carbon dioxide being detected. That being said, the hazard presented by carbon dioxide in confined spaces is no greater than

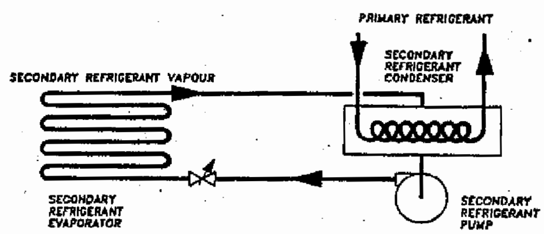


Fig. 8.

the hazard which is presented by high concentrations of halocarbon refrigerants.

Another disadvantage of carbon dioxide is the high pressures under which it must be kept to maintain it in liquid form. At 15 °C, the pressure of saturated carbon dioxide is of the order of 50 bar gauge. Such pressures are significantly greater than those experienced when using halocarbon refrigerants or ammonia. However, carbon dioxide has been widely used as a refrigerant in the past at a time when high quality pipe and fittings were not so readily available as they are now. Modern developments in hydraulic systems have resulted in the availability of relatively cheap high quality pipe, flexible pipe and fittings for pressures up to and exceeding 100 bar. The use of liquid carbon dioxide as a secondary refrigerant has been made even easier by the fact that the pipeline sizes required when using carbon dioxide are significantly smaller than the pipe sizes required to use conventional primary or secondary refrigerants. This greatly reduces the installed cost of the carbon dioxide system and leads to very high factors of safety if standard stainless steel piping is used.

It would require 2½ inch diameter pipes to circulate enough water to carry away 100 kW of heat assuming that the water were heated through 5K. To extract the same amount of heat using carbon dioxide as a volatile secondary refrigerant would require a supply pipe of about ½ inch diameter and a return pipe for the vapour of about 1 inch diameter. In practice one would possibly not use such small diameter supply pipe, but it would be possible to do so. The small size of carbon dioxide compressors and the small diameter of carbon dioxide piping were always a source of wonder to refrigerating engineers whose experience had been restricted to ammonia.

Limits of Application for Carbon Dioxide

The triple point of carbon dioxide is −57 °C at a pressure of about 4.2 bar gauge. The critical temperature of carbon dioxide is 31.1 °C at a pressure of about 73 bar gauge. There would appear to be no reason why carbon dioxide should not be used as a secondary refrigerant at temperatures ranging from about −50 °C to about +10 °C. This temperature range covers most applications of conventional refrigeration. The saturation pressure at −50 °C is about 5.8 bar gauge. The saturation pressure at +10 °C is about 44 bar gauge.

Possible Applications of CO_2 as a Volatile Secondary Refrigerant

Liquefied CO_2 can be used for low temperature cold storage or blast freezing although this should only be done if there are compelling reasons why direct expansion ammonia cannot be used. The cooling of air to low temperatures requires defrosting of the air coolers. A method of hot gas defrosting of air coolers using CO_2 vapour has been proposed. CO_2 is circulated to the expansion valves at the air coolers by means of a pump. Liquid CO_2 is also pumped by a high pressure pump into a liquid storage volume with a liquid vapour interface which is kept above 0 °C saturated. When defrosting of the CO_2 refrigerated air cooler is required, primary refrigerant is condensed in coils submerged in the high pressure liquid CO_2. At the same time valves are opened to pass warm high pressure carbon dioxide vapour to the cooler requiring to be defrosted. By this means, a type of defrosting equivalent to conventional hot gas defrosting can be obtained. This method of defrosting is illustrated in Fig. 9.

Carbon dioxide can also be used in chill stores and for air conditioning where it is desired to avoid large charges of halocarbon refrigerant. In chill stores it would be possible to use ammonia as the high temperature refrigerant and liquefied CO_2 as the low temperature secondary thus producing a refrigerating system which was completely harmless to the environment without suffering the penalties of using a viscous secondary refrigerant like propylene glycol.

For most types of air conditioning it is possible to use water as the secondary refrigerant. A water chilling package using ammonia as the primary refrigerant could be designed to have a very small charge and would be environmentally benign. However, the use of carbon dioxide as a volatile secondary would greatly reduce the cost and complexity of the piping within the building. Significant space could be saved by the elimination of large diameter water pipes. In the event of small leaks, the carbon dioxide would not damage the building structure as water would do. It is possible that in a high building the liquid carbon dioxide could be circulated to the coolers by gravity, thus eliminating the need for circulating pumps. Overall, it is expected that building air conditioning using small diameter

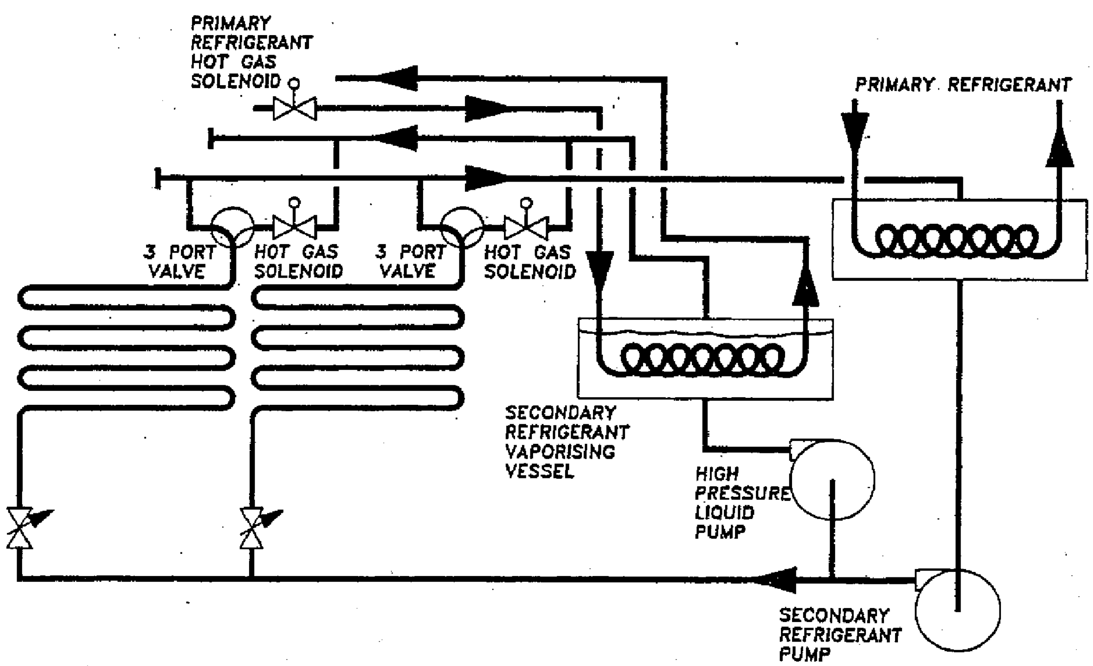


Fig. 9.

pipes carrying liquid CO_2 would be both cheaper and more efficient than using water or glycol as a secondary refrigerant. The use of ammonia and carbon dioxide would be much less damaging to the environment than the use of VRV systems with significant halocarbon charge.

Supermarkets

Supermarkets are very significant users of refrigeration. Supermarkets almost invariably use direct expansion of halocarbon refrigerants, although some are experimenting with the use of secondary refrigerants. Circulation of liquefied CO_2 as a volatile secondary refrigerant is particularly suitable for supermarket refrigeration, not only because the system is efficient and non polluting, but because it is easy and clean to alter cabinet runs and the position of display cases. It is very difficult to move refrigerated fittings without losing some halocarbon refrigerant to the atmosphere. CO_2 cases can be isolated and vented to atmosphere without causing any pollution or oil contamination.

The ultimate environmentally friendly refrigerating system would seem to be an ammonia primary refrigerating system having minimum charge and a liquefied CO_2 system as the secondary. It would, of course, be possible to use a halocarbon refrigerant in the primary circuit, but the advent of high pressure plate type heat exchangers makes it possible to use ammonia in small quantities to produce large amounts of refrigeration.

Practical Implications of using CO_2 as a Volatile Seconday Refrigerant

The acoustic velocity of CO_2 vapour is in the region of 250 m/sec. Text books suggest that dry suction lines of similar size are required for equivalent CO_2 and ammonia refrigerating systems, giving capacities about 3 times greater than would be achieved in R12 or R22 suction lines. The acoustic velocity of ammonia vapour is about twice that of CO_2 vapour but ammonia is usually pump circulated to large evaporators at a rate several times greater than would be required for dry expansion. It is unlikely that such high circulation rates would be required for an oil-free CO_2 system. Velocities of up to 25 m/sec have been employed in dry suction lines of CO_2 systems.

Large Ammonia Cold Store

A large low temperature ammonia cold store has recently been updated by the replacement of valve stations and site piping which had become severely corroded.

The refrigerating duty is 1,600 kW at −35 °C evaporating temperature. The ammonia liquid pumps were originally selected for a 5:1 overfeed ratio.

The wet suction line replacing the original suction lines is 12 inches outside diameter collecting from two 8 inch nominal bore headers in the cold store and running 75 metres to the engine room. The headers in the cold store also ran for 75 metres in each direction with diameters reducing progressively to 5 inch. The 75 metre liquid feed from the engine room is 4 inch pipe, splitting into two 3 inch headers at the cold store. Assuming the same duty, but using CO_2 with a liquid velocity of 1.5 m/sec and a suction vapour velocity of 25 m/sec, the appropriate CO_2 pipe sizes work out at 2½ inch nominal bore and 4 inch nominal bore. Calculation of pressure drop in 75 metres of 4 inch bore pipe indicated that the pressure drop was about 0.75 bar. This seems high in terms of conventional refrigeration, but because of the high pressures at which CO_2 operates, it amounts to only about 1 °C.

Twelve inch pipe probably costs about £300/m to support, install and insulate. The equivalent cost of 4 inch pipe would be about £100/m. There would be comparable savings in the distribution headers, interconnecting piping and valve stations. It can be argued that, as well as the safety benefits of reducing the ammonia charge by a factor of at least 100 and confining it to a sealed engine room, the cost reductions and performance improvements resulting from the use of CO_2 as a volatile secondary refrigerant would make it attractive. It is perhaps worth noting that 600 gallons of oil were removed from the system when it was updated. CO_2 systems as here proposed would be oil-free. The reduction in pipe size obtained by using CO_2 as a volatile secondary refrigeration is so great that part of the saving should be invested in using stainless steel pipe which could be Schedule 40 rather than the Schedule 80 mild steel pipe which was traditionally used for CO_2 systems. The pressure in the working system at −35 °C would be of the order of 11 bar gauge but hot gas defrosting would require pressures of about 50 bar gauge. Such pressures are well within the operating range of the stainless steel Schedule 40 piping and fittings.

Large Air Conditioning System on R22

A recently commissioned air conditioning installation consists of 4 separate systems with screw compressors on R22. Each system has a capacity of 850 kW. The condensers are on the roof of the multi-storey building and the water chilling units are in the basement.

Each system has a 5 inch hot gas line and a 3 inch liquid line. The lines are at least 80 metres long.

The water system has 10 inch mains, splitting to 8 inch and 6 inch to serve different parts of the building. The flow and return pipes are at least 150 metres long.

It would have been possible to site the refrigerating systems on the roof, circulating liquid CO_2 from there by pump or gravity. The evaporating temperature is +5 °C to produce chilled water at +10 °C.

The situation is different from the large ammonia cold store previously considered. The building space through which the CO_2 distribution pipes would run is of much greater value than the space in the engine room, pipe bridge and cold store yard occupied by the pipes of the previous example. The CO_2 evaporating pressure for air conditioning is also much higher, being of the order of 44 bar gauge. The density of the CO_2 vapour at this pressure is relatively high, being about 73 kg/m³. In view of this very high density it was decided to restrict the design vapour velocity to 12 m/sec to reduce inertia forces at bends and fittings which might have caused noise and vibration. However the design liquid line velocity was increased to 3 m/sec to compensate for the additional space taken up the suction line. The main suction line worked out at a 5 inch nominal bore pipe and the liquid line worked out at 4 inch nominal bore using the design velocities given previously. The reduction in space required within the building and the reduction in installed cost would appear to be significant.

It is worth noting that, when the CO_2 evaporating temperature is high, the liquid and vapour lines work out at rather similar sizes. This is because of the very high vapour density and the significant reductions of latent heat as temperatures are raised towards the critical.

Trial Installation at Marks & Spencer, Kilmarnock

A small demonstration unit has been installed at the premises of Marks & Spencer p.l.c., Kilmarnock. Marks & Spencer contributed towards the cost of the installation and the project was supervised by Oscar Faber (Scotland) Limited.

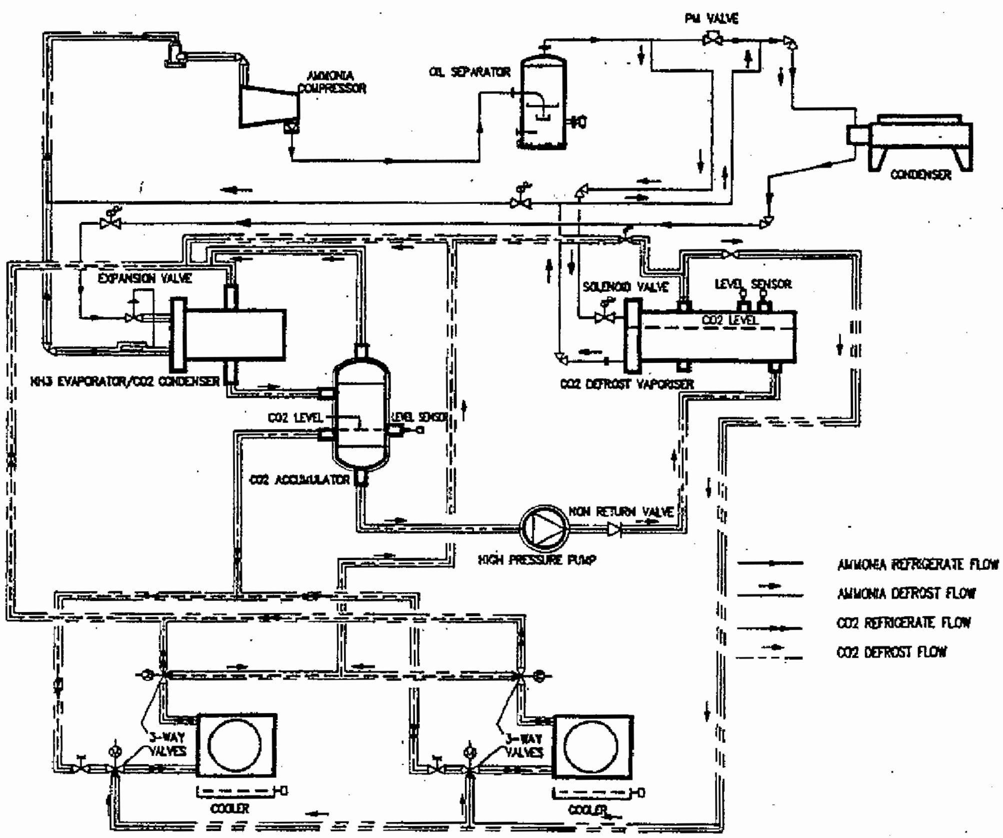
The installation is arranged to provide additional refrigeration for a small −23 °C cold store which is designed to operate on R502. A simplified schematic diagram of the installation is shown in Fig. 10.

The purpose of the installation is to gain experience of the use of carbon dioxide in this way and to demonstrate that the patented hot gas defrosting system is a practicable proposition.

The primary refrigerant is ammonia which is pumped by a Bitzer OSNA 5341-K ammonia compressor running at 1,450 rpm. The ammonia is condensed in an air cooled condenser, the high pressure liquid being fed to a Danfoss TEA20-5 thermostatic expansion valve. The ammonia evaporates in a top fed high pressure plate and shell heat exchanger manufactured by APV Baker. The ammonia vapour returns from the heat exchanger to the screw compressor. The plate type heat exchanger in which the ammonia evaporates serves as a low temperature condenser for the carbon dioxide. The condensed carbon dioxide drains to a liquid CO_2 receiver from which it flows by gravity to the inlets of two finned air coolers. The liquid CO_2 passes through expansion valves into the natural circulation air coolers and returns by natural convection to the top of the CO_2 condenser.

Defrosting

The carbon dioxide liquid receiver has a preferential feed to the suction of a high pressure liquid pump



SCHEMATIC DIAGRAM OF VOLATILE SECONDARY SYSTEM
MARKS & SPENCERS PLC – KILMARNOCK

Fig. 10.

which, under the control of a level sensor in a high pressure plate and shell defrosting vessel, will supply liquid under pressure to the vessel. When defrosting is required, high pressure ammonia vapour is circulated within the plates of the defrost vessel causing carbon dioxide to vaporise at high pressure and at a temperature of approximately 10 °C. The high pressure CO_2 vapour is fed to the air cooler being defrosted where it condenses within the tubes of the air cooler, thus melting the frost on the outside of the cooler.

At the time of writing the system has not been running long enough to draw any firm conclusions but the project looks very encouraging.

The installation is more complex than is justified for a single small cold store but it illustrates the principle that a large cold store or a large number of refrigerated display cases could be refrigerated and defrosted in a similar manner.

Conclusions

Environmental pressures will militate against the use of direct expansion refrigerating systems. Safety regulations will make it increasingly onerous to use refrigerating systems with large ammonia charges. A solution to both problems is to use secondary refrigerants but it appears that the cost of using conventional secondary refrigerants, especially at low temperatures, will be high.

The use of liquefied carbon dioxide at pressure as a volatile secondary refrigerant would appear to overcome many of the global and local problems associated with the use of conventional refrigerating systems. Liquefied carbon dioxide can be used at temperatures ranging from −50 °C to +10 °C without exceeding the pressures used in modern hydraulic systems.

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APPENDIX 1

Pressures in T316 stainless steel schedule 40 piping

Pipe size NB	Pipe size OD mm	Bore mm	Bore area mm^2	Sch. 40 wall thickness mm	Test pressure Bar G	Burst pressure Bar G	Max. working pressure Bar G	R note (5)
½"	21.34	15.8	0.196×10^3	2.77	636	1272	382	7.6
¾"	26.67	20.9	0.344×10^3	2.87	527	1055	316	6.3
1"	33.40	26.6	0.577×10^3	3.38	496	992	298	6.0
1¼"	42.16	35.0	0.964×10^3	3.56	414	828	248	5.0
1½"	48.26	40.9	1.31×10^3	3.68	374	747	224	4.5
2"	60.33	52.5	2.17×10^3	3.91	318	635	191	3.8
2½"	73.03	62.7	3.09×10^3	5.16	346	692	208	4.2
3"	88.90	77.9	4.77×10^3	5.49	303	605	182	3.6
3½"	101.60	90.1	6.39×10^3	5.74	277	554	166	3.3
4"	114.30	102.3	8.21×10^3	6.02	258	516	155	3.1
5"	141.30	128.2	12.9×10^3	6.55	227	454	136	2.7
6"	168.28	154.1	18.6×10^3	7.11	207	414	124	2.5
8"	219.08	202.7	32.3×10^3	8.18	183	366	110	2.2
10"	273.05	254.5	50.9×10^3	9.27	166	333	100	2.0
12"	323.85	304.8	73.0×10^3	9.53	144	288	87	1.7

Notes
1. Design strength values taken from BS5500: 1991, Table 2.3.
2. Test pressure is based on minimum yield stress of 245 N/mm^2.
3. Burst pressure is based on minimum tensile stress of 490 N/mm^2.
4. Maximum working pressure is based on a design stress of 147 N/mm^2.
5. R is the ratio of maximum pipe working pressure to the maximum system working pressure (arbitrarily set at 50 Bar G).

APPENDIX 2

CO_2 dry suction lines

Capacity in kW at various saturated suction temperatures*

Nominal pipe size	Suction temperature °C: −40	−20	0	10
½"	10.0	15.0	21.3	23.3
¾"	19.9	32.7	43.8	38.4
1"	37.5	59.4	82.1	91.6
1¼"	77.9	123.0	169.0	186.0
1½"	117.0	186.0	264.0	292.0
2"	225.0	354.0	493.0	545.0
2½"	353.0	551.0	776.0	859.0
3"	618.0	1020.0	1380.0	1530.0
3½"	904.0	1440.0	2030.0	2220.0
4"	1260.0	2000.0	2790.0	3070.0
5"	2340.0	3700.0	5160.0	5700.0
6"	3700.0	5860.0	8170.0	9020.0

*Capacities are based on a pressure drop of 2 °C/100 m in Schedule 40 pipe.
To calculate multipliers for other pressure drops use the expression:

$$\text{Multiplier} = \left(\frac{p'}{p}\right)^{0.535}$$

Note: the pressure drops must be in units of pressure not °C.

APPENDIX 3

CO_2 liquid lines

Capacity in kW at various temperatures*

Nominal pipe size	Liquid temperature °C: −40	−20	0	10
½"	211	171	126	100
¾"	370	300	222	176
1"	581	486	359	285
1¼"	1040	842	621	493
1½"	1410	1150	845	671
2"	2330	1890	1390	1110
2½"	3320	2700	1990	1580
3"	5130	4170	3070	2430
3½"	6860	5540	4100	3260
4"	8820	7180	5280	4200
5"	13900	11300	8307	6590
6"	20000	16300	12000	9520

*Capacities are based on a liquid velocity of 3 m/sec in Schedule 40 pipe.

Discussion report on the paper "Development of Improved Secondary Refrigerants" by Dr Forbes Pearson, B.Sc., Ph.D., A.R.C.S.T., M.I.Mech.E., Fellow. Chairman: Mr A. H. Brown.

The President submitted a written question on behalf of Mr G. Hodgins who was unable to attend. The question concerned calcium chloride brine, and whether the speaker was aware that the traditional inhibitors were now banned, due to potential contamination of foodstuffs, injury to operators, and disposal problems. He further asked whether there were any commercially available safe alternatives which were just as effective.

Dr Pearson answered that he was not aware of the ban, nor of any other inhibitors. He added that calcium brine was not to be recommended in the vicinity of foodstuffs.

Mr C. Dellino stated that 30 years ago his company had designed CO_2 liquefiers which had stood on top of liquid CO_2 vessels. He asked Dr Pearson whether he saw any great difficulties in putting the CO_2 liquefier and the storage vessel into one, and added that he was of the opinion that if the project was large enough this could be done.

Secondly, he wondered why there were expansion valves between the static down leg of the storage vessel and the cooler, and asked whether it was to maintain the pressure difference in defrost. He also wanted to know if it would be possible to use an ordinary flooded system.

He thirdly asked for what pressure the system was designed when it warmed up to ambient, and wanted more information on the design of the finned coolers.

Dr Pearson replied that there was no reason why the vessels should not be integrated.

In answer to the second question, he agreed that the system could work without expansion valves, but there would be a danger of overfeeding the cooler to such an extent that there would be a very high density mixture of liquid and vapour returning to the condenser. He felt that this would cause an unnecessary pressure drop. He added that if there were a distributor on the cooler, it would also be a good idea to have a valve ahead of the distributor.

In reply to the third question about ambient pressure, he explained that if there were a hot gas defrost one would need at least a 10c (45 bar) condensing pressure in the evaporator. He went on to say that he had used a nominal pressure of about 50 bar with the idea, on a large plant, to have a small slave compressor to keep down the pressure during standstill. He said there was no problem with pressure at low temperatures.

In relation to the fourth question, Dr Pearson said there had been no problems with regard to pressure tests, and the coolers were made of welded stainless steel.

Professor Haselden first of all stated that the papers from Dr Pearson were not only of high technical quality, but of very significant practical import.

He went on to say that the application of vaporizing CO_2 as a heat transfer medium had been applied in two ways: firstly, by the substitution of volatile CO_2 for water in air conditioning of large buildings. Professor Haselden felt this method provided potential power and capital saving.

Further, he was of the opinion that it would seem to be far more preferable to have the total ammonia system contained on the roof of the building, and would make ammonia much more acceptable as a refrigerant.

He lastly felt that a CO_2 system would not be ideal where the air conditioning installation was used as a heat pump during the winter. However, there seemed, in his view, to be a real promise of lower installation costs. Where a secondary refrigerant was introduced in place of direct evaporation there would be the need for extra power. He asked Dr Pearson to comment on the potential power penalty which would arise in that eventuality.

Dr Pearson said that he had not thought about using CO_2 in a reversed cycle system, but could not see any easy way of using it at the moment.

In terms of the substitution of pump circulated ammonia, he felt that large ammonia systems were notorious for containing a great deal of oil. He was of the opinion that not only would the CO_2 be totally clean, but it would also have the advantage of smaller temperature drops due to pressure drop in the return lines. He said the plant in the example was at 5 times overfed in order to try to get oil moving. This implied a much denser fluid returning and a much higher pressure drop.

Mr K. Dunsdon felt there was a need for better engineering qualities with the CO_2 pressures which were involved, rather than traditional pipe fitting techniques presently in use. He asked whether or not he was putting his clients at risk by putting CO_2 into systems.

Dr Pearson stated that high pressure was a perceived difficulty, but felt that CO_2 had been used historically quite satisfactorily with more primitive materials. He added that now there were even more suitable materials, although he agreed it required more care. He was of the view that it was not a technology beyond modern fitters, and felt that the standards of fitters were very high.

Mr P. Harraghy was concerned about the prevalent use of R22 on smaller refrigerant packages and the use of propylene glycol or ethylene glycol in large chilled production areas. He noted Dr Pearson's description of ethylene glycol as "sweet tasting and mildly toxic" although Figs 2 and 3 showed it had better heat transfer characteristics than propylene glycol. He asked for Dr Pearson's comments.

He said that in the past it had been common to have had large quantities of ammonia or R12 pumped through steel coils submerged in a tank of water to produce ice storage. The need to reduce primary refrigerant meant there was no longer a desire to have large quantities of primary refrigerants circulated directly through coils. There had been a push towards using small primary refrigerant systems cooling glycol, which was used to form the ice. He felt there was a need to move back towards using CO_2 and, by having two evaporators on a chilling package, one could swing from either condensing CO_2 to make the ice by subsequent evaporation or chilling water directly for circulating throughout the rest of the building.

Dr Pearson, in answer to the first point regarding propylene glycol, felt it was a very popular secondary refrigerant and would still be used. He said there were worries about product liability, particularly in the food industry, and that there had been attempts to find a non-toxic substitute. He felt that propylene glycol had a dreadful viscosity characteristic. He was of the view that calcium chloride was also impractical at low temperatures because of its toxicity and because of the very rapid change of freezing point at concentrations near the eutectic point. Dr Pearson thought that low temperature secondaries would be expensive to run and would consume a lot more power than CO_2. He did not think that CO_2 would be used in very small systems, as it would be too costly, but it would be ideal for large systems.

He went on to say that it could be used in conjunction with ice tanks and it would cut down the primary refrigerant charge enormously.

With reference to the two evaporatoıs on an ice plant, he stated that in certain cases there might be advantages. He explained that the reason why glycol had been introduced into ice bank accumulators was so they could be installed by a plumber.

Mr Ratcliffe asked whether it would be too simplistic to say that Dr Pearson's vaporizer, or hot gas generator, had been described in a paper many years ago.

Dr Pearson agreed that it could have been as there was little which was new.

Dr Tatsis felt that Dr Pearson had put forward a pragmatic solution to a real environmental problem faced by the phasing out of CFCs. He inquired whether it was possible to carry out some comparative calculations between the potential energy gains or losses in comparison to other types of secondary refrigerants. He further asked whether there was gaining or losing on the secondary global warming potential due to the running efficiencies, or otherwise, of the system. He said he was interested to know which of the three potential applications mentioned may be on the positive side. If that turned out to be the case in relation to Marks and Spencers, he asked if they, or anybody else, would be interested in a case study to establish the relative merits of the technology which had been talked about that evening.

Dr Pearson replied that in cases where people were already using a secondary, such as distribution of chilled water for air conditioners, he felt it was obvious the CO_2 would require less power. He went on to say that in cases of putting in CO_2 in place of direct expansion ammonia it would be a close run thing; but the effect of oil could not be be neglected.

Dr Pearson further stated that the installation at Marks and Spencers was not a suitable example of power saving because it was so small. It ran the smallest screw compressor at half speed and was purely a demonstration model.

Mr G. Lang asked whether there were any other commercial applications for this type of system outside of Marks and Spencers in Scotland.

Dr Pearson replied that the use of CO_2 as a volatile substance was an original idea and had not been tried anywhere before. He added that Marks and Spencers had kindly allowed them to use their store as they felt there was a need to cope with the problems arising from the phasing out of CFCs.

Mr Lang said that he had heard of other systems in Scandinavia and assumed they were totally different.

Dr Pearson described them as having a glycol chiller with the glycol being used for high temperature cabinets and as a coolant liquid for condensers to reject heat from the low temperature systems. He confirmed that the use of the screw compressor was just to make it simple.

Mr A. Pearson referred to Dr Pearson's mention of VRV in his presentation, and the use of his phrase "not long for this world". He asked for his comments on CO_2 as a pseudo VRV system with specific regard to toxicology.

Dr Pearson replied that he had used the phrase "not long for this world" because VRV depended on halocarbon refrigerants, and there would be increasing prejudice against that type of refrigerant.

In reply to the question of toxicology, he said that the halocarbon commonly used in VRV systems was R22, and that the allowable exposure level was 1,000 ppm. He added that the British and American safety standards however related not to AEL but to the amount of charge which could be suddenly released into the smallest humanly occupied space served by the system — the figure was 0.17 kg per cubic metre much higher than the AEL — but most of the larger VRV systems did not even comply with such quantities. However, the Japanese and new European standards were 0.3 kg per cubic metre. He went on to say that the British and American standards were based on cardiac sensitization, while Japan and the rest of Europe based their standards on narcotic effects which were not felt at quite such a low threshold as the cardiac sensitization.

CO_2, Dr Pearson felt, was different as ıt was in the atmosphere, at a concentration of 300 ppm and had an AEL of 5,000 ppm (0.5%) but after concentrations of about 9% were reached it would inhibit breathing. He further thought that the industry had to be aware of the effects of both halocarbons and carbon dioxide in confined spaces.

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COPY

Application No.	Ref.	Date
04 812 840.9 - 2301	P40722EP-K/C8-J	19.02.2009
Applicant Liebert Corporation		

Communication pursuant to Article 94(3) EPC

The examination of the above-identified application has revealed that it does not meet the requirements of the European Patent Convention for the reasons enclosed herewith. If the deficiencies indicated are not rectified the application may be refused pursuant to Article 97(2) EPC.

You are invited to file your observations and insofar as the deficiencies are such as to be rectifiable, to correct the indicated deficiencies within a period

of 4 months

from the notification of this communication, this period being computed in accordance with Rules 126(2) and 131(2) and (4) EPC. One set of amendments to the description, claims and drawings is to be filed within the said period on separate sheets (R. 50(1) EPC).

Failure to comply with this invitation in due time will result in the application being deemed to be withdrawn (Art. 94(4) EPC).

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Szilagyi, Barnabas
Primary Examiner
For the Examining Division

Enclosure(s): 2 page/s reasons (Form 2906)
Pearson et al: Development of Improved Secondary Refrigerants,

Registered Letter
EPO Form 2001 12.07CSX

The examination is being carried out on the **following application documents:**

Description, Pages

2-7	as published	
1	filed with telefax on	28.01.2008

Claims, Numbers

1-16	filed with telefax on	03.11.2008

Drawings, Sheets

1/3-3/3	as published

1 The following document (D6) is cited by the Examiner (see Guidelines C-VI, 8.2 and 8.3). A copy of the document is annexed to the communication and the numbering will be adhered to in the rest of the procedure:

D6: Pearson S.F. et al., Development of Improved Secondary refrigerants, The Proceedings of The Institute of Refrigeration, vol. 89, pp.65-80,1992

2 As explained below, the features "*and wherein the cooling system (10) is controlled so that the temperature of the working fluid going to the air-to-fluid heat exchanger (30) is above the dew point in the space surrounding the heat load*" in the apparatus claim 1 relate to a method of using, operating the apparatus rather than clearly defining it in terms of its technical features. Concerning an apparatus claim, under such features must be construed as meaning merely apparatus **suitable for carrying out** the claimed functional features.

Bearing in mind the above remark the following objections has been made during the examination.

2 The present application does not meet the requirements of Article 52(1) EPC because the subject-matter of claim 1 does not involve an inventive step within the meaning of Article 56 EPC.

2.1 The subject-matter of claim 1 is formulated so broadly that even the disclosure of D6 falls within scope of said claim.
Said document discloses (cf. page 72, "Volatile secondary refrigerants" - page 74; Fig. 8)

a volatile working fluid;
a pump (Fig. 8);
an air-to-fluid heat exchanger (Fig. 8) in fluid communication with the pump and in thermal communication with the heat load;
a fluid-to-fluid heat exchanger (Fig. 8) having a first fluid path in fluid communication with the air-to-fluid heat exchanger and the pump, and a second fluid path connected to the heat exchange system, the first and second fluid paths being in thermal communication with one another;

The above system **is suitable for being operated** so that the temperature of the working fluid going to the air-to-fluid heat exchanger is above the dew point in the space surrounding the heat load.

In order to facilitate the examination of the conformity of the amended application with the requirements of Article 123(2) EPC, the applicant should clearly identify the amendments made, irrespective of whether they concern amendments by addition, replacement or deletion, and indicate the passages of the application as filed on which these amendments are based (see Guidelines E-II, 1).

LOCKE LIDDELL & SAPP LLP
JUL 12 2010

Robert Alan Swanson
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June 29, 2010

Albert B. Deaver, Jr., Esq.
Locke Lord Bissell & Liddell LLP
600 Travis Street
Suite 3400
Houston, Texas 77002

Dear Mr. Deaver,

As I was perusing some published patent applications and files, I noticed your patent application US 2005/0120737 A1, published June 9, 2005. I see that the January 2010 amendment to the claims adds limitations related to the ambient dew point. This language can be found in claim 1 of granted European Patent EP 1 143 778 B1. I bring this to your attention so that you may disclose to the United States Patent Office that the European Patent is prior art to your patent application.

Sincerely,

Robert A. Swanson

Robert A. Swanson, Esq.

encl: European Patent EP 1 143 778 B1

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Zeichen/Ref./Réf.	Anmeldung Nr./Application No./Demande no./Patent Nr./Patent No./Brevet no.
P40722EP-K/C8-J	04812840.9 - -

Anmelder/Applicant/Demandeur/Patentinhaber/Proprietor/Titulaire
Liebert Corporation

COMMUNICATION

The European Patent Office herewith transmits

- ☐ the European search report
- ☐ the declaration under Rule 45 EPC
- ☐ the partial European search report under Rule 45 EPC
- ☒ the European search report under Rule 112 EPC

relating to the above-mentioned European patent application. Copies of the documents cited in the search report are enclosed.

The following specifications given by the applicant have been approved by the Search Division :

☐ Abstract ☐ Title ☐ Figure

☐ The abstract was modified by the Search Division and the definitive text is attached to this communication.

☐ The following figure will be published with the abstract, since the Search Division considers that it better characterises the invention than the one indicated by the applicant.

Figure:

☐ Additional copy(copies) of the documents cited in the European search report.

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REFUND OF THE SEARCH FEE

If applicable under Article 10 Rules relating to fees, a separate communication from the Receiving Section on the refund of the search fee will be sent later.

EPO Form 1507 02.93

European Patent Office

EUROPEAN SEARCH REPORT

under Rule 112 EPC

Application Number
EP 04 81 2840

Category	DOCUMENTS CONSIDERED TO BE RELEVANT Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (IPC)
	On request of the applicant the present search report has been drawn up for claim 8 relating to invention 2 -----		INV. F25B25/00
X	US 6 148 626 A (IWAMOTO KAZUKI [JP]) 21 November 2000 (2000-11-21) * the whole document * -----	8	
X	US 2002/066280 A1 (OHKAWARA YOSHIO [JP]) 6 June 2002 (2002-06-06) * paragraph [0023] - paragraph [0071]; figures 1-3 * -----	8	
			TECHNICAL FIELDS SEARCHED (IPC)
			F25B F25D

Place of search	Date of completion of the search	Examiner
Munich	11 June 2007	Szilagyi, Barnabas

CATEGORY OF CITED DOCUMENTS

X : particularly relevant if taken alone
Y : particularly relevant if combined with another document of the same category
A : technological background
O : non-written disclosure
P : intermediate document

T : theory or principle underlying the invention
E : earlier patent document, but published on, or after the filing date
D : document cited in the application
L : document cited for other reasons

& : member of the same patent family, corresponding document

EPO FORM 1503 03.82 (P04C36)

5

ANNEX TO THE EUROPEAN SEARCH REPORT ON EUROPEAN PATENT APPLICATION NO. EP 04 81 2840

This annex lists the patent family members relating to the patent documents cited in the above-mentioned European search report.
The members are as contained in the European Patent Office EDP file on
The European Patent Office is in no way liable for these particulars which are merely given for the purpose of information.

11-06-2007

Patent document cited in search report	Publication date	Patent family member(s)	Publication date
US 6148626 A	21-11-2000	JP 3095377 B2	03-10-2000
		JP 11183005 A	06-07-1999
US 2002066280 A1	06-06-2002	JP 3504608 B2	08-03-2004
		JP 2002174438 A	21-06-2002
		KR 20020045518 A	19-06-2002
		SG 91942 A1	15-10-2002
		TW 504562 B	01-10-2002

EPO FORM P0459

For more details about this annex : see Official Journal of the European Patent Office, No. 12/82

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003217

Application No.	Ref.	Date
04 812 840.9 - 2301	P40722EP-K/C8-J	28.03.2008
Applicant: Liebert Corporation		

Communication pursuant to Article 94(3) EPC

The examination of the above-identified application has revealed that it does not meet the requirements of the European Patent Convention for the reasons enclosed herewith. If the deficiencies indicated are not rectified the application may be refused pursuant to Article 97(2) EPC.

You are invited to file your observations and insofar as the deficiencies are such as to be rectifiable, to correct the indicated deficiencies within a period

of 4 months

from the notification of this communication, this period being computed in accordance with Rules 126(2) and 131(2) and (4) EPC.

One set of amendments to the description, claims and drawings is to be filed within the said period on separate sheets (R. 50(1) EPC).

Failure to comply with this invitation in due time will result in the application being deemed to be withdrawn (Art. 94(4) EPC).

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Szilagyi, Barnabas
Primary Examiner
for the Examining Division

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Enclosure(s): 0 page/s reasons (Form 2906)
1996 ASHRAE HANDBOOK, HVAC SYSTEMS AND EQUIPMENTS

Registered Letter
EPO Form 2001 12.07CSX

The examination is being carried out on the **following application documents:**

Description, Pages

2-7	as published	
1	filed with telefax on	28.01.2008

Claims, Numbers

1-16	filed with telefax on	28.01.2008

Drawings, Sheets

1/3-3/3	as published

1 The following document (D5) is cited by the Examiner (see Guidelines C-VI, 8.2 and 8.3). A copy of the document is annexed to the communication and the numbering will be adhered to in the rest of the procedure:

D5: 1996 ASHRAE HANDBOOK, Heating, Ventilating, and Air-Conditioning SYSTEMS AND EQUIPMENT, pages 12.11-12.12

2 With letter dated 28.01.2008 you filed new claims 7-16 without indicating on which passages of the original application the features of the above claims are based. Since the features of said claims cannot be deducted from claims 1-9 as published you are invited to indicate the passages of the application as filed on which the features of said claims are based (see Guidelines E-II, 1).

Moreover, the examining division is of the opinion that the amendments filed with the letter dated 28.01.2008 introduce subject-matter which extends beyond the content of the application as filed, contrary to Article 123(2) EPC. The amendments concerned are the following:

2.1 Claim 1: according to the description, page 7, lines 6-8 " the temperature in the environment surrounding the equipment is maintained above the dew point to ensure that condensation does not occur". Whether that control is carried out by the controller (100) cannot be seen from the description as published, and said claim, therefore, does not meet the requirements of Article 123(2) EPC.

2.2 Claim 10: "..volatile fluid is a non water-based fluid." There is no basis in the description for this feature.

2.3 Claim 12:"... the flow regulator is adapted to control the amount of volatile working fluid .... independently of fluid pressure". Lines 20-21 on page 3 of the description clearly state, that "the flow regulator 32 preferably maintains a constant output flow independent of the inlet pressure over the operating pressure range of the system." Since, the term "fluid pressure" implies a broader scope than the term "inlet pressure" does, said claim infringes Article 123(2) EPC.

2.4 Claim 15: "receiver (50) is adapted to accumulate ... based upon temperature and/or heat load". No support can be found for this feature in the description.

3 Furthermore, notwithstanding the above-mentioned objections, the subject-matter of claim 1 does not involve an inventive step within the meaning of Article 56 EPC, and the requirements of Article 52(1) EPC are not therefore met.

3.1 Document D2 discloses (the references in parentheses applying to this document) a cooling system for transferring heat from a heat load to a heat exchange system, the cooling system comprising:
- a volatile working fluid;
- a pump (18);
- an air-to-fluid heat exchanger (12) (cf. column 3, line 62 - column 4, line 2) in fluid communication with the pump (18) and in thermal communication with the heat load; and
- a fluid-to-fluid heat exchanger (20) having a first fluid path in fluid communication with the air-to-fluid heat exchanger (12) and the pump (18), and a second fluid path connected to the heat exchange system, the first and second fluid paths being in thermal communication with one other.
- control system for controlling the flow of working fluid in the secondary system and for maintaining the temperature in an environment within a pre-defined temperature range (cf. paragraph 18).

The subject-matter of claim 1 therefore differs from this known cooling system in that the control system is arranged to maintain the temperature in the environment surrounding the heat load above the dew point to prevent condensation.

The problem to be solved by the present invention may therefore be regarded as that of avoiding condensing of moisture on heat exchangers.

Document D5 (page 12.12, left column, third whole paragraph), which is regarded as representing the common knowledge of a skilled person, mentions the dew-point temperature as one of the constraints that should be taken into account in designing control systems for chilled liquid systems. That is, the skilled person is aware of the problem set out above. If necessary to find a solution for the problem, he could set the temperature range above the dew point using merely his technical knowledge and in so doing he would arrive at a cooling system corresponding to that of claim 1.

3.2 For the sake of completeness, it is pointed out that the objection of lack of inventive step set out above could also have been substantiated starting from any of documents D1-D4 as the closest prior art and combining it with the common knowledge of a skilled person.

3.3 The features of claims 2-16 are known from at least one of the documents D1-D4 and thus, said claims are not allowable, Article 56 EPC.

In order to facilitate the examination of the conformity of the amended application with the requirements of Article 123(2) EPC, the applicant should clearly identify the amendments made, irrespective of whether they concern amendments by addition, replacement or deletion, and indicate the passages of the application as filed on which these amendments are based (see Guidelines E-II, 1).

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APPLICATION NUMBER	FILING or 371(c) DATE	GRP ART UNIT	FIL FEE REC'D	ATTY.DOCKET.NO	TOT CLAIMS	IND CLAIMS
13/601,481	08/31/2012	3744	2290	0021944-080US-1	29	5

CONFIRMATION NO. 9349

26720
LOCKE LORD BISSELL & LIDDELL LLP
600 TRAVIS STREET SUITE 2800
HOUSTON, TX 77002-3095

FILING RECEIPT

OC000000056517160

Date Mailed: 09/17/2012

Receipt is acknowledged of this non-provisional patent application. The application will be taken up for examination in due course. Applicant will be notified as to the results of the examination. Any correspondence concerning the application must include the following identification information: the U.S. APPLICATION NUMBER, FILING DATE, NAME OF APPLICANT, and TITLE OF INVENTION. Fees transmitted by check or draft are subject to collection. Please verify the accuracy of the data presented on this receipt. **If an error is noted on this Filing Receipt, please submit a written request for a Filing Receipt Correction. Please provide a copy of this Filing Receipt with the changes noted thereon. If you received a "Notice to File Missing Parts" for this application, please submit any corrections to this Filing Receipt with your reply to the Notice. When the USPTO processes the reply to the Notice, the USPTO will generate another Filing Receipt incorporating the requested corrections**

Applicant(s)

Steven A. BORROR, Columbus, OH;
Frank E. DIPAOLO, Dublin, OH;
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Stephen C. SILLATO, Westerville, OH;

Assignment For Published Patent Application

LIEBERT CORPORATION, Columbus, OH

Power of Attorney: None

Domestic Priority data as claimed by applicant

This application is a CON of 10/904,889 12/02/2004 PAT 8261565
which claims benefit of 60/527,527 12/05/2003

Foreign Applications (You may be eligible to benefit from the **Patent Prosecution Highway** program at the USPTO. Please see http://www.uspto.gov for more information.)

If Required, Foreign Filing License Granted: 09/13/2012

The country code and number of your priority application, to be used for filing abroad under the Paris Convention, is **US 13/601,481**

Projected Publication Date: 12/27/2012

Non-Publication Request: No

Early Publication Request: No

Title

COOLING SYSTEM FOR HIGH DENSITY HEAT LOAD

Preliminary Class

165

PROTECTING YOUR INVENTION OUTSIDE THE UNITED STATES

Since the rights granted by a U.S. patent extend only throughout the territory of the United States and have no effect in a foreign country, an inventor who wishes patent protection in another country must apply for a patent in a specific country or in regional patent offices. Applicants may wish to consider the filing of an international application under the Patent Cooperation Treaty (PCT). An international (PCT) application generally has the same effect as a regular national patent application in each PCT-member country. The PCT process **simplifies** the filing of patent applications on the same invention in member countries, but **does not result** in a grant of "an international patent" and does not eliminate the need of applicants to file additional documents and fees in countries where patent protection is desired.

Almost every country has its own patent law, and a person desiring a patent in a particular country must make an application for patent in that country in accordance with its particular laws. Since the laws of many countries differ in various respects from the patent law of the United States, applicants are advised to seek guidance from specific foreign countries to ensure that patent rights are not lost prematurely.

Applicants also are advised that in the case of inventions made in the United States, the Director of the USPTO must issue a license before applicants can apply for a patent in a foreign country. The filing of a U.S. patent application serves as a request for a foreign filing license. The application's filing receipt contains further information and guidance as to the status of applicant's license for foreign filing.

Applicants may wish to consult the USPTO booklet, "General Information Concerning Patents" (specifically, the section entitled "Treaties and Foreign Patents") for more information on timeframes and deadlines for filing foreign patent applications. The guide is available either by contacting the USPTO Contact Center at 800-786-9199, or it can be viewed on the USPTO website at http://www.uspto.gov/web/offices/pac/doc/general/index.html.

For information on preventing theft of your intellectual property (patents, trademarks and copyrights), you may wish to consult the U.S. Government website, http://www.stopfakes.gov. Part of a Department of Commerce initiative, this website includes self-help "toolkits" giving innovators guidance on how to protect intellectual property in specific countries such as China, Korea and Mexico. For questions regarding patent enforcement issues, applicants may call the U.S. Government hotline at 1-866-999-HALT (1-866-999-4158).

LICENSE FOR FOREIGN FILING UNDER

Title 35, United States Code, Section 184

Title 37, Code of Federal Regulations, 5.11 & 5.15

GRANTED

The applicant has been granted a license under 35 U.S.C. 184, if the phrase "IF REQUIRED, FOREIGN FILING LICENSE GRANTED" followed by a date appears on this form. Such licenses are issued in all applications where the conditions for issuance of a license have been met, regardless of whether or not a license may be required as set forth in 37 CFR 5.15. The scope and limitations of this license are set forth in 37 CFR 5.15(a) unless an earlier license has been issued under 37 CFR 5.15(b). The license is subject to revocation upon written notification. The date indicated is the effective date of the license, unless an earlier license of similar scope has been granted under 37 CFR 5.13 or 5.14.

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NOT GRANTED

No license under 35 U.S.C. 184 has been granted at this time, if the phrase "IF REQUIRED, FOREIGN FILING LICENSE GRANTED" DOES NOT appear on this form. Applicant may still petition for a license under 37 CFR 5.12, if a license is desired before the expiration of 6 months from the filing date of the application. If 6 months has lapsed from the filing date of this application and the licensee has not received any indication of a secrecy order under 35 U.S.C. 181, the licensee may foreign file the application pursuant to 37 CFR 5.15(b).

PATENT APPLICATION FEE DETERMINATION RECORD Substitute for Form PTO-875	Application or Docket Number 13/601,481

APPLICATION AS FILED - PART I

FOR	NUMBER FILED (Column 1)	NUMBER EXTRA (Column 2)	SMALL ENTITY RATE($)	SMALL ENTITY FEE($)	OR	OTHER THAN SMALL ENTITY RATE($)	OTHER THAN SMALL ENTITY FEE($)
BASIC FEE (37 CFR 1.16(a), (b), or (c))	N/A	N/A	N/A			N/A	380
SEARCH FEE (37 CFR 1.16(k), (i), or (m))	N/A	N/A	N/A			N/A	620
EXAMINATION FEE (37 CFR 1.16(o), (p), or (q))	N/A	N/A	N/A			N/A	250
TOTAL CLAIMS (37 CFR 1.16(i))	29 minus 20 =	* 9			OR	x 60 =	540
INDEPENDENT CLAIMS (37 CFR 1.16(h))	5 minus 3 =	* 2				x 250 =	500
APPLICATION SIZE FEE (37 CFR 1.16(s))	If the specification and drawings exceed 100 sheets of paper, the application size fee due is $310 ($155 for small entity) for each additional 50 sheets or fraction thereof. See 35 U.S.C. 41(a)(1)(G) and 37 CFR 1.16(s).						0.00
MULTIPLE DEPENDENT CLAIM PRESENT (37 CFR 1.16(j))							0.00
* If the difference in column 1 is less than zero, enter "0" in column 2.			TOTAL			TOTAL	2290

APPLICATION AS AMENDED - PART II

AMENDMENT A

	CLAIMS REMAINING AFTER AMENDMENT (Column 1)		HIGHEST NUMBER PREVIOUSLY PAID FOR (Column 2)	PRESENT EXTRA (Column 3)	SMALL ENTITY RATE($)	SMALL ENTITY ADDITIONAL FEE($)	OR	OTHER THAN SMALL ENTITY RATE($)	OTHER THAN SMALL ENTITY ADDITIONAL FEE($)
Total (37 CFR 1.16(i))	*	Minus	**	=	x =		OR	x =	
Independent (37 CFR 1.16(h))	*	Minus	***	=	x =		OR	x =	
Application Size Fee (37 CFR 1.16(s))									
FIRST PRESENTATION OF MULTIPLE DEPENDENT CLAIM (37 CFR 1.16(j))							OR		
					TOTAL ADD'L FEE		OR	TOTAL ADD'L FEE	

AMENDMENT B

	CLAIMS REMAINING AFTER AMENDMENT (Column 1)		HIGHEST NUMBER PREVIOUSLY PAID FOR (Column 2)	PRESENT EXTRA (Column 3)	SMALL ENTITY RATE($)	SMALL ENTITY ADDITIONAL FEE($)	OR	OTHER THAN SMALL ENTITY RATE($)	OTHER THAN SMALL ENTITY ADDITIONAL FEE($)
Total (37 CFR 1.16(i))	*	Minus	**	=	x =		OR	x =	
Independent (37 CFR 1.16(h))	*	Minus	***	=	x =		OR	x =	
Application Size Fee (37 CFR 1.16(s))									
FIRST PRESENTATION OF MULTIPLE DEPENDENT CLAIM (37 CFR 1.16(j))							OR		
					TOTAL ADD'L FEE		OR	TOTAL ADD'L FEE	

* If the entry in column 1 is less than the entry in column 2, write "0" in column 3.
** If the "Highest Number Previously Paid For" IN THIS SPACE is less than 20, enter "20".
*** If the "Highest Number Previously Paid For" IN THIS SPACE is less than 3, enter "3".
The "Highest Number Previously Paid For" (Total or Independent) is the highest found in the appropriate box in column 1.

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APPLICATION NUMBER	FILING OR 371(C) DATE	FIRST NAMED APPLICANT	ATTY. DOCKET NO./TITLE
13/601,481	08/31/2012	Steven A. BORROR	0021944-080US-1

CONFIRMATION NO. 9349

26720
LOCKE LORD BISSELL & LIDDELL LLP
600 TRAVIS STREET SUITE 2800
HOUSTON, TX 77002-3095

PUBLICATION NOTICE

OC000000058351657

Title:COOLING SYSTEM FOR HIGH DENSITY HEAT LOAD

Publication No.US-2012-0324930-A1
Publication Date:12/27/2012

NOTICE OF PUBLICATION OF APPLICATION

The above-identified application will be electronically published as a patent application publication pursuant to 37 CFR 1.211, et seq. The patent application publication number and publication date are set forth above.

The publication may be accessed through the USPTO's publically available Searchable Databases via the Internet at www.uspto.gov. The direct link to access the publication is currently http://www.uspto.gov/patft/.

The publication process established by the Office does not provide for mailing a copy of the publication to applicant. A copy of the publication may be obtained from the Office upon payment of the appropriate fee set forth in 37 CFR 1.19(a)(1). Orders for copies of patent application publications are handled by the USPTO's Office of Public Records. The Office of Public Records can be reached by telephone at (703) 308-9726 or (800) 972-6382, by facsimile at (703) 305-8759, by mail addressed to the United States Patent and Trademark Office, Office of Public Records, Alexandria, VA 22313-1450 or via the Internet.

In addition, information on the status of the application, including the mailing date of Office actions and the dates of receipt of correspondence filed in the Office, may also be accessed via the Internet through the Patent Electronic Business Center at www.uspto.gov using the public side of the Patent Application Information and Retrieval (PAIR) system. The direct link to access this status information is currently http://pair.uspto.gov/. Prior to publication, such status information is confidential and may only be obtained by applicant using the private side of PAIR.

Further assistance in electronically accessing the publication, or about PAIR, is available by calling the Patent Electronic Business Center at 1-866-217-9197.

Office of Data Managment, Application Assistance Unit (571) 272-4000, or (571) 272-4200, or 1-888-786-0101

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APPLICATION NO.	FILING DATE	FIRST NAMED INVENTOR	ATTORNEY DOCKET NO.	CONFIRMATION NO.
13/601,481	08/31/2012	Steven A. BORROR	0021944-080US-1	9349

26720 7590 01/31/2013

LOCKE LORD BISSELL & LIDDELL LLP
600 TRAVIS STREET SUITE 2800
HOUSTON, TX 77002-3095

EXAMINER	
ABDUR RAHIM, AZIM	

ART UNIT	PAPER NUMBER
3744	

NOTIFICATION DATE	DELIVERY MODE
01/31/2013	ELECTRONIC

Please find below and/or attached an Office communication concerning this application or proceeding.

The time period for reply, if any, is set in the attached communication.

Notice of the Office communication was sent electronically on above-indicated "Notification Date" to the following e-mail address(es):

hoip@lockelord.com

PTOL-90A (Rev. 04/07)

Office Action Summary	Application No.	Applicant(s)	
	13/601,481	BORROR ET AL.	
	Examiner	Art Unit	
	AZIM ABDUR RAHIM	3744	

-- The MAILING DATE of this communication appears on the cover sheet with the correspondence address --

Period for Reply

A SHORTENED STATUTORY PERIOD FOR REPLY IS SET TO EXPIRE *3* MONTH(S) OR THIRTY (30) DAYS, WHICHEVER IS LONGER, FROM THE MAILING DATE OF THIS COMMUNICATION.

- Extensions of time may be available under the provisions of 37 CFR 1.136(a). In no event, however, may a reply be timely filed after SIX (6) MONTHS from the mailing date of this communication.
- If NO period for reply is specified above, the maximum statutory period will apply and will expire SIX (6) MONTHS from the mailing date of this communication.
- Failure to reply within the set or extended period for reply will, by statute, cause the application to become ABANDONED (35 U.S.C. § 133). Any reply received by the Office later than three months after the mailing date of this communication, even if timely filed, may reduce any earned patent term adjustment. See 37 CFR 1.704(b).

Status

1)☒ Responsive to communication(s) filed on *07 September 2012*.

2a)☐ This action is **FINAL**. 2b)☒ This action is non-final.

3)☐ An election was made by the applicant in response to a restriction requirement set forth during the interview on _____; the restriction requirement and election have been incorporated into this action.

4)☐ Since this application is in condition for allowance except for formal matters, prosecution as to the merits is closed in accordance with the practice under *Ex parte Quayle*, 1935 C.D. 11, 453 O.G. 213.

Disposition of Claims

5)☒ Claim(s) *1-29* is/are pending in the application.

5a) Of the above claim(s) _____ is/are withdrawn from consideration.

6)☐ Claim(s) _____ is/are allowed.

7)☒ Claim(s) *1-29* is/are rejected.

8)☐ Claim(s) _____ is/are objected to.

9)☐ Claim(s) _____ are subject to restriction and/or election requirement.

* If any claims have been determined allowable, you may be eligible to benefit from the **Patent Prosecution Highway** program at a participating intellectual property office for the corresponding application. For more information, please see http://www.uspto.gov/patents/init_events/pph/index.jsp or send an inquiry to PPHfeedback@uspto.gov.

Application Papers

10)☐ The specification is objected to by the Examiner.

11)☒ The drawing(s) filed on *31 August 2012* is/are: a)☒ accepted or b)☐ objected to by the Examiner.

Applicant may not request that any objection to the drawing(s) be held in abeyance. See 37 CFR 1.85(a).

Replacement drawing sheet(s) including the correction is required if the drawing(s) is objected to. See 37 CFR 1.121(d).

Priority under 35 U.S.C. § 119

12)☐ Acknowledgment is made of a claim for foreign priority under 35 U.S.C. § 119(a)-(d) or (f).

a)☐ All b)☐ Some * c)☐ None of:

1.☐ Certified copies of the priority documents have been received.

2.☐ Certified copies of the priority documents have been received in Application No. _____.

3.☐ Copies of the certified copies of the priority documents have been received in this National Stage application from the International Bureau (PCT Rule 17.2(a)).

* See the attached detailed Office action for a list of the certified copies not received.

Attachment(s)

1) ☒ Notice of References Cited (PTO-892)

2) ☐ Information Disclosure Statement(s) (PTO/SB/08) Paper No(s)/Mail Date _____.

3) ☐ Interview Summary (PTO-413) Paper No(s)/Mail Date. _____ .

4) ☐ Other: _____.

DETAILED ACTION

Double Patenting

1. A rejection based on double patenting of the "same invention" type finds its support in the language of 35 U.S.C. 101 which states that "whoever invents or discovers any new and useful process ... may obtain a patent therefor ..." (Emphasis added). Thus, the term "same invention," in this context, means an invention drawn to identical subject matter. See *Miller v. Eagle Mfg. Co.*, 151 U.S. 186 (1894); *In re Ockert*, 245 F.2d 467, 114 USPQ 330 (CCPA 1957); and *In re Vogel*, 422 F.2d 438, 164 USPQ 619 (CCPA 1970).

A statutory type (35 U.S.C. 101) double patenting rejection can be overcome by canceling or amending the conflicting claims so they are no longer coextensive in scope. The filing of a terminal disclaimer cannot overcome a double patenting rejection based upon 35 U.S.C. 101.

2. Claims 6-25 are rejected under 35 U.S.C. 101 as claiming the same invention as that of claims 8-10, 12-22, 25, 34, 35, 37, 38, 41, 42 & 50 of prior U.S. Patent No. 8,261,565 to Borrer et al. (Borror). This is a double patenting rejection.

Claims 6-25 are provisionally rejected under 35 U.S.C. 101 as claiming the same invention as that of claims 6-25 of copending Application No. 13/607,934. This is a provisional double patenting rejection since the conflicting claims have not in fact been patented.

3. The nonstatutory double patenting rejection is based on a judicially created doctrine grounded in public policy (a policy reflected in the statute) so as to prevent the unjustified or improper timewise extension of the “right to exclude” granted by a patent and to prevent possible harassment by multiple assignees. A nonstatutory obviousness-type double patenting rejection is appropriate where the conflicting claims are not identical, but at least one examined application claim is not patentably distinct from the reference claim(s) because the examined application claim is either anticipated by, or would have been obvious over, the reference claim(s). See, e.g., *In re Berg*, 140 F.3d 1428, 46 USPQ2d 1226 (Fed. Cir. 1998); *In re Goodman*, 11 F.3d 1046, 29 USPQ2d 2010 (Fed. Cir. 1993); *In re Longi*, 759 F.2d 887, 225 USPQ 645 (Fed. Cir. 1985); *In re Van Ornum*, 686 F.2d 937, 214 USPQ 761 (CCPA 1982); *In re Vogel*, 422 F.2d 438, 164 USPQ 619 (CCPA 1970); and *In re Thorington*, 418 F.2d 528, 163 USPQ 644 (CCPA 1969).

A timely filed terminal disclaimer in compliance with 37 CFR 1.321(c) or 1.321(d) may be used to overcome an actual or provisional rejection based on a nonstatutory double patenting