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(54) **Method for estimation the thermal load of a circuit for a service fluid at outlet from a refrigerating machine**

(57) In a refrigerating machine (3) for an air-conditioning system (1), which is provided with one or more fan coils (2) and with a hydronic circuit (15) having a delivery branch (16) for circulation of a service fluid (5) from the refrigerating machine (3) to the fan coils (2) and a return branch (17) for return of the service fluid (5) at inlet to the refrigerating machine (3), via a pair of temperature sensors (21, 22), a delivery temperature (TDLV) of the service fluid (5) at outlet from the refrigerating machine (3) and a return temperature (TRET) of the service fluid (5) at inlet to the refrigerating machine (3) are measured, and the thermal load (FL) of the hydronic circuit (15) is estimated (106, 107, 109, 111) by processing, via a Kalman filter, the measurements of the delivery temperature (TDLV) and return temperature (TRET).

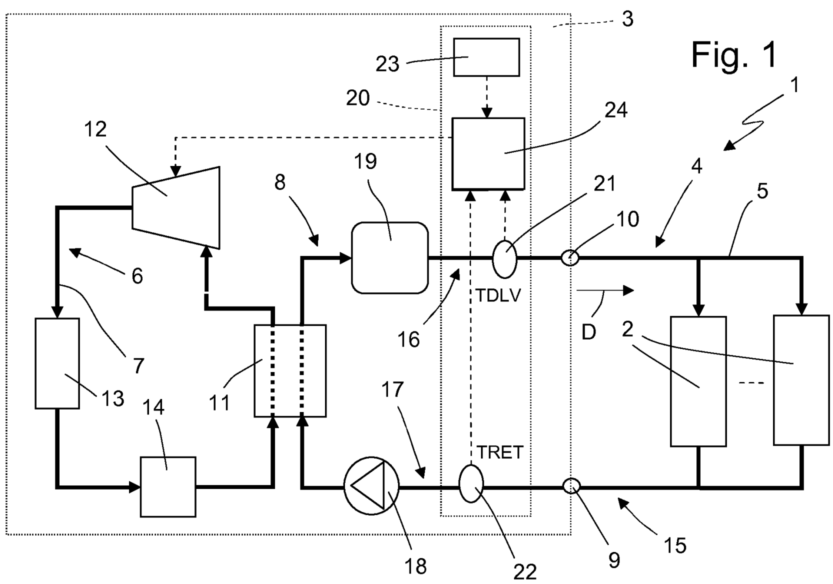

Fig. 1

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Description

[0001] The present invention relates to a method for estimating the thermal load of a circuit for a service fluid at outlet from a refrigerating machine.

[0002] In particular, the present invention finds advantageous, though not exclusive, application in the estimation of the thermal load of a hydronic circuit of centralized air-conditioning systems, to which the ensuing description will make explicit reference, without this implying any loss of generality.

[0003] As is known, a centralized air-conditioning system for control of ambient temperature of a building comprises a plurality of fan coils, which are appropriately distributed inside the building and are connected to one another via a hydraulic circuit, and a centralized refrigerating machine designed to cool a service fluid, and in particular a liquid coolant substantially constituted by water, and to convey said service fluid to the various fan coils through said hydraulic circuit.

[0004] Said refrigerating machine, commonly referred to as "chiller", comprises an internal circuit, circulating in which is an working fluid constituted by a refrigerant fluid, an output circuit, which closes on the hydraulic circuit of the air-conditioning system in an area corresponding to an inlet and an outlet of the refrigerating machine, a heat exchanger traversed by the internal circuit and by the output circuit for heat exchange between the working fluid and the service fluid, and one or more compressors for actuating a refrigerating cycle on the working fluid via compression of the working fluid itself. The hydraulic circuit of the air-conditioning system and the output circuit of the refrigerating machine form a so-called hydronic circuit.

[0005] Moreover known are electronic control systems for controlling turning-on and turning-off of the compressors in such a way that the temperature of the service fluid at inlet to or outlet from the refrigerating machine, i.e. the return temperature or the delivery temperature, respectively, of the service fluid, will reach a pre-determined set point.

[0006] Said control systems basically implement a control logic of a proportional type in which turning-on and turning-off of the compressors is carried out on the basis of a direct comparison between a measurement of the return temperature or delivery temperature of the service fluid and a pair of temperature thresholds.

[0007] The control systems referred to above present intrinsic limits due to temporal constraints between the instants of turning-on and turning-off of the compressors for the purpose of lengthening the service life thereof. Said constraints in effect limit the differential between the temperature thresholds to a minimum value, below which the compressors are forced to operate in technically prohibitive conditions of operation that can cause damage thereto. More in general, said temporal constraints mean that the control systems mentioned above do not enable, on the one hand, a good degree of precision of regulation of the temperature of the service fluid and, on the other hand, maximization of the energy efficiency of the air-conditioning system.

[0008] The aim of the present invention is to provide a method for estimating the thermal load of a hydronic circuit of an air-conditioning system governed by a refrigerating machine and to provide a control device for the refrigerating machine implementing said method that will enable regulation of the delivery temperature in a precise way, and maximization of the energy efficiency of the system, and, at the same time, will be easy and inexpensive to produce.

[0009] Provided according to the present invention are a method for estimating the thermal load of a service circuit for a service fluid at outlet from a refrigerating machine, a control device for a refrigerating machine, and a refrigerating machine in accordance with the annexed claims.

[0010] The present invention will now be described with reference to the annexed drawings, which illustrate a non-limiting example of embodiment thereof and in which:

- Figure 1 illustrates a block diagram of an air-conditioning system comprising a refrigerating machine provided with a control device in accordance with the present invention;
- Figures 3 to 7 illustrate a flowchart of the method for estimating the thermal load of a circuit for a service fluid at outlet from the refrigerating machine of Figure 1 in accordance with the present invention; and
- Figure 2 and Figures 8a to 9b illustrate calculation curves and tables of parameters used in the flowchart of Figures 3 to 7.

[0011] In Figure 1, designated as a whole by 1 is an air-conditioning system comprising a plurality of fan coils 2 appropriately distributed inside a building (not illustrated) of which it is intended to control the ambient temperature, and a refrigerating machine 3 designed to cool and cause circulation, along a hydraulic circuit 4 that connects the fan coils 2 to the refrigerating machine 3 itself, of a service fluid 5, and in particular a liquid coolant substantially constituted by water.

[0012] The refrigerating machine 3 typically comprises an internal circuit 6, circulating in which is an working fluid 7 constituted by a refrigerant fluid, and an output circuit 8, which closes on the hydraulic circuit 4 of the system 1 in an area corresponding to an inlet 9 and an outlet 10 of the refrigerating machine 3. Set along the internal circuit 6 are a series of devices for actuating a refrigerating cycle on the working fluid 7, and in particular: a first heat exchanger 11 traversed by the internal circuit 6 and by the output circuit 8 functioning as evaporator, i.e., for causing evaporation at low pressure of the working fluid 7 by absorbing heat from the service fluid 5; a compressor 12, preferably of a scroll type, for performing an adiabatic compression on the working fluid 7 in the vapour state; a second heat exchanger 13 functioning as condenser, i.e., for causing condensation of the working fluid 7 in such a way that it can release on the outside the heat previously absorbed; and an expansion valve 14 for cooling and causing partial evaporation of the working liquid 7 in such a way that it is ready for another cycle.

[0013] The hydraulic circuit 4 of the system 1 and the output circuit 8 of the refrigerating machine 3 form a so-called hydronic circuit 15, comprising a delivery branch 16, along which the service fluid 5 circulates in a direction D oriented from the heat exchanger 11 to the fan coils 2, and a return branch 17, along which the service fluid 5 returns to the heat exchanger 11. The circulation of the service fluid 5 in the direction D is guaranteed by a pump 18 set along the return branch 17.

[0014] The refrigerating machine 3 is provided with a storage tank 19 set along the delivery branch 16 at a short distance from the heat exchanger 11 for producing a thermal inertia in the hydronic circuit 15 that slows down the dynamics of the system 1 in such a way as to prevent undesirable phenomena of oscillation in the regulation valves (not illustrated) of the fan coils 2. The presence of the storage tank 19 is optional.

[0015] The refrigerating machine 3 further comprises a control device 20 for controlling turning-on and turning-off of the compressor 12 as a function of a delivery temperature TLDV of the service fluid 5.

[0016] In greater detail, the control device 20 comprises a first temperature sensor 21 set along the delivery branch 16 at output from the storage tank 19, i.e., at the outlet 10 of the refrigerating machine 3, for measuring the delivery temperature TDLV of the service fluid 5, a second temperature sensor 22 set along the return branch 17 in an area corresponding to the inlet 9 of the refrigerating machine 3 for measuring the return temperature TRET of the service fluid 5, which corresponds to a desired value of the temperature of evaporation of the working fluid 7 in the heat exchanger 11, a keypad 23 for receiving commands imparted by a user, and an electronic control unit 24 connected to the sensors 21 and 22, to the keypad 23 and to the compressor 12.

[0017] The electronic control unit 24 is designed to control turning-on and turning-off of the compressor 12 as a function of the comparison between the delivery temperature TDLV and a pair of delivery-temperature thresholds in such a way that the delivery temperature TDLV will converge on a delivery-temperature set point TSET comprised between the two delivery-temperature thresholds.

[0018] The electronic control unit 24 implements a method for estimating the thermal load of the hydronic circuit 15 in accordance with the present invention, said method being described in detail hereinafter for the case where the service fluid 5 is cooled for cooling the environments in which the fan coils 2 are arranged.

[0019] In addition, the electronic control unit 24 is designed to regulate the delivery temperature TDLV by adapting the set point TSET and the temperature thresholds to the thermal load estimated.

[0020] The temperature thresholds comprise a lower threshold TLOW and an upper threshold THIG linked to the set point TSET on the basis of the relations:

$$\mathrm{THIG} = \mathrm{TSET} + \mathrm{DM}\cdot\mathrm{R}; \tag{1}$$

$$\mathrm{TLOW} = \mathrm{TSET} - \mathrm{DM}\cdot(1-\mathrm{R}), \tag{2}$$

where DM is a temperature differential between the delivery-temperature thresholds, hereinafter referred to more simply as "delivery differential", and R is a parameter, hereinafter referred to as "dividing ratio", comprised between 0 and 1 that defines the ratio with which the delivery differential DM is divided between the two thresholds so as to be able to render them asymmetrical with respect to the set point TSET.

[0021] As may be noted, corresponding to a regulation of the set point TSET, due to the adaptation to the thermal load, is a regulation of the same sign of the lower threshold TLOW and upper threshold THIG. Typically, in controls on the delivery temperature TDLV, the lower threshold TLOW and upper threshold THIG are maintained symmetrical with respect to the set point TSET; i.e., they are linked to the set point TSET via a dividing ratio R of 0.5. However, the lower threshold TLOW and upper threshold THIG can be adjusted irrespective of the set point TSET by acting on the delivery differential DM and on the dividing ratio R.

[0022] To a first approximation, the thermal load of the hydronic circuit 15 corresponds to the thermal load offered by the environment to be cooled. The more the thermal load decreases, the lower the heat exchange between the environment and the fan coils 2, and the less the need to cool the service fluid 5. Consequently, adaptation of the set point TSET envisages increase of the set point TSET as the thermal load decreases. Since the coefficient of performance (COP) of a refrigerating machine 3 increases as the evaporation temperature in the exchanger increases, it follows that the increase of the set point TSET as the thermal load decreases leads to an increase in the overall efficiency of the system 1.

[0023] The estimated thermal load is given in terms of fraction of load FL, i.e., in terms of ratio between the power that the refrigerating machine 3 must supply to cool the environment and the maximum refrigerating power that can be delivered by the refrigerating machine 3 in given nominal conditions.

[0024] Turning-on or turning-off of the compressor 12 must respect precise temporal constraints between successive events of turning-on and/or turning-off in order to safeguard the integrity of the compressor 12 itself, i.e., it must respect a minimum time Δt_ON_min between turning-on and turning-off, a minimum time Δt_OFF_min between turning-off and turning-on, and a minimum time Δt_ON_ON_min between two successive turning-on events. In order to respect said temporal constraints, the compressor 12 should, during an on-off cycle defined between two successive turning-on events, remain on and remain off, respectively, for a theoretical on time Δt_ON and a theoretical off time Δt OFF depending upon the estimated fraction of thermal load FL as follows:

$$\Delta t\_ON = \begin{cases} \Delta t\_ON\_min & FL < FL1 \\ \Delta t\_ON\_ON\_min \cdot FL & FL1 \le FL \le FL2 \\ \Delta t\_OFF\_min \cdot \dfrac{FL}{1-FL} & FL > FL2 \end{cases} \qquad (3)$$

$$\Delta t\_OFF = \begin{cases} \Delta t\_ON\_min \cdot \dfrac{1-FL}{FL} & FL < FL1 \\ \Delta t\_ON\_ON\_min \cdot (1-FL) & FL1 \le FL \le FL2 \\ \Delta t\_OFF\_min & FL > FL2 \end{cases} \qquad (4)$$

where: $FL1 = \dfrac{\Delta t\_ON\_min}{\Delta t\_ON\_ON\_min}$; $FL2 = \dfrac{\Delta t\_OFF\_min}{\Delta t\_ON\_ON\_min}$.

[0025] Figure 2 shows the graph of the theoretical on and off times Δt_ON and Δt_OFF as the fraction of load FL varies in the case of the following temporal constraints:

- Δt_ON_min = 60 s;
- Δt_OFF_min = 180 s; and
- Δt_ON_ON_min = 360 s.

[0026] The asymptotic evolutions at the extremes of the range of variation of the fraction of load FL mean that the compressor 12 remains on or off for extremely long times at points corresponding, respectively, to fractions of load FL close to 1 or 0.

[0027] Figure 3 illustrates a flowchart that describes the regulation of the delivery temperature TDLV as a function of the estimate of the thermal load.

[0028] With reference to Figure 3, upon turning-on of the refrigerating machine 3, the method envisages a step of initialization of variables (block 100), in which:

- the set point TSET is set to a minimum value TSETmin of 7°C, which corresponds to the value that the set point TSET ideally assumes at a point corresponding to the maximum load in such a way that the refrigerating machine 3 will immediately start cooling to maximum to cope with a possibly high initial load;
- the delivery differential DM is set to a default value DMdef of 4.8°C;
- the dividing ratio R is set to a default value Rdef of 0.5; and
- two variables t_ON and t_OFF, referred to hereinafter as "turning-on instant" and "turning-off instant", respectively, of the compressor 12, are set to 0; and
- a variable τ_WAIT, referred to hereinafter as "wait time", is set to 0.

[0029] The delivery temperature TDLV and the return temperature TRET are measured via the respective sensors 21 and 22 (block 101).

[0030] Periodically, a measurement of the delivery temperature TDLV is compared with the lower threshold TLOW (block 102) and the upper threshold THIG (block 103). If the delivery temperature TDLV is lower than or equal to the lower threshold TLOW, then the compressor 12 is turned off (block 104). Instead, if the delivery temperature TDLV is higher than or equal to the upper threshold THIG, then the compressor 12 is turned on (block 105). The instants of time of the events of turning-off and turning-on are stored in the respective variables turning-off instant t_OFF (block 106) and turning-on instant t_ON (block 107). In addition, at each turning-on event a counter N_ON for counting the number of turning-on events is incremented (block 108).

[0031] The turning-on event determines start of the on-off cycle of the compressor 12, and at a point corresponding to said event a series of calculations is triggered, which leads to the estimation of the thermal load and to the adaptation of the set point TSET and of the thresholds TLOW and THIG to the estimated thermal load.

[0032] In particular, on the basis of the turning-on instant t_ON and turning-off instant t_OFF, an effective on time Δt_ON_real, an effective off time Δt_OFF_real, and an effective cycle time Δt_TOT_real of the compressor 12 are determined, the latter being equal to the sum Δt_ON_ real + Δt_OFF_real (block 109), and a mean delivery temperature TDLVmean is determined by averaging the measurements of delivery temperature TDLV over the effective cycle time Δt_TOT_real (block 110).

[0033] In accordance with the present invention, the thermal load is estimated as a function of the measurements of delivery temperature TDLV and return temperature TRET and is supplied, as mentioned previously, in terms of an estimated fraction of load FL (block 111).

[0034] Once the estimation of the fraction of load FL has been performed, the theoretical on time Δt_ON and the theoretical off time t_OFF are calculated, respectively, via Eq. (3) and Eq. (4) and a theoretical cycle time Δt_TOT, equal to the sum Δt_ON + Δt_OFF, is calculated (block 112).

[0035] At this point, the set point TSET is adjusted via an adaptation thereof to the fraction of load FL (block 113). However, the adaptation of the set point TSET to the fraction of load FL is enabled only after verification of the fact that the number of turning-on events N_ON has reached a minimum number of turning-on events N_ON_min, preferably equal to 4 (block 114). This control has the purpose of enabling a sufficient stabilization of the process of estimation of the fraction of load FL in so far as the process of estimation is perturbed by the regulation of the set point TSET.

[0036] After the regulation of the set point TSET, the wait time τ_WAIT is set to a value calculated by applying the following formula:

$$\tau\_WAIT = 2.5 \cdot \Delta TSET \cdot (\Delta t\_TOT), \qquad (5)$$

where ΔTSET is a set-point step produced by the regulation of the set point TSET with respect to the previous value assumed by the set point TSET itself, as will be explained hereinafter, and a down count is activated starting from said value of wait time t_WAIT (block 115). Only at the end of said down count is the regulation of the set point TSET enabled again (block 116). Also this solution has the purpose of enabling a sufficient stabilization of the process of estimation of the fraction of load FL.

[0037] Next, the lower threshold TLOW and upper threshold THIG are adjusted as a function of the fraction of load FL, of the mean delivery temperature TDLVmean, and of the set point TSET (block 117).

[0038] Figure 4 illustrates a portion of flowchart regarding block 111 of Figure 3, which illustrates the substeps regarding determination of the fraction of load FL of the hydronic circuit 15 in accordance with the present invention.

[0039] The method is based upon the hypothesis that the system constituted by the air-conditioning system 1 and by the environment to be cooled is a thermally insulated system, for which it is possible to write an energy-balance equation in terms of temperature of the type:

$$\Delta TQ = TDLV - TRET + k \cdot \frac{dTRET}{dt} \qquad (6)$$

where ΔTQ is the difference of temperature between the inlet and the outlet of the set of fan coils 2 produced by the thermal power that the environment supplies to the system 1, and k is a parameter, hereinafter referred to as system parameter, which depends upon the characteristics of capacity and mass flowrate of the hydronic circuit 15, and in particular k = ρ.Vtot/m, where ρ is the density of the service fluid 5 expressed in kg/m^3, Vtot is the volume of the entire hydronic circuit 15 expressed in m^3, and m is the mass flowrate of the hydronic circuit 15 expressed in kg/s.

[0040] In particular, with reference to Figure 4, the method for estimating the thermal load envisages estimating the system parameter k to tune a subsequent estimation of the fraction of load FL with the characteristics of capacity and mass flowrate of the hydronic circuit 15 (block 200), and subsequently of acquiring samples of delivery temperature TDLV(n) and return temperature TRET(n) by sampling the outputs of the sensors 21 and 22 with a sampling period ts (block 201) and estimating the temperature difference ΔTQ by processing the samples of delivery temperature TDLV (n) and return temperature TRET(n) via a discrete Kalman filter (block 202) that expresses Eq. (6) as system in the discrete state space (DSS) according to the following matrix form:

$$\begin{cases} x(n+1) = F \cdot x(n) + G \cdot u(n) \\ y(n) = H \cdot x(n) + J \cdot u(n) \end{cases} \tag{7}$$

where u(n), x(n) and y(n) are, respectively, the vector of the inputs, of the states, and of the outputs of the system at the discrete instant n, and where

$$u(n) = \begin{bmatrix} TDLV(n) \\ TRET(n) \end{bmatrix}, \quad x(n) = y(n) = \begin{bmatrix} \Delta TQ(n) \\ TRET(n) \end{bmatrix} \tag{8}$$

$$F = \begin{bmatrix} 1 & 0 \\ \frac{ts}{k} & 1 - \frac{ts}{k} \end{bmatrix}, \quad G = \begin{bmatrix} 0 \\ \frac{ts}{k} \end{bmatrix}, \quad H = \begin{bmatrix} 0 & 1 \end{bmatrix}, \quad J = \begin{bmatrix} 0 \end{bmatrix}. \tag{9}$$

[0041] The transformation of Eq. (6) into the system defined by Eqs. (7), (8) and (9) is based upon the further hypothesis that the thermal power that the environment transfers to the system 1 is constant during the sampling period ts. Hence, the estimation of the difference of temperature ΔTQ is supplied in discrete form ΔTQ(n).

[0042] The difference of temperature ΔTQ estimated undergoes a low-pass filtering (block 203), for example via a first-order Chebyshev filter with a cut-off pulsation of 0.003 rad/s and a peak pass-band ripple of 3dBp, and is then processed to obtain a mean value ΔTQmean over the measured cycle time ΔtCYCLE (block 204).

[0043] Finally, there is calculated a mean value ΔTCHmean of a difference of temperature ΔTCH between the temperatures TDLV and TRET over the portion of the cycle time ΔtCYCLE corresponding to the period in which the compressor 12 is turned on in so far as the difference of temperature ΔTCH is zero when the compressor 12 is turned off (block 205), and the fraction of load FL sought is calculated as ratio between the mean values ΔTQmean and ΔTCHmean (block 206),

$$FL = \frac{\Delta TQmean}{\Delta TCHmean} \tag{10}$$

[0044] Eq. (10) follows directly from the definition of fraction of load given previously.

[0045] Figure 5 illustrates a portion of flowchart that describes in greater detail the step of estimation of the system parameter k indicated in block 200 in Figure 4.

[0046] The system parameter k is estimated on the basis of a formula obtained from an energy-balance equation in terms of temperature similar to Eq. (6) and expressed as a function of temperatures of which measurements are available, i.e., of the delivery temperature TDLV and of the return temperature TRET. Said formula has the following form:

$$k(t)=\frac{TDLV(t-\frac{\tau 1}{2})-TRET(t+\frac{\tau 1}{2})-TDLV(t+\tau 2)+TRET(t)}{\frac{dTDLV}{dt}(t+\tau 2)} \quad (11)$$

where $\tau 1$ and $\tau 2$ are the heat-propagation delays introduced, respectively, by the pipes of the hydronic circuit 15 and by the storage tank 19, assuming the outlet of the heat exchanger 11 as origin of a thermal variation, and t is an instant of time in which the system parameter k is estimated.

[0047] With reference to Figure 5, during each on-off cycle of the compressor 12, the effective on time At_ON_real and off time Δt_OFF_real are compared with one another (block 300): if the duration Δt_ON_real is greater than the duration Δt_OFF_real, then the system parameter k will be estimated just before turning-off of the compressor 12, namely, t = t_OFF - ε (block 301); otherwise, the system parameter k will be estimated just before turning-on of the compressor 12, namely, t = t_ON - ε (block 302). The deviation time ε is preferably five seconds.

[0048] The heat-propagation delay $\tau 2$ due to the storage tank 19 is determined as a difference between the turning-on instant t_ON and a first sing-inversion instant, in which the first derivative of the delivery temperature TDLV passes from a positive value to a negative value (block 303). By calculating, instead, the difference between the turning-on instant t_ON and a second sing-inversion instant, in which the first derivative of the return temperature TRET passes from a positive value to a negative value, there is obtained a heat-propagation delay $\tau 3$ for the entire hydronic circuit 15, i.e., between the outlet and the inlet of the exchanger (block 304). Hence, the heat-propagation delay $\tau 1$ due to the pipes is given by $\tau 1 = \tau 3 - \tau 2$ (block 305).

[0049] It is clear that, in the absence of the storage tank 19, we will not obtain the corresponding heat-propagation delay $\tau 2$, and the delay of propagation $\tau 1$ due to the pipes would be equal to $\tau 3$.

[0050] At this point, it is possible to apply Eq. (11) for calculating a value of the system parameter k at the instant of time t (block 306).

[0051] The estimation of the system parameter k in the way described above corresponds to an estimation of the characteristics of capacity and mass flowrate of the hydronic circuit 15 that enables automatic tuning of the estimation of the fraction of load FL to the characteristics of the system 1. This operation is certainly necessary upon initial turning-on of the refrigerating machine 3 after it has been connected to a new system 1, but also during normal operation of the system 1 itself for identifying variations of load due to de-activation of one or more fan coils 2.

[0052] Figure 6 illustrates a portion of flowchart that describes in greater detail the step of regulation of the set point TSET indicated in block 113 in Figure 3.

[0053] This step envisages calculation of a new value of the set point TSET using a formula that expresses the temperature of set point TSET as a function of the estimated fraction of load FL (block 400) :

$$TSET=\begin{cases} TSET\,max & if\ FL \le FLI \\ TSET\,min+(TSET\,max-TSET\,min)\cdot\frac{(1-FL)}{(1-FLI)} & otherwise \end{cases} \quad (12)$$

where TSETmin is the minimum value of the set point TSET corresponding to the fraction of maximum load equal to unity, TSETmax is a maximum value of the set point TSET corresponding to the fraction of load that is equal to zero, and FLI is a value of fraction of load that separates the relation into a first portion in which the set point TSET is constant and a second portion in which the set point TSET decreases linearly as the fraction of load FL varies.

[0054] Eq. (12) is considered for three different sets of values of the parameters TSETmin, TSETmax and FLI gathered together in the table appearing in Figure 8a. Figure 8b represents the three different versions of Eq. (12) via three respective curves plotted in the plane TSET, FLI and designated by C1, C2 and C3.

[0055] The three sets of parameters TSETmin, TSETmax and FLI, and hence the three curves C1, C2 and C3, correspond to three different modes of operation of the refrigerating machine 3 which can be selected by the user. Corresponding to the curve C1 is a default mode of operation, which ensures the best compromise between energy efficiency of the refrigerating machine 3 and precision of regulation of the delivery temperature TDLV, in so far as the new value of the set point TSET can vary between the minimum value TSETmin and the maximum value TSETmax for a wide range of values of fractions of load FL, i.e., from 0.3 to 1. Corresponding to the curve C2, instead, is a mode of operation that ensures the best energy efficiency, in so far as the new value of the set point TSET is equal to a high value (TSETmax) for a wide range of values of fractions of load FL, i.e., from 0 to 0.6, thus maximizing the coefficient of performance of the refrigerating machine 3. Finally, corresponding to the curve C3 is a mode of operation that ensures the best control of humidity in so far as the new value of the set point TSET is different for each value of fraction of load FL and can assume a maximum value TSETmax that is lower than that of the other curves C1 and C2.

[0056] After the new value of the set point TSET has been calculated, a set-point step ΔTSET is determined by calculating the difference between the value just calculated and the preceding value of the set point TSET and setting an upper limit for the step ΔTSET at a maximum value ΔTSETmax preferably of 4°C (block 401), and the set point TSET is updated by applying instantaneously the set-point step ΔTSET to the preceding set point TSET (block 402).

[0057] Figure 7 illustrates a portion of flowchart that describes in greater detail the step of regulation of the lower threshold TLOW and upper threshold THIG indicated in block 117 in Figure 3.

[0058] This step envisages calculation of a first error parameter Eδt, which defines the relative error between the effective cycle time Δt_TOT_real and the theoretical cycle time Δt_TOT (block 500), as follows:

$$E\Delta t = \frac{\Delta t\_TOT\_real - \Delta t\_TOT}{\Delta t\_TOT} \qquad (13)$$

[0059] If is found that the error EΔt is negative (block 501), it means that the delivery differential DM is too small, i.e., it does not enable respect of the temporal constraints between successive turning-on and/or turning-off events, and hence it is necessary to increase it by an amount ΔDM, preferably of 0.2°C (block 502); otherwise, a new value of delivery differential DM is determined in such a way that said error EΔt tends to zero (block 503). In particular, the new value of delivery differential DM is determined using a numeric method for search for the zeroes of a function, known as chord method or secant method, by applying it to a target function constituted by the error EΔt as a function of the delivery differential DM. This regulation has the purpose of maximizing the number of turning-on events of the compressor 12 per hour in due respect of the aforesaid temporal constraints.

[0060] In addition, a second error parameter ETDLV is calculated, which defines the relative error between the mean delivery temperature TDLVmean and the set point TSET (block 504),

$$ETDLV = \frac{TDLVmean - TSET}{TSET} \qquad (14)$$

and a new value of dividing ratio R is determined in such a way that said error ETDLV tends to zero (block 504). In a way similar to that adopted for the delivery differential DM, the chord method or secant method is used applying it to a target function constituted by the error ETDLV as a function of the dividing ratio R. This regulation has the purpose of speeding up convergence of the delivery temperature TDLV towards the set point TSET.

[0061] At this point, the lower threshold TLOW and the upper threshold THIG are calculated as a function of the dividing ratio R, of the delivery differential DM, and of the set point TSET applying Eq. (1) and Eq. (2), respectively (block 506).

[0062] The dependency of the theoretical cycle time Δt_TOT upon the fraction of load FL causes the delivery differential DM, and hence the position of the lower threshold TLOW and upper threshold THIG with respect to the set point TSET, to be in effect adapted to the fraction of load FL.

[0063] It should be noted that the working diagram of the refrigerating machine 3 illustrated in Figure 1 can generically describe also a machine designed to heat the service fluid 5 in order to heat the environments in which the fan coils 2 are set, for example, a refrigerating machine 3 of the type operating as heat pump. In said type of refrigerating machine 3, the compressor 12 is configured so as to carry out the refrigerating cycle in a reverse mode with respect to what was described previously, i.e., in such a way that the heat exchanger 11 functions as condenser for transferring heat from the working fluid 7 to the service fluid 5, and the heat exchanger 13 functions as evaporator.

[0064] The method for estimating the thermal load in accordance with the present invention is hence applicable also to the case where the refrigerating machine 3 is designed to heat the service fluid 5, it being sufficient simply to reverse the mechanism of some of the steps described and change the value of some parameters, and in particular:

- upon turning-on of the heat pump, the set point TSET is set to a maximum value TSETmax of 45°C, which corresponds to the value that the set point TSET assumes ideally at a point corresponding to the maximum thermal load in such a way that the heat pump will immediately start to heat to the maximum to cope with a possibly high initial load; in fact, there is an increase of efficiency as the condensation temperature decreases;

- if the measurement of the delivery temperature TDLV exceeds the upper threshold THIG, then the compressor 12 is turned off;
- if the measurement of the delivery temperature TDLV is below the lower threshold TLOW, then the compressor 12 is turned on;
- the set point TSET is decreased when the thermal load decreases, that is, the new set point TSET is expressed as a linear increasing function of the fraction of load FL of the type:

$$TSET = \begin{cases} TSET\,min & if\ FL \le FLI \\ TSET\,max + (TSET\,min - TSET\,max)\cdot \dfrac{(1-FL)}{(1-FLI)} & otherwise \end{cases} \qquad (15)$$

- Eq. (14) is considered for three different sets of values of the parameters TSETmin, TSETmax and FLI gathered in the table appearing in Figure 9a and generating the curves C1, C2 and C3 illustrated in Figure 9b;
- the maximum value ΔTSETmax at which the amplitude of regulation of the set point TSET is to be limited is preferably 5°C; and
- the default value DMdef of the differential DM is preferably 6°C.

[0065] The main advantage of the method for estimating the thermal load of a hydronic circuit 15 described above as compared to the known art is that it increases the overall efficiency of the system 1 albeit maintaining a good precision of regulation of the delivery temperature TDLV of the service fluid 5 in the hydronic circuit 15.

[0066] In fact, the adaptation of the set point TSET to the thermal load of the hydronic circuit 15 enables the refrigerating machine 3 to respond promptly to the variations of thermal load of the environment of which it is intended to control the temperature in such a way that the evaporation temperature can increase in the case where the machine is configured for cooling the service fluid 5, or in such a way that the condensation temperature can decrease in the case where the machine is configured for heating the service fluid 5, thus maximizing the coefficient of performance in all the operating conditions.

[0067] In addition, the adaptation of the two delivery-temperature thresholds TLOW and THIG to the estimated fraction of load FL enables maximization of the number of turning-on events of the compressor 12 per hour within the limit imposed by the temporal constraints between successive turning-on and/or turning-off events via regulation of the delivery differential DM and enables speeding up of the convergence of the delivery temperature TDLV to the set point TSET via the regulation of the dividing ratio R.

[0068] Another advantage is that it enables automatic adaptation of the refrigerating machine 3 to the type of system 1 in which it is installed and rapid identification of variations of thermal load due to de-activation of one or more fan coils 2, thanks to the operation of estimation of the system parameter k that expresses the characteristics of capacity and flowrate of the system 1.

[0069] Finally, the possibility of switching from regulation for a service fluid to be cooled to that for a service fluid to be heated by simply varying some parameters renders the method easy to implement in the electronic control unit 24 of any refrigerating machine 3 of a reversible type, i.e. one provided with an reverse valve arranged along the internal circuit 6 to be able to reverse the refrigerating cycle so as to enable an operation in cooling mode or in heating mode.

Claims

1. A method for estimating the thermal load of a service circuit (15) for a service fluid (5) at outlet from a refrigerating machine (3) of an air-conditioning system (1); the refrigerating machine (3) comprising a compressor (12), and the air-conditioning system comprising fan-coil means (2); the service circuit (15) comprising a delivery branch (16) for circulation of the service fluid (5) from the refrigerating machine (3) to the fan-coil means (2), and a return branch (17) for return of the service fluid (5) at input to the refrigerating machine (3); the method being **characterized in that** it comprises:

- measuring (101), via a pair of temperature sensors (21, 22), a delivery temperature (TDLV) of the service fluid (5) at outlet from the refrigerating machine (3) and a return temperature (TRET) of the service fluid (5) at inlet to the refrigerating machine (3); and

- supplying (106, 107, 109, 111) an estimate of the thermal load (FL) by processing, via a Kalman filtering, the measurements of the delivery temperature (TDLV) and of the return temperature (TRET).

2. The method according to Claim 1, wherein said estimate of the thermal load is supplied (106, 107, 109, 111) in terms of fraction of load (FL) referred to a maximum power that can be delivered by said refrigerating machine (3).

3. The method according to Claim 1 or Claim 2, wherein supplying (106, 107, 109, 111) an estimate of the thermal load (FL) comprises:

- tuning (106, 107, 109, 200) the estimate of the thermal load (FL) as a function of an estimate of characteristics of capacity and mass flowrate of said service circuit (15).

4. The method according to any one of the preceding claims, wherein supplying (106, 107, 109, 111) an estimate of the thermal load (FL) comprises:

- acquiring (201) samples of delivery temperature (TDLV(n)) and of return temperature (TRET(n)) according to a given sampling period (ts); and
- estimating (202) a first difference of temperature (ΔTQ) defined between the inlet and the outlet of the fan-coil means (2) by processing the samples of delivery temperature (TDLV (n)) and return temperature (TRET (n)) via a discrete Kalman filter that expresses an energy balance of said air-conditioning system (1) as a system in the discrete state space.

5. The method according to Claim 4, wherein supplying (106, 107, 109, 111) an estimate of the thermal load (FL) comprises:

- calculating (204-206) a fraction of thermal load (FL) referred to a maximum power that can be delivered by said refrigerating machine (3) and as a function of said first difference of temperature (ΔTQ) and of a second difference of temperature (ΔTCH) defined as a difference between said delivery temperature (TDLV) and said return temperature (TRET).

6. The method according to Claim 5, wherein calculating (204-206) a fraction of thermal load (FL) comprises:

- calculating (204, 205) a mean value (ΔTQmean) of said first difference of temperature (ΔTQ) and a mean value (ΔTCHmean) of said second difference of temperature (ΔTCH); and
- calculating (206) the fraction of load as ratio between the mean value (ΔTQmean) of the first difference of temperature (ΔTQ) and the mean value (ΔTCHmean) of the second difference of temperature (ΔTCH).

7. The method according to Claim 6, wherein said mean value (ΔTQmean) of said first difference of temperature (ΔTQ) is calculated over an on-off cycle of said compressor (12), and said mean value (ΔTCHmean) of said second difference of temperature (ΔTCH) is calculated over the portion of said cycle where the compressor (12) is turned on.

8. The method according to any one of Claims 4 to 7, wherein supplying (106, 107, 109, 111) an estimate of the thermal load (FL) comprises:

- carrying out (203) a low-pass filtering of said first estimated difference of temperature (ΔTQ).

9. The method according to any one of Claims 4 to 8, wherein said discrete Kalman filter has: input variables comprising a discrete series constituted by said samples of delivery temperature (TDLV(n)) and a discrete series constituted by said samples of return temperature (TRET(n)); output variables coinciding with the state variable and comprising the series of samples of return temperature (TRET(n)) and a series of values of first difference of temperature (ΔTQ (n)); and matrices for processing the input and state variables, the elements of which are a function of a system parameter (k) that depends upon characteristics of capacity and mass flowrate of said service circuit (15).

10. The method according to any one of Claims 3 to 9, wherein tuning the estimate of the thermal load (FL) as a function of an estimate of characteristics of capacity and mass flowrate of said service circuit (15) comprises:

- estimating (106, 107, 109, 200) a system parameter (k) proportional to a density (ρ) of the service fluid (5), proportional to a total volume (Vtot) of said service circuit (15), and inversely proportional to a mass flowrate

(m) of said service circuit (15).

11. The method according to Claim 10, wherein estimating (106, 107, 109, 200) the system parameter (k) comprises:

- determining (109), for each on-off cycle of said compressor (12), an effective on time (Δt_ON_real) and an effective off time (Δt_OFF_real) of the compressor (12);
- determining (300-302) an estimation instant (t) as a function of the result of a comparison between the effective on time (Δt_ON_real) and the effective off time (Δt_OFF_real); and
- calculating (306) a value of the system parameter (k) at the estimation instant (t).

12. The method according to Claim 10 or Claim 11, wherein estimating (106, 107, 109, 200) the system parameter (k) comprises:

- determining (303-305) heat-propagation delays (τ1, τ2, τ3) introduced by said service circuit (15);
- calculating (306) a value of the system parameter (k) as a function of said delivery temperature (TDLV), of a return temperature (TRET) of said service fluid (5) at inlet to the refrigerating machine (3), and of the heat-propagation delays (τ1, τ2, τ3).

13. The method according to any one of Claims 10 to 12, wherein estimating (106, 107, 109, 200) the system parameter (k) comprises:

- storing (106, 107), for each on-off cycle of said compressor (12), a turning-on instant (t_ON) and a turning-off instant (t_OFF) of the compressor (12); determining (300-302) an estimation instant (t) comprising:
- in the case where said effective on time (Δt_ON_real) is longer than said effective off time (Δt_OFF_real), calculating (301) the estimation instant (t) as a difference between the turning-off instant (t_OFF) and a given deviation time (ε);
- otherwise, calculating (302) the estimation instant (t) as a difference between the turning-on instant (t_ON) and the deviation time (ε).

14. The method according to Claim 12 or Claim 13, wherein estimating (106, 107, 109, 200) said system parameter (k) comprises:

- storing (107), for each on-off cycle of said compressor (12), a turning-on instant (t_ON) of the compressor (12); determining (303-305) heat-propagation delays (τ1, τ2, τ3) introduced by said service circuit (15) comprising:
- determining (307) a first heat-propagation delay (τ3) as a function of said turning-on instant (t ON) of the compressor (12) and of a first instant in which the first derivative of said return temperature (TRET) passes from a positive value to a negative value.

15. The method according to Claim 14, wherein said refrigerating machine (3) comprises means for storage (19) of said service fluid (5) set along a delivery branch (16) of said service circuit (15) for producing a thermal inertia; determining (303-305) heat-propagation delays (τ1, τ2, τ3) introduced by said service circuit (15) comprising:

- determining (305) a second heat-propagation delay (τ2) as a function of said turning-on instant (t_ON) of the compressor (12) and of a second instant in which the first derivative of said delivery temperature (TRET) passes from a positive value to a negative value; and
- determining (305) a third heat-propagation delay (τ1) as a function of the first heat-propagation delay (τ3) and of the second heat-propagation delay (τ2).

16. The method according to any one of the preceding claims, comprising:

- turning on and turning off (102-105) said compressor (12) as a function of a measurement of the delivery temperature (TDLV) in such a way that the delivery temperature (TDLV) itself will converge to a delivery-temperature set point (TSET); and
- using said estimate of the thermal load (FL) for regulating said delivery temperature (TDLV) by adapting the delivery-temperature set point (TSET) to the estimate of the thermal load (FL) itself.

17. The method according to any one of the preceding claims, comprising:

- comparing (102, 103) a measurement of said delivery temperature (TDLV) with a pair of delivery-temperature thresholds (TLOW, THIG) set in a given way with respect to a delivery-temperature set point (TSET); and
- using said estimate of the thermal load (FL) for regulating said delivery temperature (TDLV) by adapting the position of the thresholds (TLOW, THIG), with respect to said set point (TSET), to the estimate of the thermal load (FL) itself.

18. A control device for a refrigerating machine (3), the control device (20) comprising first temperature-sensor means (21) for measuring a delivery temperature (TDLV) of a service fluid (5) at outlet from the refrigerating machine (3) and a control unit (24) designed to control the refrigerating machine (3) in such a way that the delivery temperature (TDLV) will converge to a set point (TSET), and being **characterized in that** it comprises second temperature-sensor means (22) for measuring a return temperature (TRET) of the service fluid (5) at inlet to the refrigerating machine (3) and **in that** the control unit (24) implements the method according to any one of Claims 1 to 17.

19. A refrigerating machine (3) comprising a compressor (12) and a control device (20) for turning on and turning off the compressor (12) as a function of a measurement of delivery temperature (TDLV) of a service fluid (5) at outlet from the refrigerating machine (3), and **characterized in that** the control device (20) is of the type claimed in Claim 18.

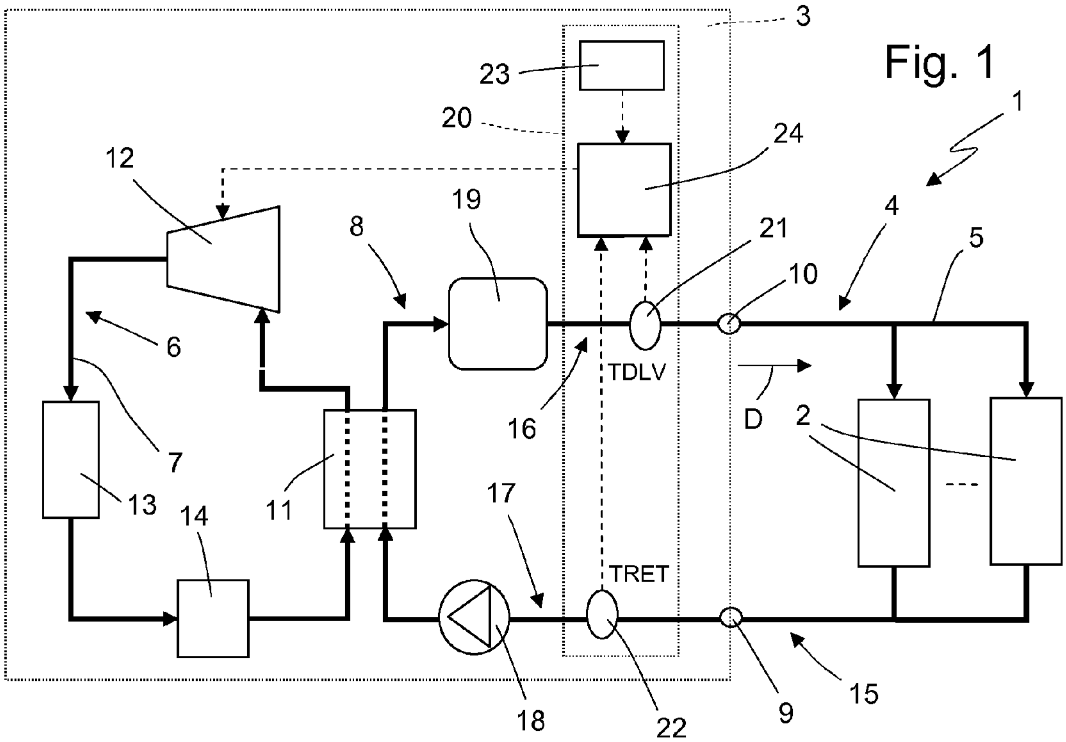
3
23
20
24
1
12
19
8
21
4
5
10
6
TDLV
16
D
2
7
13
14
11
17
TRET
18
22
9
15

Fig. 1

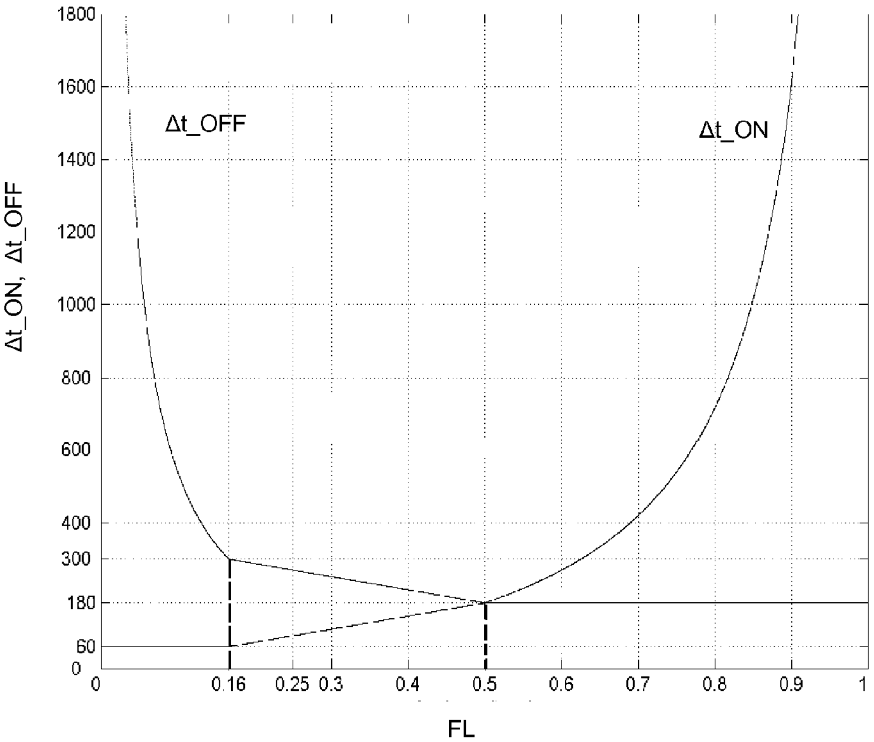
1800
1600
1400
1200
1000
800
600
400
300
180
60
0
Δt_OFF
Δt_ON
Δt_ON, Δt_OFF
0
0.16
0.25
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1
FL

Fig. 2

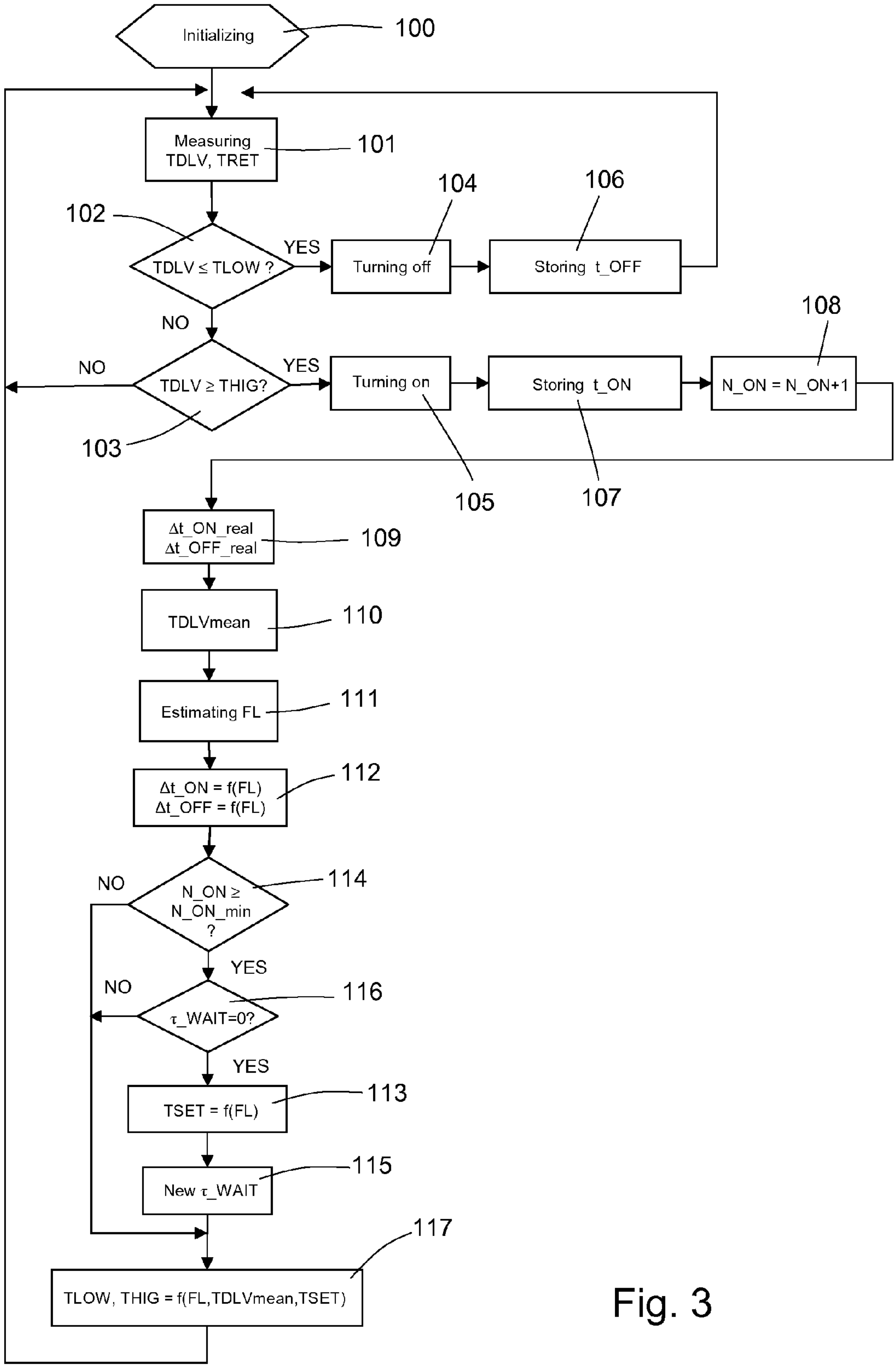
Initializing
100
Measuring TDLV, TRET
101
102
TDLV ≤ TLOW ?
YES
NO
Turning off
104
Storing t_OFF
106
TDLV ≥ THIG?
103
YES
NO
Turning on
105
Storing t_ON
107
N_ON = N_ON+1
108
Δt_ON_real
Δt_OFF_real
109
TDLVmean
110
Estimating FL
111
Δt_ON = f(FL)
Δt_OFF = f(FL)
112
N_ON ≥ N_ON_min ?
114
NO
YES
τ_WAIT=0?
116
NO
YES
TSET = f(FL)
113
New τ_WAIT
115
TLOW, THIG = f(FL,TDLVmean,TSET)
117

Fig. 3

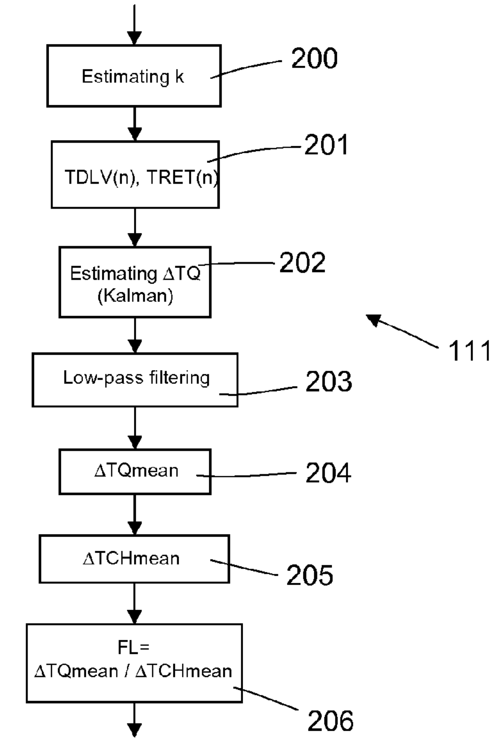

Fig. 4

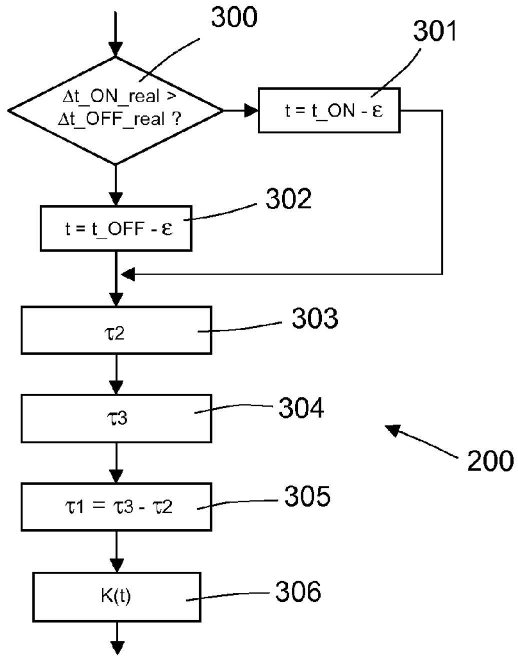

Fig. 5

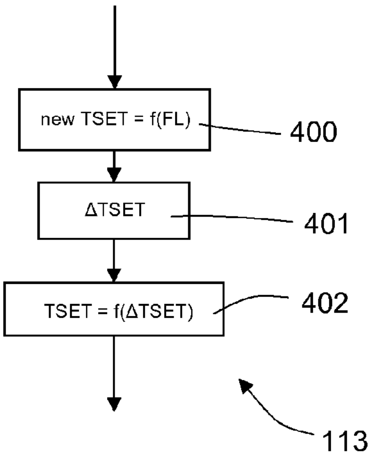

Fig. 6

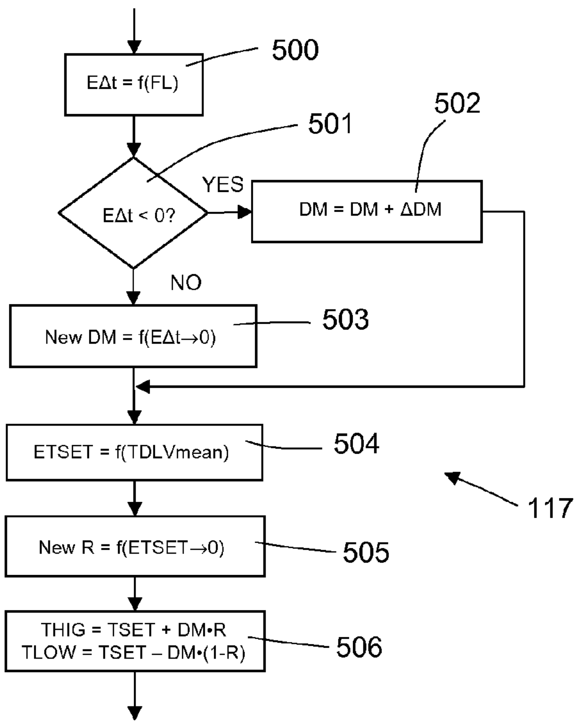

Fig. 7

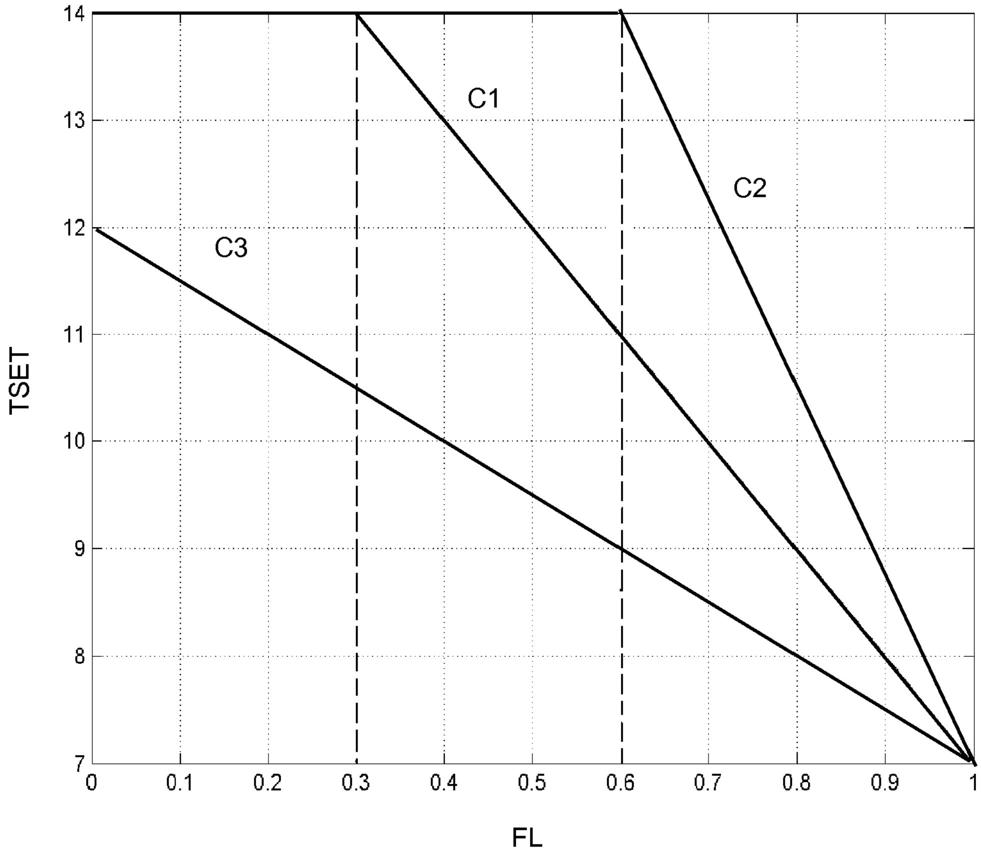

Fig. 8a

curve	TSETmin (°C)	TSETmax (°C)	FLI
C1	7	14	0,3
C2	7	14	0,6
C3	7	12	0

Fig. 8b

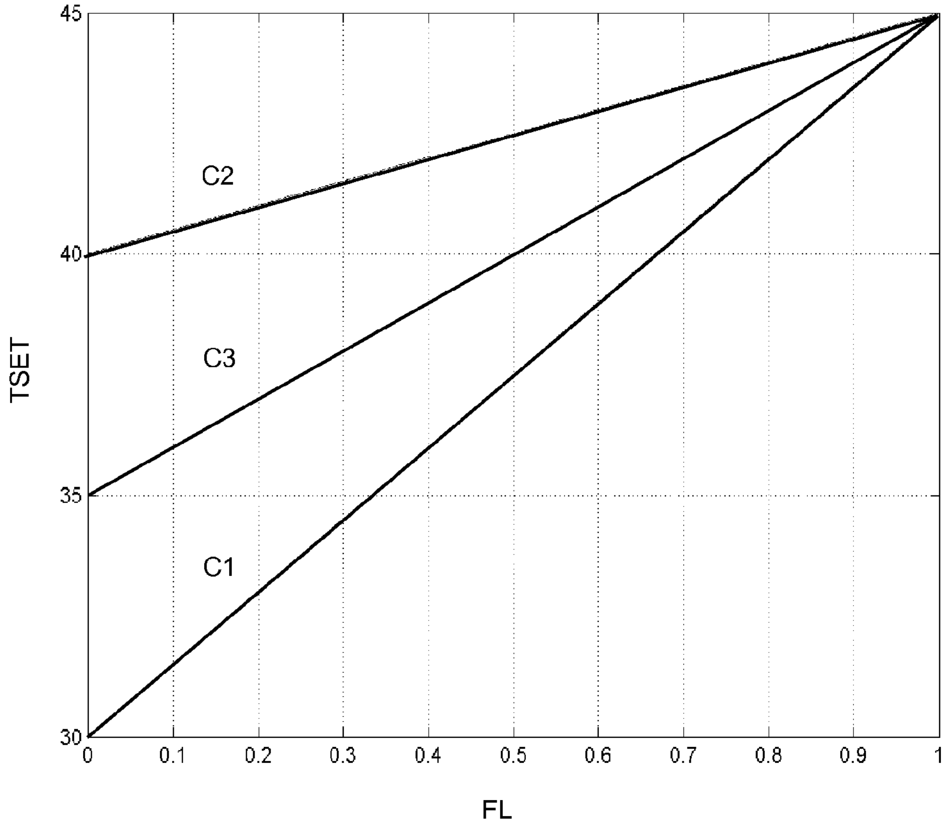

Fig. 9a

curve	TSETmin (°C)	TSETmax (°C)	FLI
C1	30	45	0
C2	40	45	0
C3	35	45	0

Fig. 9b