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(74) Representative: **Rastelli, Franco**
Dott. Franco Cicogna & C. SRL
Via Visconti di Modrone, 14/A
20122 Milano (IT)

sired at the evaporator outlet on the basis of the vaporization ratio desired at the outlet and the measured vaporization pressure; calculating the refrigerant flow that provides a required enthalpy change between the inlet and the outlet of the evaporator given the estimated heat flow by inverting, knowing the pressure upstream and downstream of the thermal expansion valve and the density of the liquid refrigerant at the inlet, which is a function of the measured temperature, the characteristic discharge rule of the valve for calculating the opening necessary for achieving the calculated flow; and sending the previously calculated opening signal for the valve to an actuator of said thermal expansion valve.



Description

[0001] The present invention relates, in a first aspect thereof, to a method for controlling overheating or superheating in compression refrigeration cycles with a regenerative heat exchanger.

[0002] In a second aspect thereof, the present invention relates to a system for the implementation of the inventive method.

BACKGROUND OF INVENTION

[0003] As is known, the objective of the control system in compression refrigeration cycles is to maximize the thermal power taken by the evaporator from a heat-carrying fluid, normally the air of a cooling chamber of an air-conditioned area, at the same time avoiding sending a two-phase mixture to the compressor, which could damage it.

[0004] In conventional refrigeration cycles, the terminal section of the evaporator acts as a superheater, so as to send dry superheated vapour to the compressor.

[0005] In particular, the control objective is achieved with a simple feedback operation, where the level of superheating is measured at the evaporator outlet and the opening of the thermal expansion valve at the evaporator inlet is suitably modulated so that superheating remains around an opportune "SET POINT" value or reference point. The latter must be sufficiently high to ensure the absence of liquid drops at the compressor intake, with a safety margin during the transients, but at the same time it must be restricted to avoid high gas delivery temperatures. In any case, the presence of the superheating section forces the evaporator to work at a temperature, and therefore at a pressure, lower than that which it would be possible to achieve in the absence of the section, penalizing the cooling efficiency.

[0006] It is also known that it is possible to significantly improve this efficiency by using the waste heat of the high-temperature refrigerating fluid at the outlet of the condenser to superheat the refrigerating fluid in a heat recuperator separate from the evaporator. In this way, it is possible to reduce the temperature difference between evaporating fluid and air to be cooled at will, provided that the convective exchange surface and/or coefficient between the two is increased, thereby improving the cycle's thermodynamic efficiency.

[0007] In this plant engineering or structural configuration, the first control objective is to maintain the vaporization ratio at the evaporator outlet at a predetermined value of less than one, possibly at a value close to that of dry-out so as to ensure the best possible use of the heat exchange surface. The second objective is to ensure sufficient superheating of the refrigerating fluid at the outlet of the recuperator, i.e. at the inlet of the compressor so as to avoid damaging it.

[0008] The simplest method for achieving this objective is to use the same control strategy used in conventional cycles, namely measuring the level of vapour superheating at the outlet of the recuperator and using a feedback control that acts on the opening of the thermal expansion valve so as to bring it to and keep it approximately at an opportune set point value, corresponding to the cycle's optimal conditioning conditions.

[0009] Unfortunately, this configuration tends to very easily exhibit unstable behaviour, characterized by strong oscillations.

[0010] In particular, it is very difficult, if not impossible, to design a control rule that ensures stable behaviour in all of the system's possible operating conditions.

[0011] The origin of this difficulty is due to the combination of three phenomena. The first is the propagation delay between variations in flowrate of the thermal expansion valve and the corresponding variations in vaporization ratio at the evaporator outlet.

[0012] The second phenomenon is the extreme non-linearity of the relationship between the vaporization ratio at the evaporator outlet, and therefore at the recuperator inlet, and the superheating level at the recuperator outlet that is used for feedback. This non-linearity is due to the strong dependency on the vaporization ratio of the convective exchange coefficient in the inlet section of the recuperator, which provides the final evaporation section for the refrigerating fluid.

[0013] The third phenomenon is given by the further coupling introduced in the process by the recuperator, whereby even a modest increase in the vaporization ratio at the inlet entails a strong drop in the exchange coefficient and therefore a drop in the heat taken from the hot side, i.e. an increase in temperature on the hot side, which entails an increase in the vaporization ratio downstream of the thermal expansion valve and therefore a further increase in the vaporization ratio at the evaporator outlet.

[0014] This positive feedback mechanism is destabilizing and results in hysteresis phenomena that have been experimentally confirmed.

[0015] In conclusion, just the superheating measurement downstream of the recuperator is inadequate for implementing feedback adjustment that is reliable and stable in all process operating conditions, unless integrated with other measurements.

SUMMARY OF INVENTION

[0016] The aim of the present invention is therefore that of providing a method of controlling superheating in compression refrigeration cycles with regenerative heat exchanger that enables stabilizing the vaporization ratio at the evaporator outlet in an indirect manner.

[0017] Within the above-mentioned aim, a main object of the present invention is to provide a method of the indicated type that provides the sought stabilization through measurement and evaluation of at least the following parameters:

- temperature of the air to be cooled;
- temperature of the fluid upstream of the thermal expansion valve;
- pressure upstream of the thermal expansion valve;
- evaporation pressure.

[0018] Another object of the present invention is to provide a method of the indicated type that can be implemented with extremely simple and inexpensive instrumentation, in particular a single pressure sensor on the circuit's low-pressure line, positioned at the evaporator outlet, as well as a single temperature sensor and a single pressure sensor, both positioned upstream of the thermal expansion valve.

[0019] A further object of the present invention is to provide a method of the indicated type that operates on the basis of a new and novel control and regulation algorithm capable of stabilizing operation of the cooling system by directly compensating the effects of: changes in temperature of the air to be cooled; changes in condensation pressure upstream of the thermal expansion valve; changes in flow and temperature of the condenser's refrigerating fluid; and changes in evaporation pressure and in the heat exchange of the recuperator with subsequent variations in the enthalpy content of the refrigerant at the outlet of the recuperator and at the inlet of the thermal expansion valve.

[0020] A further object of the present invention is to provide a method of the indicated type that is extremely effective not only for local stabilization of the system, but also for large perturbations that involve the entire system.

[0021] A further object of the present invention is to provide a method of the indicated type that can be easily adapted to handle possible failures of the fans serving the evaporators.

[0022] A further object of the present invention is to provide a control and regulation system for implementing the inventive method, it being possible to configure this system as a modular device applicable to any cooling chamber or air-conditioned room and/or similar area, either of new construction or even of a pre-existing type, this system ensuring, thanks to the implementation of the inventive method, the maximization of the thermal power taken from the heat-carrying refrigerant by the evaporator, at the same time avoiding sending a two-phase mixture to the system's compressor, which might damage it.

[0023] The last, but not least object of the present invention is to provide a refrigeration cycle control and regulation system that can be built from readily and commercially available materials/components with reliable operation, as well as economically competitive costs.

[0024] According to one aspect of the present invention, the previously mentioned aim and the objects, as well as further objects, which shall become clearer hereinafter, are achieved by a method for controlling superheating in a refrigeration cycle system operating by compressing a refrigerating fluid with a regenerative heat exchanger, according to claim 1.

[0025] Other characteristics of the inventive method are defined in the dependent claims.

[0026] The aforementioned aim and objects are also achieved by a system for implementing the inventive method as set forth in the appended claims concerning the system.

BRIEF DESCRIPTION OF DRAWINGS

[0027] Further characteristics and advantages of the method and system according to the present invention shall become clearer from the following detailed description of currently preferred embodiments thereof, shown by way of indicative and non-limitative example in the accompanying drawings, in which:

FIG. 1 contains the pseudocode of the stabilization algorithm for the vaporization ratio at the outlet 9 of the evaporator 6 at a value close to that required;

FIG. 2 is a block diagram of the overall configuration of the control, regulation and stabilization system, forming the subject of the present invention; and

FIG. 3 shows the layout of a refrigeration cycle system with recuperator, in which use is made of the method and system of the invention.

DESCRIPTION OF PREFERRED EMBODIMENTS

[0028] A layout of the refrigeration cycle system with recuperator to which the method of the invention refers is shown in Figure 3. The system comprises a compressor 1, a condenser 2, a receiver 3 for liquid refrigerant, a heat recuperator or regenerative heat exchanger 4, a thermal expansion valve 5, an evaporator 6 and a transmission line 8, to the valve 5, for the signal detected by the sensor 7.

[0029] When the system is running, the refrigerating gas is compressed in the compressor 1, then enters the condenser 2 where it changes state, passing to a liquid phase, after which is then sent to the liquid receiver 3. The liquid leaving the receiver 3 is sent to the recuperator 4, where it is cooled by the gas leaving the evaporator 6. From the recuperator 4, the liquid is then made to enter the thermal expansion valve 5, where adiabatic expansion occurs, which forms a liquid/gas mixture that enters the evaporator 6. In the latter, a change of state occurs from the liquid to gas phase, which, in turn, enters the recuperator 4. The cycle ends with sending the gas from the recuperator 4 to the compressor 1.

[0030] As mentioned, the main problem in regulating the refrigeration cycle with recuperator 4 is that the variations in vaporization ratio of the evaporator 6 occur at its outlet 9 with a large delay with respect to the conditions at the inlet 10, but this variation is not directly detectable because a two-phase mixture that is in nearly isobaric and isothermal conditions passes through the evaporator and no reliable and inexpensive methods exist for measuring the vaporization ratio of this mixture.

[0031] Variations in vaporization ratio occur, even in a brutal manner, only in the recuperator 4, where latent heat transforms into sensible heat, but when this occurs, it is too late to take action on the thermal expansion valve 5 without causing oscillations and instability in process operation.

[0032] Determining the level of superheating of the vapour at the inlet of the compressor 1, which must be controlled, would in any case require measuring:

- the pressure P1 of the vapour at the outlet of the recuperator 4, necessary for calculating the corresponding saturation temperature; and
- the temperature T1 of the vapour measured by the sensor 7 at the outlet of the recuperator 4.

[0033] In any case, the temperature T2 of the air at the inlet of the evaporator 6 must also be measured, to allow its regulation by switching the compressor 1 on and off.

[0034] Thus, the fundamental idea of the inventive method was that of introducing additional process measurements that enable directly stabilizing the ratio of the liquid/gas mixture at the evaporator outlet, which is defined as the ratio between the mass of the gas phase and the total mass. In particular, these additional measurements regard:

- the temperature T3 of the refrigerating fluid upstream of the thermal expansion valve 5; and
- the pressure P2 of the fluid upstream of the thermal expansion valve 5.

[0035] In principle, implementation of the proposed method would also require measuring the evaporation pressure inside the evaporator 6. To keep down instrumentation costs, given the relatively small load losses on the circuit's low-pressure line, it is possible to ignore these load losses and use only one pressure sensor P1 on this line, positioned at the outlet of the recuperator 4 at point 11. In this way, the additional instrumentation that needs to be installed on the circuit is reduced to a single temperature sensor T3 and a single pressure sensor P2, both positioned upstream of the thermal expansion valve 5 at point 10.

[0036] It is also necessary to know:

- the nominal discharge coefficient $K_{v,nom}$ and the characteristic opening curve $f(\theta_v)$ of the thermal expansion valve 5, such that the effective discharge coefficient based on the opening θ_v is given by $K_v = K_{v,nom} \cdot f(\theta_v)$. The value of function $f()$ varies from zero (valve completely closed) to 1 (valve completely open). Both data items can be deduced from data provided by the manufacturer of the valve and/or experimentally measured via bench tests.
- the equivalent thermal conductance G of the evaporator 6, i.e. the factor that, multiplied by the temperature jump ΔT between the evaporation temperature of the liquid at the outlet of the evaporator 6 at pressure P1 and the temperature T2 of the air to be cooled, produces the thermal power absorbed by the evaporator 6, $Q=G \cdot \Delta T$; this value can be obtained from the evaporator design data and/or experimental measurements in a thermostatic cell, and may depend on the number and/or speed of the active fans that feed the evaporator.
- the specific enthalpy curves of the liquid and of the saturated vapour $h_{ls}(p)$ and $h_{vs}(p)$ as a function of refrigerant pressure, the density curve of the saturated liquid as a function of the saturation temperature $\rho_{ls}(T)$, the saturation temperature curve as a function of refrigerant pressure $T_s(p)$, as well as the mean specific heat C_p of the refrigerant in the undercooled liquid state. These curves are known from the literature for all industrially employed refrigerants and can be approximated via polynomials or other interpolation formulae easily implementable in industrial control-

lers.

[0037] According to the present invention, the inventive method operates on the basis of an innovative control algorithm, capable of stabilizing the vaporization ratio at the outlet of the evaporator 6 and based on the following process steps, represented by the pseudocode in Figure 1.

[0038] In a first step S1, the evaporation temperature T_{ev} is calculated from the pressure P1 measured at the recuperator outlet at point 7, which is approximately equal to the pressure inside the evaporator, according to the formula:

$$(S1) \quad T_{ev} = T_s(P1)$$

where T_s represents the saturation temperature of saturated vapour at pressure P1.

[0039] In a successive operating step S2, the thermal power Q absorbed by the evaporator 6 is calculated on the basis of the difference between the measured air temperature T2 and the evaporation temperature T_{ev} multiplied by the equivalent thermal conductance G, according to the formula:

$$(S2) \quad Q = G.(T2 - T_{ev})$$

[0040] In a successive step S3, the specific enthalpy h_{in} of the refrigerant at the inlet of the thermal expansion valve 5 is measured at point 10, as a function of its measured temperature T3, according to the formula:

$$(S3) \quad h_{in} = h_{ls}(P1) + c_p.(T3 - T_{ev})$$

where h_{ls} is the enthalpy of the saturated liquid at pressure P1 and c_p is the constant-pressure specific heat.

[0041] In a successive step S4, the desired enthalpy h_u at the outlet 9 of the evaporator 6 is calculated on the basis of the desired vaporization ratio X_v at the outlet of the evaporator 6 and at the measured evaporation pressure P1, according to the formula:

$$(S4) \quad h_u = h_{ls}(P1) + x_v.(h_{vs}(P1) - h_{ls}(P1))$$

where h_{vs} is the enthalpy of the saturated vapour at pressure P1.

[0042] In a successive step S5, given the estimated thermal power Q, the flow m_r of refrigerant that causes the required enthalpy change between the inlet 10 and outlet 9 of the evaporator 6 is calculated according to the formula:

$$(S5) \quad m_r = Q / (h_u - h_{in})$$

[0043] In a successive step S6, knowing the pressure upstream P2 and downstream P1 of the thermal expansion valve 5 and the density of the refrigerating liquid at the inlet 10, which is a function of the measured temperature T3, the necessary discharge coefficient of the valve 5 for achieving the flow calculated in step S5 is then calculated. In this regard, it may be appropriate to introduce a correction dp that takes into account the additional load losses on the distributor 12, according to the formula:

$$(S6) \quad K_v = m_r / \sqrt{\rho_{ls}(T3).(P2 - P1 - dp)}$$

where the sqrt() function calculates the square root, K_v is the discharge constant of the thermal expansion valve and ρ_{ls} is the density of the saturated liquid at temperature T3.

[0044] In the final step S7, the opening of the corresponding valve is calculated, using the inverse function of the opening characteristic. The corresponding signal is sent to the actuator of the thermal expansion valve, according to the formula:

$$(S7) \quad \theta_v = f^{-1}(K_v / K_{v,nom})$$

where θ_v is the level of opening of the valve and f^{-1} is an inverse function of the ratio $K_v/K_{v \text{ nominale}}$.

[0045] The steps of the algorithm that regulate the inventive method and are briefly described above enable achieving, in normal running under nominal conditions, as has been proven by the Applicants, the desired vaporization ratio of the liquid/gas mixture at the outlet 9 of the evaporator 6.

[0046] In particular, stabilization is achieved by directly compensating, according to the inventive method, the effect of the following phenomena:

- change in air temperature T2;
- change in condensation pressure P2 upstream of the thermal expansion valve 5, possibly caused by changes in the flow and temperature of the refrigerating fluid of the condenser;
- changes in evaporation pressure P1;
- changes in heat exchange in the recuperator 4, with consequent variations in the enthalpy content of the refrigerant leaving the recuperator and entering the thermal expansion valve.

[0047] In particular, an expert in the field will understand that the compensation of the last two effects eliminates the destabilizing positive feedback of the prior art that has been previously mentioned.

[0048] Experimental surveys carried out by the Applicants have shown that the previously described method is effective for stabilizing the system.

[0049] The previously described base algorithm, on which the inventive method is built, is capable of stabilizing operation of the system. However, uncertainty in the values of the parameters, in particular the equivalent conductance of the evaporator and measurement errors, can result in stabilizing operation of the evaporator at actual values of vaporization ratio at the outlet of the evaporator 6 and of superheating at the inlet of the compressor 1 that are significantly different from those required.

[0050] As it is not possible to measure the value of the vaporization ratio at the evaporator outlet directly, by introducing a direct feedback, provision is instead made, according to the invention, to use the control diagram shown in Fig. 2. Block S represents the above-described algorithm, where X_{ev}° represents the vaporization ratio x_v° required at the outlet of the evaporator 6. Block R represents a conventional proportional-integral-derivative (PID) controller. Block P represents the process to be controlled, where T_{mv} and P_{mv} correspond to the temperature and pressure upstream of the valve T3 and P2, P_{ev} corresponds to the evaporation pressure P1, and T_{vr} corresponds to the temperature of the vapour at the outlet of the recuperator T4. Block DT represents the calculation of the level of superheating from the pressure and temperature values measured at the outlet of the recuperator at point 7, which can be calculated with the formula $DT = T4 - T_s(P1)$.

[0051] According to this overall control diagram, the feedback loop for the level of superheating at the outlet of the recuperator 4 is implemented by a PID controller, which acts on the value of the vaporization ratio required at the outlet of the evaporator 6 by using it as a virtual control variable. The above-described control algorithm sets the opening of the thermal expansion valve 5 so as to obtain this vaporization ratio.

[0052] Given the presence of a significant delay between the moment when the required vaporization ratio is changed and the moment when the superheating reacts, due to the dynamics of the evaporator, it is necessary that this feedback is set with a small passband to prevent the onset of oscillations or instability. The values of the PID controller's parameters can be easily set up by an expert operator in the installation phase, considering the system resulting from the connection of blocks S and P as the system to be controlled. This task can also be carried out by an opportune self-calibrating algorithm chosen from those available on the market or in the literature. Experimental surveys carried out by the Applicants have shown that this setting up activity is not critical and that a fixed calibration of the parameters is sufficient to ensure proper functioning of the system in all possible operating conditions.

[0053] An expert in the field will understand that this strategy is fundamentally different from the standard one, in which block S in Figure 2 is absent and the feedback on the level of superheating carried out by the PID controller acts directly on the opening of the valve. In particular, as already observed, the additional feedbacks of signals P1, P2 and T3, carried out by block S, efficiently perform the role of stabilizing and regulating process operation. This facilitates the operation and the setup of the PID controller, which has the sole task of introducing a correction to the required value for the vaporization ratio x_v° , so as to obtain, under normal running, the value of the required superheating level at point 7 of the circuit, despite the uncertainty regarding the values of the process parameters.

[0054] With regard to the starting transient, it is assumed to start from conditions in which the evaporator is empty or, in any case, at a very low pressure corresponding to a minimum refrigerant content. When the compressor is started, the thermal expansion valve is initially opened to a steady value that can be adapted as a function of the evaporation pressure, to take into account the different operating conditions at various temperatures. Then there is a waiting period until the pressure exceeds a threshold value, determined on the basis of the saturation pressure reduced by an opportune margin. Once the threshold is exceeded, the previously defined control and regulation algorithm-method starts.

[0055] In this case as well, experimental tests have shown that in this way, the superheating undershoots at the

compressor inlet with respect to the set point value remain within a few degrees, and so a value of around 10-15 degrees for this set point ensures an ample safety margin.

[0056] When the compressor must be stopped, for example, based on the thermostatic control of the air temperature, it is necessary to first completely close the thermal expansion valve, so as to empty the evaporator; the compressor can be stopped when the pressure drops below an opportune minimum threshold.

[0057] If this sequence is not feasible, since control of the compressor is independent of that of the thermal expansion valve, it is preferable to modify the start procedure in this way: upon starting the compressor, the valve is kept closed until the pressure drops below the minimum threshold, after which the previously described sequence is followed.

[0058] Starting with an empty evaporator serves to guarantee the repeatability of the manoeuvre, in particular with regard to potentially hazardous superheating undershoots, which can entail the transitory presence of liquid at the compressor intake, with consequent mechanical damage.

[0059] According to a further characteristic of the invention, the method and the algorithm employed therein can be advantageously adapted to handle failures of the fans (not shown) serving the evaporators.

[0060] In the case of an evaporator with a single fan, the only possible strategy is that of identifying the failure condition by introducing a lower superheating threshold, sufficiently distant from the set point to avoid false alarms, but at the same time sufficiently high to avoid the intake of two-phase fluid into the compressor.

[0061] When this threshold is exceeded, the thermal expansion valve is immediately closed to avoid flooding the evaporator and the intake of two-phase fluid into the compressor.

[0062] It should be noted that in this case, it is not possible to continue operating the system in degraded conditions.

[0063] In the case of systems with N multiple fans, it can be assumed that the failure affects only one of them, and in this case it may make sense to continue operating the system in degraded conditions without stopping it, waiting for maintenance operations. It is therefore necessary to set a lower superheating alarm threshold and an even lower one for stopping.

[0064] When the first threshold is exceeded, the controller will initially assume that the functionality of one out of N fans has been lost. In a first approximation, this entails a reduction by a factor of $1/N$ of the equivalent conductance G of the evaporator. It is therefore possible to modify this parameter in the stabilization algorithm inside block S in Fig. 2 in this sense.

[0065] It should be noted that the immediate effect of this modification will be a drop in the estimated heat flow Q, which will be followed by an immediate drop in the required flow for the thermal expansion valve m_r , which will be immediately carried out by a reduction in the opening of the valve θ_v , so as to maintain the heat balance of the evaporator.

[0066] If the failure is actually due to the loss of a single fan, the system will stabilize itself in the required superheating and vaporization ratio conditions, obviously with cooling power reduced by a factor of $1/N$.

[0067] This operating condition can be reported to the supervising operator, for the latter to activate the maintenance procedure, which is not necessarily immediate.

[0068] Instead, if the superheating continues to drop and falls below the stopping threshold, then it is necessary to completely close the thermal expansion valve and stop the compressor once the evaporator is empty.

[0069] The above-described possibility of adapting the inventive method to handle failures in the fans serving the evaporators also constitutes an important aspect of the present invention, not in any way inferable from corresponding control or cooling systems of the prior art.

[0070] It can be noted from the foregoing that the invention fully achieves the intended aim and objects.

[0071] Although the invention has been previously described with reference to currently preferred embodiments thereof, it should be borne in mind that the described embodiments are susceptible to numerous modifications and variants, all falling within the scope of the inventive concept.

Claims

1. A method for controlling superheating in a refrigeration cycle system operating by compressing a refrigerating fluid with a regenerative heat exchanger (4), said system comprising compressor means (1) having an inlet and an outlet, evaporator means (6) having an inlet (10) and an outlet (9), thermal expansion valve means (5) coupled to said inlet (10) of said evaporator means (6), and recuperator means (4) having an inlet (9) and an outlet, **characterized in that** said method stabilizes, at least locally, the operation of said evaporator (6) by executing a stabilization algorithm for operation of the evaporator (6), said evaporator including, in the following order, at least the steps of: calculating the evaporation temperature (T4) from the pressure (P1) measured in the evaporator; calculating the thermal power absorbed by the evaporator (6) on the basis of the difference between air temperature (T2) and evaporation temperature (T4), multiplied by the equivalent conductance; calculating the enthalpy of the refrigerant at the inlet (10) of the thermal expansion valve (5) as a function of its measured temperature (T3); calculating the enthalpy desired at the outlet (9) of the evaporator (6) on the basis of the vaporization ratio desired at the outlet (9) and the measured

evaporation pressure P1; calculating the refrigerant flow that provides a required enthalpy change between the inlet (10) and the outlet (9) of the evaporator (6) given the estimated thermal power by inverting, knowing the pressure upstream (P2) and downstream (P1) of the thermal expansion valve (5) and the density of the liquid refrigerant at the inlet (10), which is a function of the measured temperature (T3), the characteristic discharge rule of the valve (5) for calculating the opening necessary for achieving the calculated flow; and sending the calculated opening signal of said valve to an actuator of said thermal expansion valve (5).

2. A method according to claim 1, **characterized in that** it includes a further step of limiting the speed of opening and closing said thermal expansion valve means to a maximum value compatible with the maximum speed of opening and closing said actuator means.
3. A method according to claim 1, **characterized in that** it comprises a further corrective step consisting of feeding back, with reduced passband, a superheating value to said inlet of said compressor, said superheating value acting on a value of the required vaporization ratio, used as a virtual variable of said evaporator.
4. A method according to any of the preceding claims, **characterized in that** said evaporator means comprise an evaporator with a single fan and said method includes a further step of identifying a possible failure condition of said single fan by introducing a lower superheating threshold sufficiently distant from a set point to avoid false alarms, but at the same time sufficiently high enough to prevent two-phase liquid discharge in said compressor, said thermal expansion valve means being immediately closed and said cooling system being prevented from continuing to operate upon said lower threshold being exceeded.
5. A method according to any of the preceding claims, **characterized in that** said evaporator means comprise a plurality of fan means, said method comprising, in the case where only one of said fan means fails, the step of setting a lower superheating alarm threshold and a corresponding lower stopping threshold of said system to allow said system to continue to operate in degraded conditions, without passing to a stop condition, waiting for not necessarily immediate maintenance operations on said one failed fan means to restore said system to a condition of full operability.
6. A system of controlling and regulating the method according to any of the preceding claims, **characterized in that** said system comprises at least one logic block (S) for implementing said stabilization algorithm for operation of the evaporator, at least one logic block (R) for PID-type regulation, and at least one logic block (DT) for calculating the level of superheating, all operatively and mutually interconnected.
7. A system according to claim 6, **characterized in that** said stabilization algorithm generated by said block (S) provides, in normal running and under optimal conditions, the desired vaporization ratio at the outlet of said evaporator and the desired stabilization by directly compensating at least: the change in temperature of the air, the change in condensation pressure upstream of the thermal expansion valve, the change in evaporation pressure; and the change in heat exchange in the recuperator, with consequent variations in the enthalpy content of the refrigerant at the outlet of the recuperator and at the inlet of the thermal expansion valve.

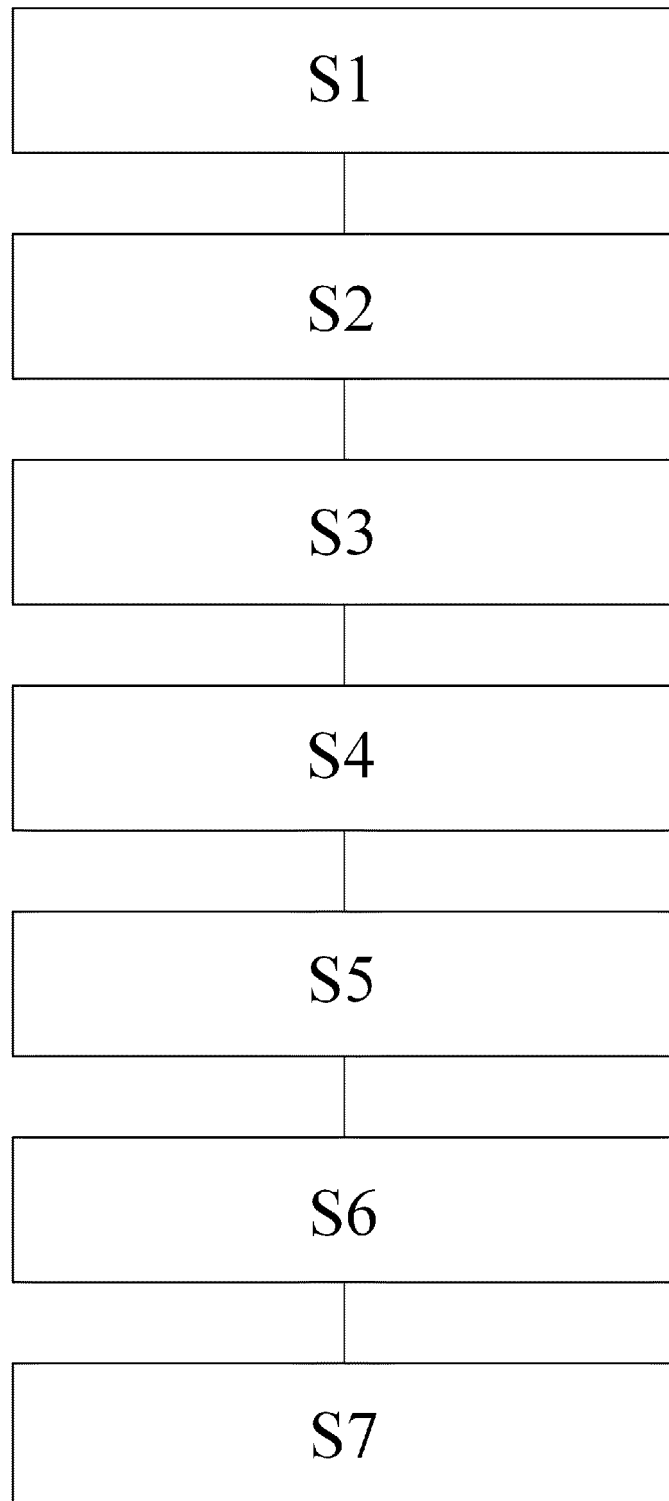


FIG.1

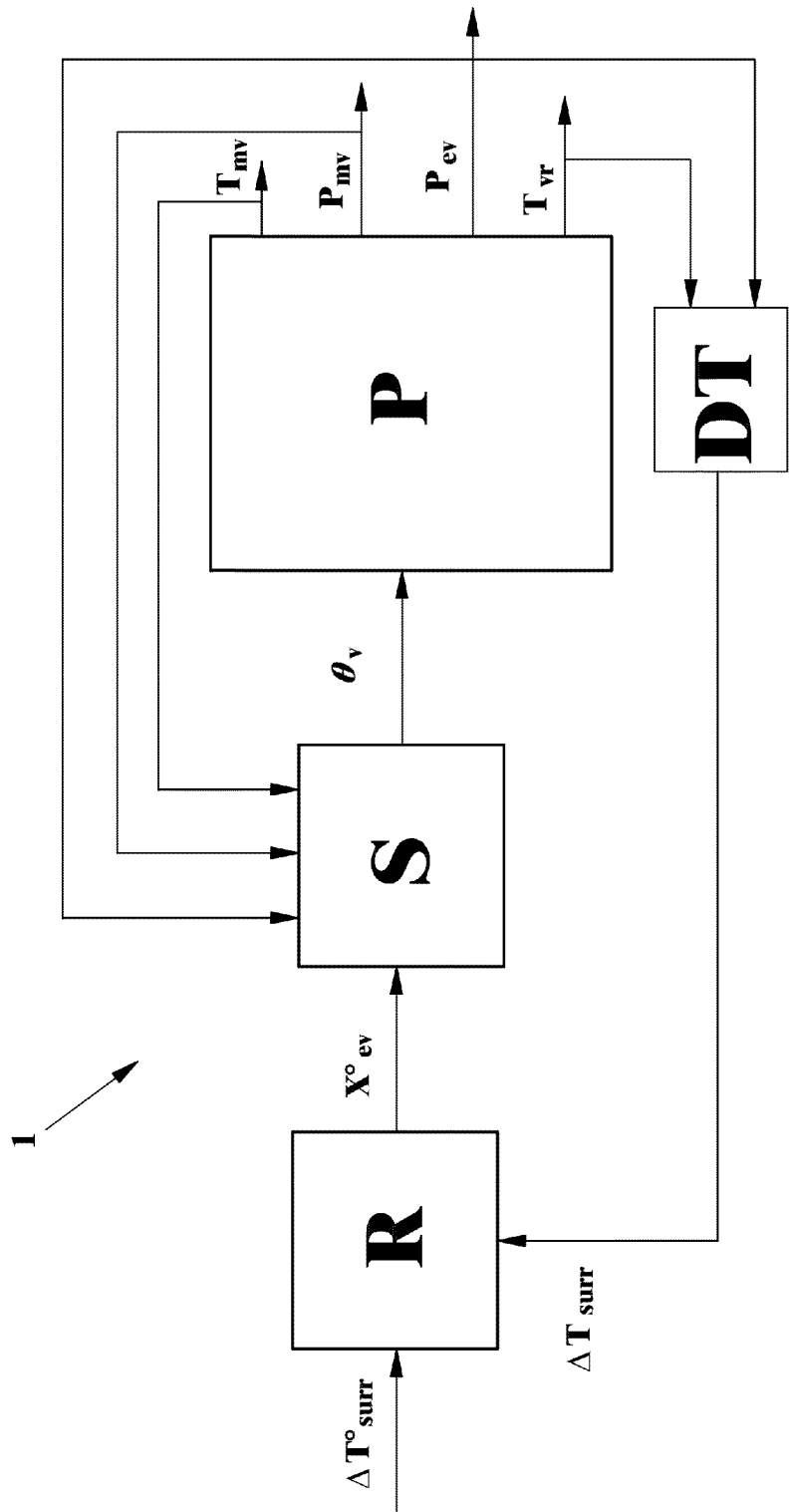


FIG.2

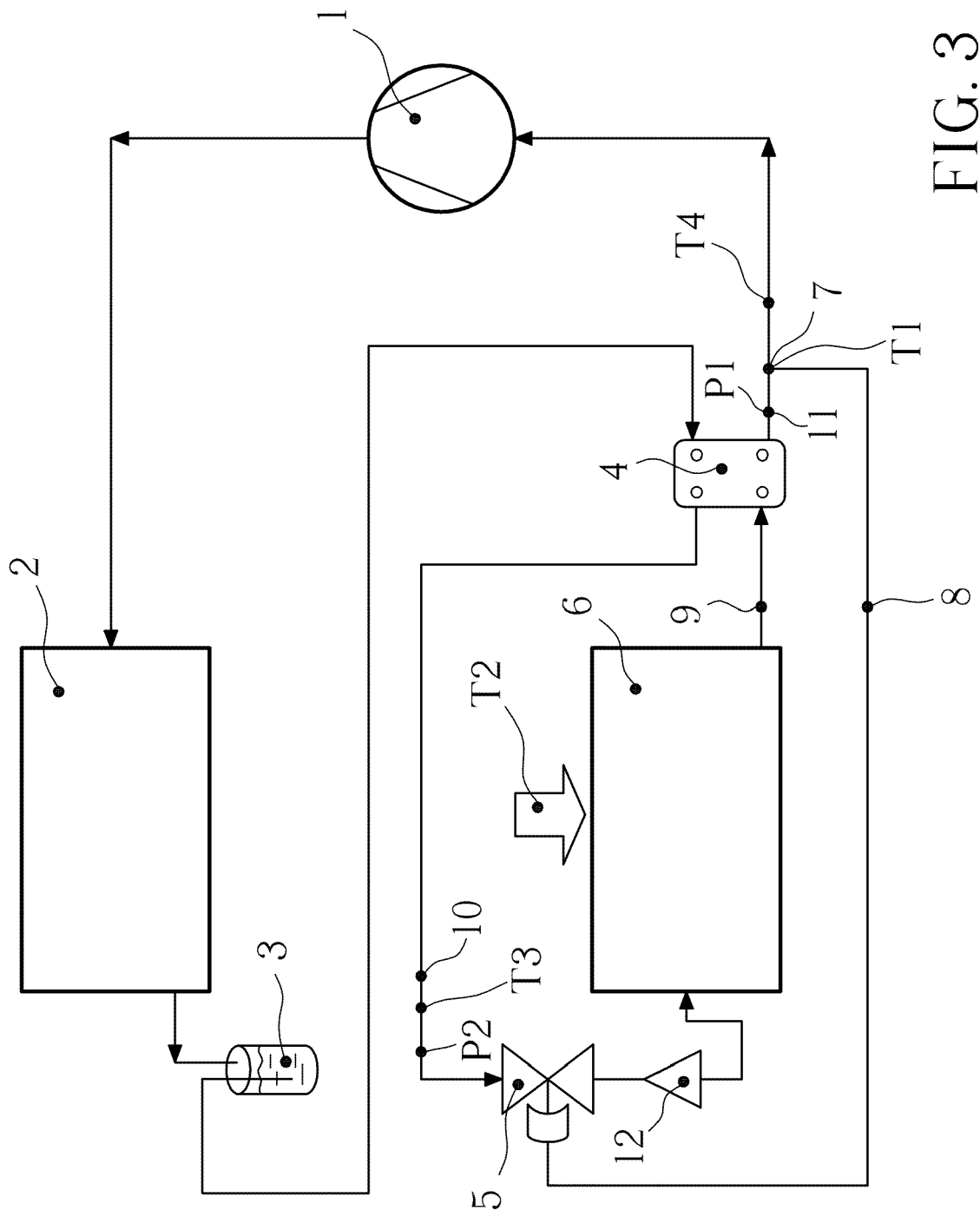


FIG. 3



EUROPEAN SEARCH REPORT

 Application Number
 EP 17 17 2660

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DOCUMENTS CONSIDERED TO BE RELEVANT			
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The present search report has been drawn up for all claims			
Place of search Munich		Date of completion of the search 27 September 2017	Examiner Lucic, Anita
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**ANNEX TO THE EUROPEAN SEARCH REPORT
ON EUROPEAN PATENT APPLICATION NO.**

EP 17 17 2660

5 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
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