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(54) **OPTIMIZED MANAGEMENT METHOD OF AN ENVIRONMENTALLY FRIENDLY HEAT PUMP**

OPTIMIERTES VERWALTUNGSVERFAHREN EINER UMWELTFREUNDLICHEN WÄRMEPUMPE

PROCÉDÉ DE GESTION OPTIMISÉ D'UNE POMPE À CHALEUR ÉCOLOGIQUE

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Description

[0001] The present invention relates to a method for the management and control of a thermodynamic machine.

[0002] The object of the present invention is a thermodynamic machine, for example a heat pump, of a conditioning apparatus of a residential and/or industrial environment, based on a compression/expansion thermodynamic cycle of a low environmental impact operating fluid and capable of ensuring optimal operating conditions and maximum efficiency and performance.

[0003] More precisely, the object of the present invention is a management method or logic of said heat pump, able to ensure optimal operating and performance conditions and preserve the functionality of the mechanical components thereof, in particular of the compressor thereof.

[0004] Even more precisely, the object of the present invention is a management method or logic of a heat pump able to optimising the temperature of a low environmental impact operating fluid at the compressor discharge (hereinafter referred to as the "delivery temperature" of the compressor), so as to ensure the maximum reliability thereof (i.e., eliminating any risk of breakage and malfunction) and ensuring the same operating range (*or envelope*) of said conditioning apparatus with refrigerants having a higher GWP.

[0005] In particular, the invention falls within the sector of heat pump conditioning apparatuses for residential and/or industrial environments (or similar scopes), where "conditioning" is indifferently referred to as "heating" or "cooling", preferably made by electrical energy.

[0006] As known, the conditioning of a building is obtained through the use of thermodynamic apparatuses and systems that comprise at least one thermodynamic machine configured to heat or cool a heat transfer fluid (e.g. water or air) intended to reach, through specific devices and/or distribution circuits, the various rooms of said building in order to release therein part of the heat energy thereof or draw it from the same.

[0007] Known thermodynamic machines are, for example, the so-called heat pumps (hereinafter also abbreviated with the acronym HP) in which an operating fluid, that circulates in a refrigerant circuit, is evaporated at low temperature, brought to high pressure, condensed and finally brought back to the evaporation pressure. Said heat pumps therefore comprise:

- at least one first heat exchanger in which the operating fluid absorbs, at constant pressure, heat energy from a first fluid F.f that is at a first temperature T.f,
- at least a second heat exchanger, in which the same operating fluid yields, at constant pressure, part of the heat energy thereof to a second fluid F.c that is at a second temperature T.c > T.f,
- a compressor driven by a motor and designed to compress said operating fluid between a minimum pressure thereof, that it has at the outlet of the first exchanger, to the maximum pressure that it has at the inlet of the second exchanger,
- a lamination valve that achieves an expansion, at a substantially constant enthalpy, and a cooling of the operating fluid.

[0008] Said first heat transfer fluid F.f from which heat is drawn is referred to as "*cold well*" while the second heat transfer fluid F.c to which heat is yielded is also known with the term of "*hot well*".

[0009] Heat pumps where the cold well consists of air and the hot well consists of water are referred to as "air-water" (or vice versa "water-air") heat pumps.

[0010] The refrigerant circuit of the aforementioned heat pump, as known, may be switched between a "cooling" and a "heating" operating mode (and vice versa) with said first and second heat exchanger that may therefore operate, if necessary, either as a condenser or as an evaporator.

[0011] What has been said so far is visually shown in the *p-h* (*pressure-enthalpy*) *diagram* of Fig. 1 showing a typical *A-B-C-D* refrigerant expansion/compression refrigeration cycle of a refrigerant, e.g. of the well-known R410A gas, in which:

- section A-B represents the compression step of the refrigerant coming from the evaporator, said refrigerant being generally discharged from the compressor in the form of overheated vapour with a pressure and a corresponding temperature, hereinafter respectively referred to as delivery pressure and temperature,
- section B-C represents the subsequent cooling and isobaric condensation step of the refrigerant during which it dissipates the heat thereof through a condenser by passing from an overheated vapour state to a saturated or subcooled liquid state,
- section C-D represents the decompression of the same refrigerant through the lamination or expansion valve so as to have at the inlet of the evaporator a refrigerant in subcooled or saturated liquid or preferably biphasic liquid-vapour conditions (as in the example in Fig. 1 - point D),
- section D-A represents the isobaric evaporation of the refrigerant in the evaporator up to an overheated degree greater than or equal to zero so as to have overheated (point A in Fig. 1) or saturated (point A of Fig. 1) vapour respectively at the compressor suction.

[0012] Increasingly stringent regulations on said heat machines and related refrigerant circuits in environmental matters are progressively requiring the use of low environmental impact refrigerant fluids (also known as reduced or low GWP - *Global Warming Potential*) refrigerants.

[0013] For example, since 2015 in Europe, a new regulation came into force, known as the "*F-GAS Certification* ", that requires to progressively reduce the use of those refrigerant gases that significantly contribute to the Earth's greenhouse effect and to the consequent global warming.

[0014] Such regulation provides that by 2030 the "equivalent CO2" (a measure that expresses the impact on global warming of a certain amount of "greenhouse-gas" compared to the same amount of "carbon dioxide") currently attributable to greenhouse or polluting refrigerant gases is reduced by 80%.

[0015] Many companies and manufacturers of heat pumps or similar air conditioning devices are therefore replacing the high greenhouse effect "traditional" refrigerant gases (e.g. the aforementioned R410A) with less polluting operating fluids.

[0016] For example, the use of a reduced GWP refrigerant gas, known as R32, belonging to the group of hydrofluorocarbonsand consisting of a difluoromethane (chemical formula: CH_2F_2) has been found very advantageous.

[0017] Such refrigerant (or other similar ones belonging to the same family or similar groups), although having a low environmental impact, is not free from drawbacks.

[0018] In particular, as shown in Fig. 4a, assuming a suction thereof to the compressor in a state of saturated or overheated vapour, R32 (or similar/equivalent refrigerants) has the drawback, compared to the refrigerants (R410A) most commonly used so far with which, in the graph in figure, is compared, of significantly increasing the delivery temperature of the heat pump compressor (obviously, the same other operating conditions being equal such as, for example, the condensation and evaporation temperatures).

[0019] There is therefore the risk that the compressor delivery temperatures resulting from the compression of a low environmental impact refrigerant may approach and sometimes exceed the maximum limit set by the compressor operator with negative effects both on the various mechanical components of the compressor and on the chemical-physical features of the lubricating oil present therein for the lubrication of the moving parts.

[0020] In fact, it is known that too high delivery temperatures may correspond to undesirable overheating of the electric motor of the compressor, and to an impairment of the lubricating properties of the oil with inevitable risks of breakdowns and malfunctions.

[0021] To maintain the delivery temperature substantially equal to that of the traditional refrigerants and avoid the aforementioned drawbacks, it is known to limit the minimum evaporation temperature at equal condensing temperature (in this regard, see Fig. 4c) or, vice versa, to limit the maximum condensing temperature at equal evaporation (Fig. 4b) or, finally, to implement a combination between the two limitations of the evaporation and condensation temperature; in all these cases, however, there is therefore a significant reduction in the operating range of the heat pump compared to that ensured by the traditional refrigerants used so far, such as R410A.

[0022] Over the last few years, some solutions have therefore been studied in order to "optimise" the delivery temperature of low environmental impact refrigerants, without deteriorating the efficiency of the compressor and/or the performance of the refrigeration cycle.

[0023] For example, in the field of the cooling/heating machines and apparatuses, the so-called "EVI" *(Enhanced Vapour Injection)* technology has been developed consisting of the injection of vapour in an intermediate stage of the compression process so as to ensure the achievement of a double benefit:

- an increase in the heating capacity, the displacement of the compressor being the same, and
- a desired reduction in the compressor delivery temperature.

[0024] Such technology provides that some liquid refrigerant, extracted from the highpressure side of the refrigeration cycle, is by-passed towards the compressor by means of a conduit whereon at least one expansion valve and a heat exchanger, generally a plate heat exchanger, are inserted, that works as a sub-cooler or economiser.

[0025] Along such bypass, the liquid refrigerant switches to the form of overheated vapour to be injected into the compressor substantially in the middle of the compression process thereof (cycle not shown in the accompanying figures).

[0026] This involves a reduction in the enthalpy of the refrigerant in the compression step and therefore the compressor delivery temperature.

[0027] It is however evident that such EVI technology, although efficient, leads to a greater constructive complexity of the heat pump and therefore higher production and marketing costs of the same and set up and management difficulties.

[0028] Alternatively, it is also known from the scientific literature how the optimisation (in particular a reduction thereof) of the compressor delivery temperature may be obtained through the suction to the compressor, and the consequent compression, of a refrigerant in a liquid-vapour biphasic state (point A" of Fig. 2 or 3).

[0029] More precisely, it has been observed how the compressor delivery temperature decreases as the humid fraction of the refrigerant entering the compressor increases and how this may be managed by regulating the opening degree of the expansion valve of the refrigeration cycle.

[0030] Document JP6594698 discloses, a method according to the preamble of claim 1 and for example, an air refrigerating/conditioning device comprising a refrigerating cycle using refrigerant R32 and means for controlling the opening degree of the electronic expansion valve to have a predetermined overheating degree at the delivery of the compressor.

[0031] However, even the regulation of said expansion valve has not proved to be free from drawbacks.

[0032] More precisely, there is a risk that the compressor delivery temperature is excessively reduced, e.g. up to below the condensation temperature of the refrigerant, with the consequent condensation thereof in the oil inside the compressor.

[0033] It is known how a condensation of the refrigerant in the compressor oil leads to dilution and impairment of the lubricating properties thereof.

[0034] This is strongly felt in rotary compressors, such as for example those of the *"High Side "* type, where the oil plays a primary function in ensuring the correct lubrication of the moving parts.

[0035] As schematically shown by way of an example in Fig. 6, this type of compressors, widely used in the heat pumps, is in fact characterised by one or more compression chambers C2 of the refrigerant, (in the example in figure two chambers), set in rotation, in opposing phase, by an electric motor C4 and completely immersed in the lubricating oil contained in the lower part of the compressor body C1, also known as oil sump C3.

[0036] Once compressed, the refrigerant discharged from one or more compression chambers C2 at the delivery temperature is therefore forced to lap and/or cross the lubricating oil before rising up the entire body C1 of the compressor C, cool the electric motor C4 and reach the outlet and connecting pipe C5 towards a heat exchanger placed downstream (the condenser of the refrigeration cycle). It is therefore clear that due to such direct interaction, the risk of dilution of the oil by the refrigerant is particularly high and harmful.

[0037] The purpose of the present invention is to provide an innovative control and management logic for a heat pump, for example of a conditioning apparatus in a residential and/or industrial environment, based on a compression/expansion thermodynamic cycle of a low environmental impact (GWP) operating fluid that obviates such kind of drawbacks.

[0038] More precisely, the object of the present invention is to provide, according to one or more variants, a management logic of said heat pump able to ensure optimal operating and performance conditions and to preserve the functionality and duration of the mechanical components thereof, in detail of the compressor thereof.

[0039] Even more precisely, the object of the present invention, at least in a preferred variant thereof, is to indicate a management method for a heat pump able to optimise the temperature of a low environmental impact (GWP) operating fluid to the compressor discharge, without compromising the operating range (*or envelope)* of said heat pump and the reliability of the same compressor.

[0040] A further object of the present invention, at least in a preferred variant thereof, is to provide a method/logic for managing a thermodynamic machine, for example a heat pump, able to ensure optimum performance under any load condition of the same thermodynamic machine and/or rotation frequency of the compressor. These and other objects, that shall become clear later, are achieved with method in accordance with the provisions of the independent claim.

[0041] Other objects may also be achieved by means of the additional features of the dependent claims.

[0042] Further features of the present invention shall be better highlighted by the following description of a preferred embodiment, in accordance with the patent claims and illustrated, purely by way of a non-limiting example, in the annexed drawing tables, wherein:

- Fig. 1 shows on a diagram P-h a diagram of a known compression/expansion refrigeration cycle of an operating fluid;
- Fig. 2 shows on a diagram P-h a diagram of a known compression/expansion refrigeration cycle of an operating fluid compared with the refrigeration cycle according to the invention;
- Fig. 3 shows on a diagram P-h, in more detail, the refrigeration cycles of Fig. 1 and 2 compared to a further standard refrigeration cycle for the same operating fluid;
- Figs. 4a-4c show on a diagram T-s a comparison between a compression/expansion refrigeration cycle of a traditional operating fluid (e.g. R410A) and a similar compression/expansion refrigeration cycle of a low environmental impact (GWP) operating fluid;
- Fig. 5 schematically and symbolically represents a heat pump of a typical conditioning apparatus (in heating mode) capable of implementing the refrigeration cycle of the previous figures;
- Fig. 6 schematically shows a "simplified" view of a *"High Side "* compressor of the heat pump of Fig. 5;
- Fig. 7 schematically shows a "simplified" view of a *"High Side "* compressor of the heat pump of Fig. 5 according to a possible variant of the invention.

[0043] The features of a preferred variant of the apparatus are now described for the conditioning of a residential and/or industrial environment and the related management logic according to the invention are now described, using the references contained in the figures.

[0044] It is noted that any dimensional and spatial term (such as "lower", "upper", "inner", "outer", "upstream", "downstream" and the like) refers to the positions of the elements as shown in the annexed figures, without any limiting intent relative to the possible operating conditions.

[0045] In the present description, by conditioning apparatus is intended a thermodynamic machine set up for the heating and/or "cooling" of a residential, industrial or similar environment.

[0046] Without any limiting intent, reference shall be made to heat pumps, preferably of the air-water type, although everything that will be said with reference thereto may be extended to any other type of heat pumps, e.g. of the water-water or air-air type, or similar/equivalent heat machines.

[0047] Fig. 5 therefore shows the diagram of a heat pump HP, preferably reversible for ambient cooling and/or heating (but for simplicity herein shown in heating mode), wherein an expansion/compression refrigeration cycle of an operating fluid, hereinafter simply referred to as "refrigerant", is realised.

[0048] As already mentioned in part, said pump HP comprises, connected to each other by means of suitable pipes 10, at least:

- a first heat exchanger 11, 12 wherein the refrigerant absorbs, at constant pressure, heat energy from a first fluid F.f, that is at a first temperature T.f and that defines the so-called *"cold well*",
- a second heat exchanger 12, 11 wherein the same refrigerant yields, at constant pressure, part of the heat energy thereof to a second fluid F.c, that is at a first temperature T.c > T.f and that corresponds to the so-called *"hot well",*
- a compressor 13, compatible to receive at suction and compress a refrigerant fluid comprising a certain percentage of wet fraction (i.e., at least in part in the liquid state), preferably of the *rotary type,* of the *"High Side "* type, driven by an electric motor and adapted to compress said refrigerant between a minimum pressure thereof that it has at the outlet of the first exchanger 11, 12 and the maximum pressure thereof that it has at the inlet of the second exchanger 12, 11,
- an expansion valve 14, placed between said first 11, 12 and second 12, 11 heat exchanger, that makes a constant enthalpy expansion and a cooling of the refrigerant.

[0049] Reference 15 also denotes a switch valve, e.g. "four-ways", that enables to convert the operation of the heat pump HP between a "cooling" mode and a "heating" mode (or vice versa).

[0050] When in "heating" mode, the refrigerant dissipates heat in the second exchanger, that therefore acts as a condenser 12, while it evaporates in the first exchanger that acts as an evaporator 11.

[0051] On the contrary, in "cooling" mode, the aforementioned first heat exchanger is the condenser 11 of the refrigerant circuit, the second exchanger the relative evaporator 12.

[0052] More precisely, therefore, the exchanger 12 is that where the heat transfer fluid intended for a user is heated or cooled, while the exchanger 11 is the one cooperating with the well where the heat yielded or subtracted from said user is absorbed or disposed of.

[0053] For descriptive simplicity, hereinafter, explicit reference will be made to a heat pump HP in "heating" mode (to which, as already said, Fig. 5 refers to without any limiting intent), although all that will be said with reference to such operating mode, may be also extended to "cooling", the aforementioned inversion of the refrigeration cycle operated by the switch valve 15 being known. Furthermore, in the example of Figure 5, reference shall be made to an air-water heat pump HP the cold well whereof F.f is the air of the environment in which it is installed while the relative hot well F.c is preferably water circulating in a specific distribution circuit for room heating.

[0054] Of course, nothing prevents said hot well from consisting of water contained inside a storage and intended for hygienic-sanitary uses.

[0055] The refrigerant circuit is then completed by at least one fan 16 that moves the air F.f through the evaporator 11 while the compressor 13 may be equipped with an accumulator 17 placed upstream of the suction section thereof and adapted to prevent, as is known, refrigerant excesses, oil or impurities therein.

[0056] A second known refrigerant accumulator 18 (referred to as "*liquid receiver*") may be provided in the proximity of the expansion valve 14 in order to compensate for any differences or variations in the levels and quantities of said refrigerant between the condenser and the evaporator.

[0057] For the purposes of the invention, a plurality of temperature sensors is also present along the refrigeration circuit.

[0058] In particular, it is envisaged:

- at least one temperature sensor T.com at the outlet of the same compressor 13 for the detection of the delivery temperature Tm thereof,
- at least one temperature sensor T.evap at the evaporator 11 for the detection of an evaporation temperature "SST",
- at least one temperature sensor T.cond at the condenser 12 for the detection of a condensation temperature "SDT".

[0059] Preferably, further temperature sensors T.f.c and T.f.f may also be provided for the measurement of the temperatures of hot well T.a and cold well T.w temperatures.

[0060] It is clear that said temperature sensors, at least those placed at the evaporator 11 and condenser 12, may be replaced by corresponding pressure sensors, given the known correlation between pressures and temperatures of a refrigerant fluid in phase-change.

[0061] It is equally known that changes in the environmental conditions in which the heat pump HP operates, e.g. of the temperatures T.c, T.f, of the relative hot and cold wells, affect the high and low pressure and/or temperature values of the refrigeration cycle and therefore lead to changes in the operating conditions of said heat pump HP.

[0062] According to the invention, the heat pump HP is configured and managed in such a way as to control the wet fraction (or percentage) of the refrigerant at the inlet of the compressor 13 by adjusting the evaporative power of the evaporator 11 and in such a way that the temperature difference between the lubricating oil of the compressor 13 and the operating fluid (refrigerant) at the delivery of the same compressor 13, is kept at least equal to or above a safety (or threshold) value such that there is no condensation of said operating fluid in said lubricant oil, thus avoiding as a consequence the dilution and the loss of the optimal chemical-physical properties thereof.

[0063] In other words, the temperature Toil of the lubricating oil should be always higher than the temperature Tm of the operating fluid at the delivery of the compressor 13 by at least one appropriate margin defined by a safety threshold OIL_SH; i.e. the following relationship is wished to be verified:

$$\mathrm{Toil} - \mathrm{Tm} \geq \mathrm{OIL\_SH} \qquad (1)$$

where said safety threshold OIL_SH (that shall be referred to over the course of the present description) is:

- that avoiding condensation of the refrigerant in the lubricating oil that is too cold due to any heat losses of the compressor and/or the too low temperature of the same operating fluid, said factors leading to an excessive cooling of said oil,
- suggested or set by the manufacturer of the compressor or by the compressor installer,
- it is preferably a value comprised between 5°C and 10°C, for example advantageously equal to 7°C (such value hereinafter being also referred to as OIL_SH_opt).

[0064] As it shall be seen more precisely below, what has just been said above (i.e., the satisfaction of the relationship (1)) is obtained by suitably controlling and regulating the delivery temperature Tm of the compressor 13 by acting on the aforementioned expansion valve 14.

[0065] This does not mean that in certain cases it is possible, alternatively or in combination, to directly heat said lubricating oil of the compressor 13.

[0066] According to a first preferred variant of the invention, therefore, the wet fraction of the refrigerant at the compressor suction 13 is increased or decreased by regulating the opening degree of the expansion valve 14, placed upstream of the evaporator 11.

[0067] It is in fact known that an increase in the opening degree of the expansion valve 14 corresponds to, at the evaporator inlet 11, an increase in the evaporation pressure and a greater quantity of refrigerant in the liquid state; this increases the amount of refrigerant that may not be evaporated by the evaporator 11 and therefore the wet fraction of the same entering the compressor 13.

[0068] On the contrary, a greater closure of the expansion valve 14 will result in a reduction in the evaporation pressure at the inlet of the evaporator 11, a lower amount of liquid refrigerant to be evaporated and therefore a lower wet fraction at the inlet of the compressor 13.

[0069] It is also known that the value of the delivery temperature Tm thereof depends directly on the percentage of the wet fraction at the inlet of the compressor 13.

[0070] For clarity of description, it is obvious that said "delivery temperature", generically indicated with the reference Tm, is the temperature "read/measured" at point B, B' ..... B^i of the refrigeration cycle (see Figures 1-3 attached to the description), i.e. at the outlet of one or more compression chambers of the compressor 13 (see, for example, Fig. 6).

[0071] In particular, it is known that said delivery temperature Tm decreases as the percentage of wet fraction of the refrigerant sucked by the compressor 13 increases. This is clearly shown in Fig. 3 where points B and B' define the delivery temperatures (with Tm_B > Tm_B') following the compression, respectively, of a refrigerant in the saturated vapour state (point A') and of a wet refrigerant (point A").

[0072] According to the invention, the delivery temperature Tm of the compressor 13 is therefore regulated and determined by regulating the wet fraction of the refrigerant to be compressed.

[0073] More precisely, the heat pump HP of the invention is configured to control the percentage of wet fraction of the refrigerant entering the compressor 13 in such a way as to make the aforementioned delivery temperature Tm equal to an "optimal" delivery temperature, hereinafter referred to as "target delivery temperature or Tm_target".

[0074] Said delivery temperature Tm_target, that, as shall be seen, is determined for every operating condition of the heat pump HP, is that temperature that, even when using a low environmental impact refrigerant (e.g. the aforementioned

R32), ensures:

- the optimal wet fraction for the refrigerant entering the compressor 13 (i.e. such as to operate in a suitable wet compression condition),
- optimal performance of the machine, said temperature compensating for the reduction of the operating *(or envelope)* range of the machine resulting from the use of said low environmental impact refrigerant (e.g. the R32), and/or
- a delivery temperature Tm of the compressor 13:

    - neither too high as to abnormally overheat the lubricating oil inside the compressor 13 and/or the relative motor, exposing it to breakages or to temporary interruptions of its operation,
    - nor too low to get excessively close to the temperature of the lubricating oil, i.e. to values that may cause the condensation of the refrigerant in the same oil and therefore the dilution thereof (also with the inevitable impairment of its ability to lubricate the moving parts of the compressor 13 and/or of other chemical-physical features thereof).

[0075] For such purpose, the expansion valve 14 of the heat pump HP is preferably an electromechanical valve and the opening degree thereof is suitably piloted and regulated, for example by means of a feedback control system, as long as the delivery temperature Tm of the compressor 13 does not approximate and/or reach the aforementioned target delivery temperature Tm_target.
[0076] Preferably, said control of the expansion valve 14 is, without any limiting intent, a control of the *Proportional-Integral-Derivative* type (hereinafter also briefly referred to as "PID control").
[0077] In other words, it has been observed that an "optimal" percentage of the wet fraction of the refrigerant at the compressor suction 13 corresponds to a delivery temperature Tm equal to a target delivery temperature Tm_target the value thereof is substantially determined as a function "*f1*" of at least:

- a first pair of parameters, variable, that depend on:

    - the environmental conditions in which the heat pump HP operates, e.g. from the temperatures T.c, T.f of the relative hot and cold wells, and/or
    - the operating conditions of the same heat pump HP, e.g. the opening degree of expansion valve 14 thereof,

- a second pair of parameters, preferably constant, representative of the type and technical features of the compressor 13 of said heat pump HP.

[0078] More precisely, said first pair of parameters preferably comprises:

- the evaporation temperature SST detected by the aforementioned temperature sensor T.evap placed at the evaporator 11, and
- the condensation temperature SDT detected by the aforementioned temperature sensor T.cond placed at the evaporator 12,

while said second pair of parameters may comprise:

- the aforementioned safety (or threshold) value OIL_SH for the difference between the temperature of the lubricating oil inside the compressor 13 and that of the refrigerant in the refrigeration circuit (at the delivery of the same compressor),
- a correction coefficient k, being it also a function of the technical features of the compressor 13, in particular of the heat insulation thereof, and adapted to take into account the inevitable heat losses between the compressor 13 and the environment (air) in which the heat pump HP operates, i.e., the heat exchange between the lubricating oil and compressor 13 and between the lubricating oil and the refrigerant.

[0079] In formula:

$$Tm\_target = f1(SDT, SST, k, OIL\_SH) \qquad (2)$$

[0080] Preferably, Tm_target may be equal to the sum between the aforementioned condensation temperature SDT, the value OIL_SH and a correction "*f2*" that, in turn, is determined according to the model and technical features of the compressor 13 and of the operating conditions of the heat pump. HP, i.e. the condensation and evaporation temperatures thereof SDT, SST and the safety (or threshold) value OIL_SH; in formula:

$$Tm\_target = SDT + OIL\_SH + f2(SDT, SST, k, OIL\_SH) \qquad (3)$$

[0081] Even more specifically, said correction *f2* is preferably equal to the algebraic sum "SDT + OIL_SH - SST" between the condensation and evaporation temperatures of the heat pump HP and the safety (or threshold) value for the acceptable temperature difference between the lubricant oil of the compressor and delivery refrigerant, which is given a "weight" k that depends on the model of the compressor 13 and the technical features thereof (i.e., corresponding to the aforementioned correction coefficient k that takes into account the heat losses to the compressor); in formula:

$$Tm\_target = SDT + OIL\_SH + k*(SDT + OIL\_SH - SST) \qquad (4)$$

[0082] It is useful to reiterate how the correction *f2 = k*(SDT + OIL SH - SST)* substantially represents a contribution that takes into account the heat losses between the compressor 13 of the heat pump HP and the environment (air) in which it operates and that may be due to an excessive cooling of the lubricating oil in the same compressor 13.
[0083] In particular, said correction *f2* takes into account the heat exchange coefficients:

- $\alpha 1$ between lubricating oil and refrigerant of the refrigeration circuit of the heat pump HP, and
- $\alpha 2$ between the same lubricating oil and the operating environment of said heat pump HP.

[0084] This is easily inferable from the following syllogism.
[0085] In the presence of heat losses between compressor 13 and environment (air), the heat balance is to be checked:

$$(Tm\_target - Toil)*\alpha 1 = (Toil - Tair)*\alpha 2 \qquad (5)$$

hence, assuming:

- a *Toil = SDT + OIL_SH* that represents the oil temperature in the ideal case of total absence of heat losses,
- a *Tair = SST* (in order to take into account the worst operating conditions for a heat pump HP; *Tair* is in fact > of the evaporation temperature),

there is obtained:

$$(Tm\_target - SDT - OIL\_SH) *\alpha 1 = (SDT + OIL\_SH - SST)*\alpha 2 \qquad (6)$$

wherefrom:

$$Tm\_target = SDT + OIL\_SH + \alpha 2/ \alpha 1*(SDT + OIL\_SH - SST) \qquad (6')$$

and wherefrom it may be further seen, how the ratio:
$\alpha 2/\alpha 1$ actually corresponds to the previously introduced and described correction coefficient k.

[0086] In other words, it has been shown that the correction coefficient $k = \alpha 2/\alpha 1$ introduced to take into account the possible cooling of the lubricating oil of the compressor 13 due to the heat losses outwards is defined as the ratio of the heat transfer coefficients between lubricating oil and refrigerant and between lubricating oil and the operating environment of the heat pump HP.
[0087] By way of an example and without any limiting intent, the correction coefficient k may be between 0.05 < k < 0.35, with values that are as lower as more effectively the compressor 13 of said heat pump HP is thermally insulated. Laboratory tests have shown that k may preferably be equal to 0.15, that may be possibly raised to 0.25 for safety reasons.
[0088] As already anticipated, according to the invention, the expansion valve 14 of the heat pump HP is piloted, preferably by means of a control PID, to regulate the opening degree thereof so as to ensure a refrigerant temperature Tm at the delivery of the compressor 13 equal to Tm_target as defined above.
[0089] It should be noted that said formula of

$$Tm\_target = SDT + OIL\_SH + k*(SDT + OIL\_SH - SST) \quad (7)$$

is recursive: in fact, at each regulation of the expansion valve 14, in addition to a variation in the delivery temperature Tm actually measured at the outlet of the compressor 13, new values of the condensing temperatures SDT and evaporation temperatures SST and therefore of the same Tm_target calculated from the formula also correspond.

[0090] Therefore, it appears more correct to define said Tm_target with the following formula:

$$Tm\_target.t = Tm.t = SDT.t + OIL\_SH + k*(SDT.t + OIL\_SH - SST.t) \quad (8)$$

where:

- *SDT.t* represents the condensing temperature in an instant t and depending on the effective value of the delivery temperature TD.t of the compressor 13 in the same instant t;
- *SST.t* represents the evaporation temperature in an instant t and depending on the effective value of the delivery temperature TD.t of the compressor 13 in the same instant t,
- Delivery *Tm_target.t* = *Tm. t* of the compressor 13 considered as ideal and optimal for the values *SDT.t* and *SST.t* just read and measured in said instant t,
- *OIL_SH* is, as seen, a compressor specific threshold value 13 and representative of a temperature difference between lubricating and the refrigerant oil and for which there is no condensation of the refrigerant in the lubricating oil (value preferably comprised between 5°C and 10°C, for example equal to OIIL_SH_opt = 7°C),
- k is the aforesaid correction coefficient that takes into account the heat losses to the compressor 13.

[0091] Therefore, according to the logic of the invention, during the control and regulation of the opening degree of the expansion valve 14 of the heat pump HP, in different and consecutive time instants t_1, t_2, ..., t_{n-1}, t_n, t_{n+1}, values of the condensation and evaporation temperatures, depending in turn on the value Tm.tn of the delivery temperature of the compressor 13 in force at the time tn of said measurement, are measured; that is, at the instant tn there shall be:

- a condensing temperature SDT.tn = SDT(Tm.tn), and
- an evaporation temperature SST.tn = SST(Tm.tn).

[0092] From such values, the value of the target delivery temperature of the compressor Tm_target to be reached at the following instant tn+1 by operating the expansion valve 14 is therefore obtained and calculated, the constants OIL_SH, OIL_SH_opt and k being known.

[0093] This means that in an instant tn+1 following tn, by further regulating the opening degree of the expansion valve 14, the aim is for a delivery temperature Tm_target.tn+1 whose value depends on that of the condensing SDT.tn, evaporation SST.tn and delivery Tm.tn temperature of the compressor 13 read in said instant tn.

[0094] The expansion valve 14 is therefore operated, more or less "abruptly" by the PID control (through the *proportional, derivative* and/or *integrative* criteria thereof), according to the difference between the last delivery temperature value Tm.tn read and measured in an instant tn and the last corresponding value calculated for the target delivery temperature Tm_target.tn+1, i.e., in formula:

$$Tm\_target.t_{n+1} = SDT.tn + OIL\_SH + k*(SDT.tn + OIL\_SH - SST.tn) \quad (10)$$

[0095] If the heat pump HP is operating in a steady state, in particular if the temperatures T.f, T.c of the cold and hot well remain substantially constant, for example because there is a continuous consumption of water that subtracts from the hot well (e.g. from one of its tanks) a thermal power substantially equal to that introduced by the heat pump HP, at a certain instant tn+1 there is obtained that:

- $Tm\_target.t_{n+1} = Tm\_target.t_n$ already reached at instant tn, and
- the expansion valve no longer has to correct the opening degree thereof.

[0096] In other words, in stationary conditions, the subsequent values of the Tm_target provided by the logic of the invention calculated on the basis of the values of the condensation and evaporation temperatures SDT, SST read in the immediately preceding instant converge to a constant and time invariant value Tm_target.

[0097] In this way, even under compression conditions of a wet operating fluid (or refrigerant) *(wet compression),* it is therefore possible to guarantee that the effective delivery temperature Tm of the compressor 13:

- remains always below a maximum permissible limit defined by the Manufacturer so as to avoid breakages and malfunctions due to an excessive overheating of the lubricating oil and/or the mechanical and electronic parts and components thereof, but
- is not too low to come too close to the temperature of the lubricating oil of said compressor, that is to values that may cause the refrigerant to condense in said oil causing the dilution thereof (i.e., which is equivalent, so that said oil remains sufficiently hot).

[0098] In accordance with a variant of the invention, and in certain operating steps of the heat pump HP, it is possible, alternatively or in combination with the regulation of the opening degree of the expansion valve 14 described above, to keep the lubricating oil of the compressor 13 sufficiently hot, and consequently prevent the refrigerant from condensing therein, by heating directly said oil; for this purpose, a heating element C7 may therefore be provided, preferably an electrical resistance C7, placed externally to the oil sump C3 of the compressor 13 (see Fig. 7).

[0099] According to such variant, during the compression of the wet refrigerant, it is therefore desired to keep the temperature difference between the lubricating oil and the refrigerant at the delivery of the compressor 13 above a certain minimum safety threshold *OIL_SH_min,* representative of a temperature of the lubricating oil sufficiently high to avoid the condensation of the refrigerant.

[0100] In particular, from the previously defined and treated formulas (in particular from formula (8)), it is possible to define such difference as:

$$OIL\_SH = [Tm - SDT*(1+k) + k*SST] / (1+k) \qquad (11)$$

and the electrical resistance C7 will be activated if said calculated value of OIL_SH is lower than the aforementioned *OIL_SH_min,* obviously taking into account an appropriate hysteresis; in formula:

- if *OIL_SH* < *(OIL_SH_min)* → the resistance is activated;
- if OIL_SH > *(OIL_SH_min + hysteresis)* → the electrical resistance remains switched off or, if already in operation, it is deactivated.

[0101] Preferably, said minimum threshold value *OIL_SH_min,* indicative for the activation or not of the electrical resistance C7, is a value lower than the safety threshold *OIL_SH_opt* to be ensured and maintained during the regulation of the previously described opening degree of the expansion valve 14.

[0102] By way of an example, since *OIL_SH_opt* was assumed to be preferably equal to 7°C, the minimum threshold value *OIL_SH_min* for the switching on/off of said electrical resistance C7 may be set substantially equal to 5°C.

[0103] In such case, for the purposes of the invention, the control and regulation of the expansion valve 14 may be associated in a synergistic and joint way with the control on the activation of the electrical resistance C7 of the compressor 13.

[0104] In fact, only the adjustment of the expansion valve 14 would be operated, in the ways seen above, as long as the difference in temperature between the lubricating oil inside the compressor 13 and the wet refrigerant compressed therein remains substantially around the set and desired value OIL_SH_opt (at which, therefore, there is no condensation of the refrigerant in the oil) while the electrical resistance C7 would activate if said oil-refrigerant temperature difference dropped below the aforementioned minimum threshold OIL_SH_min (as mentioned, equal, for example, to 5°C), as it may happen in some transient conditions of the compressor 13 (in such cases, therefore the regulation of the expansion valve 14 alone may be too slow to prevent said undesired condensation of the refrigerant in the oil).

[0105] For example, during the compressor 13 starts up with low external ambient temperatures, that is when the temperatures of the lubricating oil therein may be very low, the electrical resistance C7 is first turned on to quickly heat the oil and bring the difference between the temperature thereof and that of the refrigerant to values higher than OIL_SH_min, then, once deactivated, the aforementioned regulation of the expansion valve 14 is carried out.

[0106] Naturally, nothing prevents the possibility of controlling and measuring the temperature difference between the lubricating oil and the refrigerant, in the ways discussed above, also during and substantially along with the regulation steps of the expansion valve 14.

[0107] From the formula (11) shown just above, it is in fact evident that said difference OIL_SH between oil and refrigerant may be determined according to the delivery Tm, condensation SDT and evaporation SST temperatures of the heat pump HP that, as seen, vary at each regulation of the opening degree of the expansion valve 14, and by the aforementioned correction coefficient k for the heat losses to the compressor 13.

[0108] Therefore, it is possible to control and pilot the activation or not of the electrical resistance C7 (once the conditions indicated above have been met) substantially after each regulation of the opening degree of the expansion valve 14 or after a predetermined number of consecutive regulations of the same.

[0109] More precisely, if following the regulation of said expansion valve 14 to the consequent delivery, expansion and condensation temperature readings of the heat pump HP, and/or in the event of changed environmental or operating conditions, a lower value of OIL_SH corresponds to the minimum admissible one OIL_SH_min, the electrical resistance C7 would activate in order to quickly heat the oil and bring OIL_SH back to safety values, avoiding any risk of condensation of the refrigerant in the lubricating oil.

[0110] Finally, nothing prevents an extremely simplified form of control in which the switching on or not of the electrical resistance C7 is assigned to a direct detection of the temperature of the lubricating oil, rather than according to the aforementioned delivery, evaporation and condensation temperatures of the heat pump HP.

[0111] In such case, at least one temperature sensor may be provided for the detection of said temperature Toil of the oil of the compressor 13 adapted to the lubrication of at least the moving parts thereof (for example, as seen, for at least one or more compression chambers C2), said sensor being able to be placed, for example, in contact with the sump C3 of said compressor 13.

[0112] It is clear that several variants of the method of the invention for the control and management of the delivery temperature of a compressor of a heat pump are possible to the man skilled in the art, without departing from the novelty scopes of the inventive idea, as well as it is clear that in the practical embodiment of the invention the various components described above may be replaced with technically equivalent elements.

[0113] It has been widely seen above how, according to the invention, the method for the management and control of a thermodynamic machine (e.g., a heat pump HP) preferably operating with a low environmental impact refrigerant (GWP), and set up for winter heating and/or summer cooling of a residential, commercial, industrial or similar environment, provides for the suction and compression to the compressor 13 of said refrigerant in a "wet" state, i.e., characterised by an optimal percentage of liquid fraction, by suitably acting on the opening degree of the expansion valve 14 thereof (for example, by means of a feedback control).

[0114] In such way it is possible to regulate the delivery temperature Tm of the compressor 13, by compensating for the drawbacks resulting from the use of low GWP refrigerants, discussed so far, and continuing to ensure high overall performance to the thermodynamic machine HP.

[0115] In particular, according to the invention it is wished to adequately lower the delivery temperature Tm of the compressor 13, however avoiding that it reaches too low values that could lead the refrigerant to condense in the oil present inside the same compressor 13 with the risk of compromising the lubrication of the moving parts thereof.

[0116] As described so far, the aim of the method of the invention is therefore to regulate the wet fraction of the refrigerant so as to approximate the delivery temperature Tm of the compressor 13 to a value Tm_target value, referred to as "target delivery temperature", that represents that temperature ensuring optimal performance of the machine HP and eliminates the risk of the condensation of the refrigerant in the oil of said compressor 13.

[0117] Over the course of the present description, said "target delivery temperature" Tm_target has been defined as:

$$Tm\_target = SDT + OIL\_SH + k*(SDT + OIL\_SH - SST),$$

where,

- SDT and SST represent, respectively, the condensation and evaporation temperatures of the thermodynamic machine HP,
- OIL_SH is the safety value for the temperature difference between lubricating oil of the compressor 13 and that of the refrigerant at the delivery thereof,
- k a correction factor that takes into account the heat losses between said compressor 13 and the environment in which the same thermodynamic machine HP operates.

[0118] In general, the implementation of the method for the management and control of the invention on thermodynamic machines, for example on heat pumps HP, described so far, guarantees maximum performance and efficiency when the compressor 13 thereof, e.g. of rotary type, operates at high rotation frequency, for example at high load and/or in nominal operating conditions.

[0119] On the other hand, it has been experimentally observed that, at low rotation frequencies of the compressor 13 of the thermodynamic machine HP, the same performance results are more difficult to achieve. This is normally due to the fact that at low revolution speed of said compressor 13, suction and compression of a wet refrigerant with the most appropriate and/or optimal percentage of wet fraction, as stated several times above, may not be ensured; sometimes even a suction of the refrigerant has been observed in an overheated or strongly overheated state.

[0120] The more the refrigerant sucked in by the compressor 13 is overheated, the lower the operating efficiency thereof will be and, consequently, the overall performance of the relative thermodynamic machine HP will decrease.

[0121] Furthermore, such effect is more evident the smaller the load (or quantity) of refrigerant allowed to circulate in the thermodynamic circuit, and relative components, of the machine HP, is; a lower refrigerant load is typical of the applications involving the use of low environmental impact refrigerants, generally more flammable than those traditionally used, such as, for example, the already mentioned R32 or the like.

[0122] It is known that, sometimes, a compressor of a thermodynamic machine HP for the heating and/or cooling may operate with variable rotation frequencies, so as to adapt to the power required according to specific environmental and/or operating conditions.

[0123] By way of an example, and without any limiting intent, reference may be made to the case of compressors that are operated at low rotation frequencies, i.e. at partial load conditions, in order to evaluate the seasonal performance of the relative thermodynamic machine HP, definable by means of the well-known seasonal energy efficiency indexes SEER (*Seasonal Energy Efficiency Ratio*) and SCOP (*Seasonal Coefficient of Performance*).

[0124] By way of information, the SEER index represents the ratio between the annual energy requirement for cooling and the consumption of electrical energy intended thereto, while SCOP identifies the seasonal performance coefficient relating to heating, indicated as the ratio between the heating annual requirement and related electrical consumption.

[0125] In extreme summary, such indices are parameters that allow determining both the energy performance of the machine HP and the environmental impact thereof.

[0126] Therefore, to overcome the drawbacks described just above, according to a possible executive variant of the method for the management and control of the invention, the aforementioned target delivery temperature Tm_target is furthermore a function of at least the rotation frequency of the compressor 13. More precisely, a "correction" factor f3, that takes into account the impact of the actual rotation frequency of the compressor 13 on the delivery temperature Tm thereof, is preferably added to said target delivery temperature Tm_target.

[0127] In other words, said correction factor f3 makes it possible to optimise and ensure high efficiency and performance of the thermodynamic machine HP even when it operates under partial load conditions and/or with reduced rotation frequencies of the same compressor 13 with respect to a full load operation.

[0128] Without any limiting intent, said correction factor f3 is preferably a function of at least:

- an actual rotation frequency f_r of the compressor 13, i.e. the "reduced" frequency at which said compressor 13 operates, that may be less than or equal to a nominal and/or full load frequency of the thermodynamic machine HP, and
- a "reference" frequency f_0 of said compressor 13 that may, for example, correspond to the aforementioned frequency thereof of the nominal and/or full load condition,

being, in general, $f_r < f_0$.

[0129] In other words, according to a possible executive embodiment, said correction factor f3 enables to "correct" the target delivery temperature according to the low operating rotation frequencies of a compressor 13 and in a manner relative, for example, to the nominal one.

[0130] Such correction is all more significant and greater the lower the operating rotation frequency of the compressor 13 is.

[0131] Preferably, said correction factor f3 may be defined as proportional to the difference between said actual frequency f_r of the compressor 13 and a reference frequency f_0 thereof; in formula:

$$f3 = k3*(f_r - f_0),$$

where

- k3 represents a correction coefficient adapted to take into account the operating and/or constructive features of said compressor 13.

[0132] Without any limiting intent, the coefficient k3 may be a constant, suitably predefined, or, alternatively, a function of at least:

- the suction refrigerant temperature to the compressor 13 (e.g., the evaporation temperature SST) and/or the delivery pressure thereof, and/or
- the displacement of the compressor 13, for example a function of the inverse of the displacement, and/or
- the dissipative effects to the compressor 13, linked, for example, to the viscous friction between the rotating and moving parts thereof (in turn depending, as known, at least on the viscosity of the lubricating oil and the actual temperature thereof), and/or
- the ratio between the refrigerant load (or quantity) that may be used in the machine HP and the volume of the thermodynamic circuit thereof and related components (evaporator, condenser, etc.).

[0133] In the light of what just described, due to said correction factor f3, an improved target delivery temperature may be defined, hereinafter referred to as optimised target temperature "Tm_target_opt" and equal to:

$$\text{Tm\_target\_opt} = \text{Tm\_target} + \text{k3}^*(f_r - f_0)$$

where, for the sake of clarity:

- Tm_target_opt represents said "optimised target temperature" to take into account the contribution of the actual rotation frequency of the compressor 13,
- Tm_target represents the target delivery temperature as defined according to the first variant of the previously described method for the management and control of the invention, (see, for example, the formula Tm_target = *SDT + OIL SH + k*(SDT + OIL SH - SST)* already referred to several times for reasons of clarity and synthesis). Consequently, it is immediately clear that even said optimised target temperature "Tm_target_opt" is further a function of at least the condensation SDT, evaporation SST and oil OIL_SH temperatures, as well as the correction coefficient k of the heat losses that takes into account the heat transfer coefficients of the lubricating oil with the refrigerant ($\alpha 1$) and the operating environment of the machine HP ($\alpha 2$);
- k3*(f_r - f_0), is the aforementioned correction factor f3 of at least the actual rotation frequency of the compressor 13.

[0134] It should be noted that the correction factor f3 may assume a negative value and therefore like Tm_target_opt may be lower than the Tm_target as defined in accordance with the first variant of the invention (Tm_target_opt < Tm_target). For completeness of information, it is also specified that, also for this improved variant of the method for the management and control of a thermodynamic machine HP (e.g., of a heat pump HP) a regulation of the opening degree of the relative expansion valve 14 is envisaged, and therefore of the delivery temperature Tm of the compressor 13, preferably a reduction thereof.

[0135] With substantially similar methods, except for small adaptations within the reach of a person skilled in the art, to those already seen with reference to the first variant of the invention, described above, it is always desired to obtain that the delivery temperature Tm of the compressor 13 is reduced until converging and/or approximating, to an optimal value, now defined by the aforementioned optimised target temperature Tm_target_opt, capable of maximising the performance of the machine or heat pump HP in any load condition, without any risk of condensation of the refrigerant in the oil of said compressor 13.

[0136] For such purpose, it should be noted that also said optimised target temperature Tm_target_opt is recursive; that is, in accordance with what has already been seen with reference to the first variant of the invention, even in such case each regulation of said expansion valve 14, in addition to a variation of the delivery temperature Tm actually measured at the outlet of the compressor 13, there also correspond new values of the condensing SDT and evaporation SST temperatures, and of any parameters and coefficients depending on them, and therefore of the same Tm_target_opt.

[0137] In conclusion, it has been observed that, thanks to said optimised target temperature Tm_target_opt, there is the possibility to achieve high performance for the thermodynamic machine HP in any load condition, by sucking and compressing from the compressor 13 a refrigerant with the most suitable liquid fraction under both high-frequency and low-frequency operating conditions of the same compressor 13, and to ensure a corresponding delivery temperature Tm that is substantially the lowest possible among those that do not involve the risk of condensation of the refrigerant in the lubricating oil.

[0138] More specifically, it has been observed that the contribution of the correction factor f3 on the target temperature may lead to a delivery temperature Tm of the compressor 13 actually lower than the one obtainable, other conditions being equal, with the first variant of the invention without such contribution f3, and therefore to an improvement of the overall efficiency of the thermodynamic machine HP

[0139] For example, considering the aforementioned correction factor f3 in determining the optimised target temperature Tm_target_opt of the compressor 13, there was generally shown an increase both in the COP (Coefficient of Performance) and EER (Energy Efficiency Ratio) especially at low rotation frequencies of the same compressor 13.

Claims

1. Method for the management and control of a thermodynamic machine, e.g. a heat pump (HP), based on a compression/expansion thermodynamic cycle of an operating fluid and comprising at least:

   - a first heat exchanger (11; 12) wherein said operating fluid absorbs, at constant pressure, thermal energy from a cold well,

- a second heat exchanger (12; 11) wherein said operating fluid yields, at constant pressure, part of the thermal energy thereof to a hot well,
- an expansion valve (14) placed between said first (11; 12) and second (12; 11) heat exchanger and adapted to carry out a constant enthalpy expansion and cooling of said operating fluid,
- a compressor (13; C) adapted to compress said operating fluid between a minimum pressure thereof it has at the outlet of said first heat exchanger (11; 12) and a maximum pressure thereof it has at the inlet of said second heat exchanger (12; 11), said compressor (13; C) being able to suck and compress a wet operating fluid with a suitable percentage of liquid fraction,
- at least one temperature sensor (T.com) for the detection of the delivery temperature Tm of said compressor,
- at least one temperature sensor (T.evap) for detecting an evaporation temperature SST in said first exchanger (11; 12),
- at least one temperature sensor (T.cond) for detecting a condensation temperature SDT in said second exchanger (12; 11),

characterised in that

the temperature difference between said lubricating oil in the compressor (13; C) and said operating fluid at the delivery of the same compressor (13; C) is:

- maintained equal to or higher than a safety threshold OIL_SH such that there is no condensation of said operating fluid in said lubricating oil, i.e. such that the relationship is verified: Toil - Tm ≥ OIL_SH,
- controlled and/or maintained on said safety value OIL_SH mainly by regulating the delivery temperature Tm of said compressor (13; C),

said delivery temperature Tm of the compressor (13; C) being regulated until it approaches and/or substantially reaches an optimised target delivery temperature Tm_target_opt that is:

- that adapted to ensure:
    - an optimal wet fraction entering said compressor (13; C),
    - a delivery temperature Tm of said compressor (13; C) not too low to cause the condensation of the operating fluid in said lubricating oil,
- a function of at least the rotation frequency of said compressor (13; C).

2. Method for the management and control according to claim 1, **wherein** said optimised target delivery temperature Tm_target_opt is a function of at least:

- one actual rotation frequency f_r of the compressor (13; C) that may be less than or equal to a nominal and/or full load frequency of the thermodynamic machine (HP),
- a reference frequency t_0 of said compressor (13; C), said frequency f_0 being able to correspond to the aforementioned nominal and/or full load frequency.

3. Method for the management and control according to claim 2, **wherein** said optimised target delivery temperature Tm_target_opt is a function of a correction factor f3 proportional to the difference between said actual frequency f_r of said compressor (13; C) and said reference frequency f_0, in formula:

$$f3 = k3*(f_r - f_0)$$

where k3 represents a coefficient adapted to take into account the operating and/or constructive features of said compressor (13; C)

4. Method for the management and control according to claim 3 wherein said coefficient k3 may be:

- a constant, or
- function at least:

- of the temperature of the refrigerant in suction to said compressor (13; C), and/or
- of the pressure at the delivery of said compressor (13; C), and/or
- of the displacement of the compressor 13, and/or
- of the dissipative effects to said compressor (13; C), and/or
- of the ratio between the load of refrigerant that may be used in said thermodynamic machine (HP) and the volume of the thermodynamic circuit and related components thereof.

5. Method for the management and control according to one or more of the previous claims, **wherein** said optimised target delivery temperature Tm_target is a further function of at least:

- said evaporation temperature SST detected at the heat exchanger (11; 12) adapted to act as an evaporator,
- said condensation temperature SDT detected at the heat exchanger (12; 11) adapted to act as a condenser,
- the said safety value OIL_SH for the temperature difference between said lubricating oil contained within said compressor (13; C) and said operating fluid at its delivery,
- one correction coefficient k that takes into account the heat losses between said compressor (13; C) and the environment in which said heat pump (HP) operates, where said coefficient k, defined as $\alpha 2/\alpha 1$, by taking into account the heat exchange coefficients $\alpha 1$ and $\alpha 2$ between said lubricating oil and, respectively, said operating fluid and said operating environment of said thermodynamic machine (HP).

6. Method for the management and control according to at least claims 4 and 5, **wherein** said optimised target delivery temperature Tmtarget opt is equal to: SDT + OIL_SH + k*(SDT + OIL_SH - SST) + k3*(f_r - f_0).

7. Method for the management and control according to one or more of the previous claims, **wherein** said regulation of the delivery temperature Tm of said compressor (13; C) is obtained by regulating the opening degree of said expansion valve (14).

8. Method for the management and control according to claim 7, **wherein** said regulation of the opening degree of said expansion valve (14) is carried out means of a feedback control.

9. Method for the management and control according to any previous claim, **wherein** said operating fluid is a low environmental impact refrigerant, for example the refrigerant R32.

Patentansprüche

1. Verfahren zur Steuerung und Kontrolle einer thermodynamischen Maschine, z.B. einer Wärmepumpe (HP), basierend auf einem thermodynamischen Kompressions-/Expansionszyklus einer Betriebsflüssigkeit und mindestens umfassend:

- einen ersten Wärmetauscher (11; 12), in dem die Betriebsflüssigkeit bei konstantem Druck thermische Energie aus einer Kältequelle absorbiert,
- einen zweiten Wärmetauscher (12; 11), in dem die Betriebsflüssigkeit bei konstantem Druck einen Teil der thermischen Energie davon an eine Wärmequelle abgibt,
- ein Expansionsventil (14), das zwischen dem ersten (11; 12) und dem zweiten (12; 11) Wärmetauscher angeordnet ist und geeignet ist, eine konstante isenthalpe Expansion und Kühlung der Betriebsflüssigkeit durchzuführen,
- einen Kompressor (13; C), der geeignet ist, die Betriebsflüssigkeit zwischen einem Mindestdruck davon, den sie am Austritt des ersten Wärmetauschers (11; 12) aufweist, und einem Höchstdruck davon, den sie am Eintritt des zweiten Wärmetauschers (12; 11) aufweist, zu komprimieren, wobei der Kompressor (13; C) eine nasse Betriebsflüssigkeit mit einem geeigneten Prozentsatz an Flüssigkeitsanteil ansaugen und komprimieren kann,
- mindestens einen Temperatursensor (T.com) zum Erfassen der Abgabetemperatur Tm des Kompressors,
- mindestens einen Temperatursensor (T.evap) zum Erfassen einer Verdampfungstemperatur SST in dem ersten Tauscher (11; 12),
- mindestens einen Temperatursensor (T.cond) zum Erfassen einer Kondensationstemperatur SDT in dem zweiten Tauscher (12; 11),

dadurch gekennzeichnet, dass

die Temperaturdifferenz zwischen dem Schmieröl im Kompressor (13; C) und der Betriebsflüssigkeit am Austritt des Kompressors (13; C):

- gleich oder größer als ein Sicherheitsgrenzwert OIL_SH derart gehalten wird, dass keine Kondensation der Betriebsflüssigkeit im Schmieröl auftritt, d.h. derart, dass die Beziehung erfüllt wird: Toil-Tm ≥ OIL_SH,
- im Wesentlichen durch Regulieren der Abgabetemperatur Tm des Kompressors (13; C), kontrolliert und/oder auf dem Sicherheitswert OIL_SH gehalten wird,

wobei die Abgabetemperatur Tm des Kompressors (13; C) reguliert wird, bis sie sich einer optimierten Soll-Abgabetemperatur Tm_target_opt nähert und/oder diese im Wesentlichen erreicht, das heißt:

- jene, die geeignet ist, um zu gewährleisten:
  - einen optimalen Nassanteil, der in den Kompressor (13; C) eintritt,
  - eine Abgabetemperatur Tm des Kompressors (13; C), die nicht zu niedrig ist, um die Kondensation der Betriebsflüssigkeit in dem Schmieröl zu verursachen,
- eine Funktion von mindestens der Drehfrequenz des Kompressors (13; C).

2. Verfahren zur Steuerung und Kontrolle nach Anspruch 1, **wobei** die optimierte Soll-Abgabetemperatur Tm_target_opt eine Funktion ist von mindestens:

- einer tatsächlichen Drehfrequenz f_r des Verdichters (13; C), die kleiner oder gleich einer Nenn- und/oder Volllast-Frequenz der thermodynamischen Maschine (HP) sein kann,
- einer Referenzfrequenz f_0 des Kompressors (13; C), wobei die Frequenz f_0 der vorgenannten Nenn- und/oder Volllast-Frequenz entsprechen kann.

3. Verfahren zur Steuerung und Kontrolle nach Anspruch 2, **wobei** die optimierte Soll-Abgabetemperatur Tm_target_opt eine Funktion von einem Korrekturfaktor f3 ist, der proportional zur Differenz zwischen der tatsächlichen Frequenz f_r des Kompressors (13; C) und der Referenzfrequenz f_0 ist, in der Formel:

$$f3 = k3*(f_r - f_0)$$

wobei k3 einen Koeffizienten darstellt, der geeignet ist, die Betriebs- und/oder Konstruktionsmerkmale des Kompressors (13; C) zu berücksichtigen.

4. Verfahren zur Steuerung und Kontrolle nach Anspruch 3, wobei der Koeffizient k3 sein kann:

- eine Konstante oder
- eine Funktion mindestens:
  - von der Temperatur des Kältemittels beim Ansaugen zum Kompressor (13; C) und/oder
  - von dem Druck am Austritt des Kompressors (13; C) und/oder
  - von dem Hubraum des Kompressors 13 und/oder
  - von den dissipativen Effekten auf den Kompressor (13; C) und/oder
  - von dem Verhältnis zwischen der Last des Kältemittels, das in der thermodynamischen Maschine (HP) verwendet werden kann, und dem Volumen des thermodynamischen Kreislaufs und der zugehörigen Komponenten davon.

5. Verfahren zur Steuerung und Kontrolle nach einem oder mehreren der vorhergehenden Ansprüche, **wobei** die optimierte Soll-Abgabetemperatur Tm_target eine weitere Funktion ist von mindestens:

- der Verdampfungstemperatur SST, die am Wärmetauscher (11; 12) erfasst wird, der geeignet ist, als Verdampfer zu fungieren,
- der Kondensationstemperatur SDT, die am Wärmetauscher (12; 11) erfasst wird, der geeignet ist, als Kondensator zu fungieren,
- dem Sicherheitswert OIL_SH für die Temperaturdifferenz zwischen dem Schmieröl, das in dem Kompressor

(13; C) enthalten ist, und der Betriebsflüssigkeit bei ihrer Abgabe,
- einem Korrekturkoeffizienten k, der die Wärmeverluste zwischen dem Kompressor (13; C) und der Umgebung, in der die Wärmepumpe (HP) arbeitet, berücksichtigt, wobei der Koeffizient k, der als $\alpha 2/\alpha 1$ definiert ist, die Wärmeaustauschkoeffizienten $\alpha 1$ und $\alpha 2$ zwischen dem Schmieröl und jeweils der Betriebsflüssigkeit und der Betriebsumgebung der thermodynamischen Maschine (HP) berücksichtigt.

6. Verfahren zur Steuerung und Kontrolle mindestens nach Anspruch 4 und 5, **wobei** die optimierte Soll-Abgabetemperatur Tm_target_opt gleich ist: SDT + OIL_SH + k*(SDT + OIL_SH - SST) + k3*(f_r - f_0).

7. Verfahren zur Steuerung und Kontrolle nach einem oder mehreren der vorhergehenden Ansprüche, **wobei** das Regulieren der Abgabetemperatur Tm des Kompressors (13; C) durch Regulieren des Öffnungsgrades des Expansionsventils (14) erhalten wird.

8. Verfahren zur Steuerung und Kontrolle nach Anspruch 7, **wobei** das Regulieren des Öffnungsgrades des Expansionsventils (14) mittels einer Rückkopplungskontrolle ausgeführt wird.

9. Verfahren zur Steuerung und Kontrolle nach einem der vorhergehenden Ansprüche, **wobei** die Betriebsflüssigkeit ein Kältemittel mit geringer Umweltbelastung, beispielsweise das Kältemittel R32 ist.

Revendications

1. Procédé de gestion et de contrôle d'une machine thermodynamique, par exemple une pompe à chaleur (HP), basé sur un cycle thermodynamique de compression/détente d'un fluide de fonctionnement et comprenant au moins :

- un premier échangeur de chaleur (11 ; 12) dans lequel le fluide de fonctionnement absorbe, à pression constante, l'énergie thermique d'un puits froid,
- un deuxième échangeur de chaleur (12 ; 11) dans lequel le fluide de fonctionnement cède, à pression constante, une partie de son énergie thermique à un puits chaud,
- un détendeur (14) placé entre le premier (11 ; 12) et le second (12 ; 11) échangeur de chaleur et adapté pour effectuer une détente à enthalpie constante et un refroidissement dudit fluide de fonctionnement,
- un compresseur (13 ; C) adapté pour comprimer ledit fluide de fonctionnement entre une pression minimale qu'il a à la sortie dudit premier échangeur de chaleur (11 ; 12) et une pression maximale qu'il a à l'entrée dudit second échangeur de chaleur (12 ; 11), ledit compresseur (13 ; C) étant capable d'aspirer et de comprimer un fluide de fonctionnement humide avec un pourcentage approprié de fraction liquide,
- au moins un capteur de température (T.com) pour la détection de la température de refoulement Tm dudit compresseur,
- au moins un capteur de température (T.evap) pour détecter une température d'évaporation SST dans ledit premier échangeur (11 ; 12),
- au moins un capteur de température (T.cond) pour détecter une température de condensation SDT dans ledit second échangeur (12 ; 11),

caractérisé en ce que

la différence de température entre ladite huile de lubrification dans le compresseur (13 ; C) et ledit fluide de fonctionnement au refoulement du même compresseur (13 ; C) est :

- maintenue égale ou supérieure à un seuil de sécurité OIL_SH tel qu'il n'y ait pas de condensation dudit fluide de fonctionnement dans ladite huile de lubrification, c'est-à-dire tel que la relation soit vérifiée : Toil - Tm $\geq$ OIL_SH,
- contrôlée et/ou maintenue sur ladite valeur de sécurité OIL_SH principalement en régulant la température de refoulement Tm dudit compresseur (13 ; C),

ladite température de refoulement Tm du compresseur (13 ; C) étant régulée jusqu'à ce qu'elle approche et/ou atteigne sensiblement une température de refoulement cible optimisée Tm_target_opt, c'est-à-dire :

- qui s'est adaptée pour garantir :

- une fraction humide optimale entrant dans ledit compresseur (13 ; C),
- une température de refoulement Tm dudit compresseur (13 ; C) pas trop basse pour provoquer la condensation du fluide de fonctionnement dans ladite huile de lubrification,

- en fonction au moins de la fréquence de rotation dudit compresseur (13 ; C).

2. Procédé de gestion et de contrôle selon la revendication 1, dans lequel ladite température de refoulement cible optimisée Tm_target_opt est une fonction d'au moins :

- une fréquence de rotation réelle f_r du compresseur (13 ; C) qui peut être inférieure ou égale à une fréquence nominale et/ou de pleine charge de la machine thermodynamique (HP),
- une fréquence de référence f_0 dudit compresseur (13 ; C), ladite fréquence f_0 pouvant correspondre à la fréquence nominale et/ou de pleine charge susmentionnée.

3. Procédé de gestion et de contrôle selon la revendication 2, **dans lequel** ladite température de refoulement cible optimisée Tm_target_opt est fonction d'un facteur de correction f3 proportionnel à la différence entre ladite fréquence réelle f_r dudit compresseur (13 ; C) et ladite fréquence de référence f_0, selon la formule :

$$f3 = k3*(f_r - f_0)$$

où k3 représente un coefficient adapté à la prise en compte des caractéristiques de fonctionnement et/ou de construction dudit compresseur (13 ; C).

4. Procédé de gestion et de contrôle selon la revendication 3 dans lequel ledit coefficient k3 peut être :

- une constante, ou
- fonction au moins :

  - de la température du fluide frigorigène à l'aspiration dudit compresseur (13 ; C), et/ou
  - de la pression au refoulement dudit compresseur (13 ; C), et/ou
  - de la cylindrée du compresseur 13, et/ou
  - des effets dissipatifs dudit compresseur (13 ; C), et/ou
  - du rapport entre la charge de fluide frigorigène pouvant être utilisée dans ladite machine thermodynamique (HP) et le volume du circuit thermodynamique et de ses composants.

5. Procédé de gestion et de contrôle selon une ou plusieurs des revendications précédentes, **dans lequel** ladite température de refoulement cible optimisée Tm_target est une fonction supplémentaire d'au moins :

- ladite température d'évaporation SST détectée au niveau de l'échangeur de chaleur (11 ; 12) adapté pour agir en tant qu'évaporateur,
- ladite température de condensation SDT détectée au niveau de l'échangeur de chaleur (12; 11) adapté pour agir en tant que condenseur,
- ladite valeur de sécurité OIL_SH pour la différence de température entre ladite huile de lubrification contenue dans ledit compresseur (13 ; C) et ledit fluide de fonctionnement à son refoulement,
- un coefficient de correction k qui prend en compte les pertes de chaleur entre ledit compresseur (13 ; C) et l'environnement dans lequel fonctionne ladite pompe à chaleur (HP), où ledit coefficient k, défini comme $\alpha 2/\alpha 1$, en prenant en compte les coefficients d'échange de chaleur $\alpha 1$ et $\alpha 2$ entre ladite huile de lubrification et, respectivement, ledit fluide de fonctionnement et ledit environnement de fonctionnement de ladite machine thermodynamique (HP).

6. Procédé de gestion et de contrôle selon au moins les revendications 4 et 5, **dans lequel** ladite température cible de distribution optimisée Tm target opt est égale à : SDT + OIL_SH + k*(SDT + OIL_SH - SST) + k3*(f_r - f_0).

7. Procédé de gestion et de contrôle selon l'une ou plusieurs des revendications précédentes, **dans lequel** la régulation de la température de refoulement Tm dudit compresseur (13 ; C) est obtenue en régulant le degré d'ouverture dudit détendeur (14).

8. Procédé de gestion et de contrôle selon la revendication 7, **dans lequel** la régulation du degré d'ouverture du détendeur (14) est effectuée au moyen d'une commande par rétroaction.

9. Procédé de gestion et de contrôle selon toute revendication précédente, **dans lequel** le fluide de fonctionnement est un fluide frigorigène à faible impact environnemental, par exemple le fluide frigorigène R32.

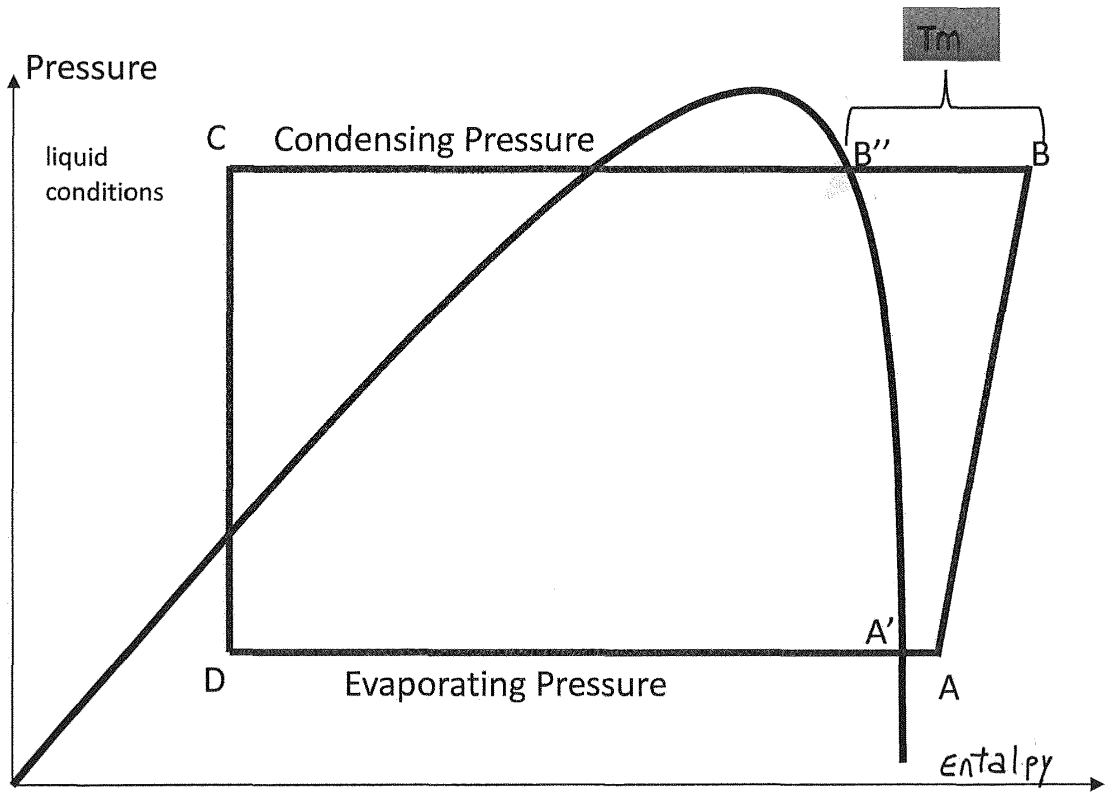

Tm
Pressure
liquid
conditions
C
Condensing Pressure
B''
B
A'
D
Evaporating Pressure
A
Entalpy

Fig. 1

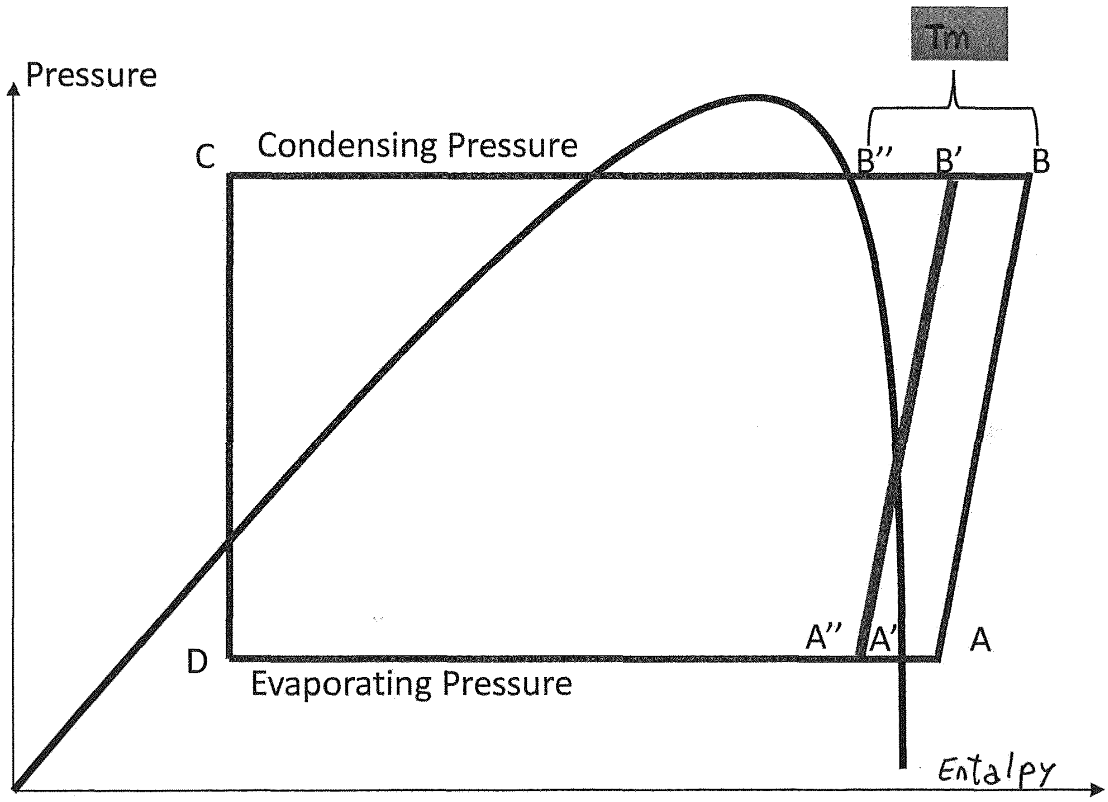

Tm
Pressure
C
Condensing Pressure
B''
B'
B
A''
A'
A
D
Evaporating Pressure
Entalpy

Fig. 2

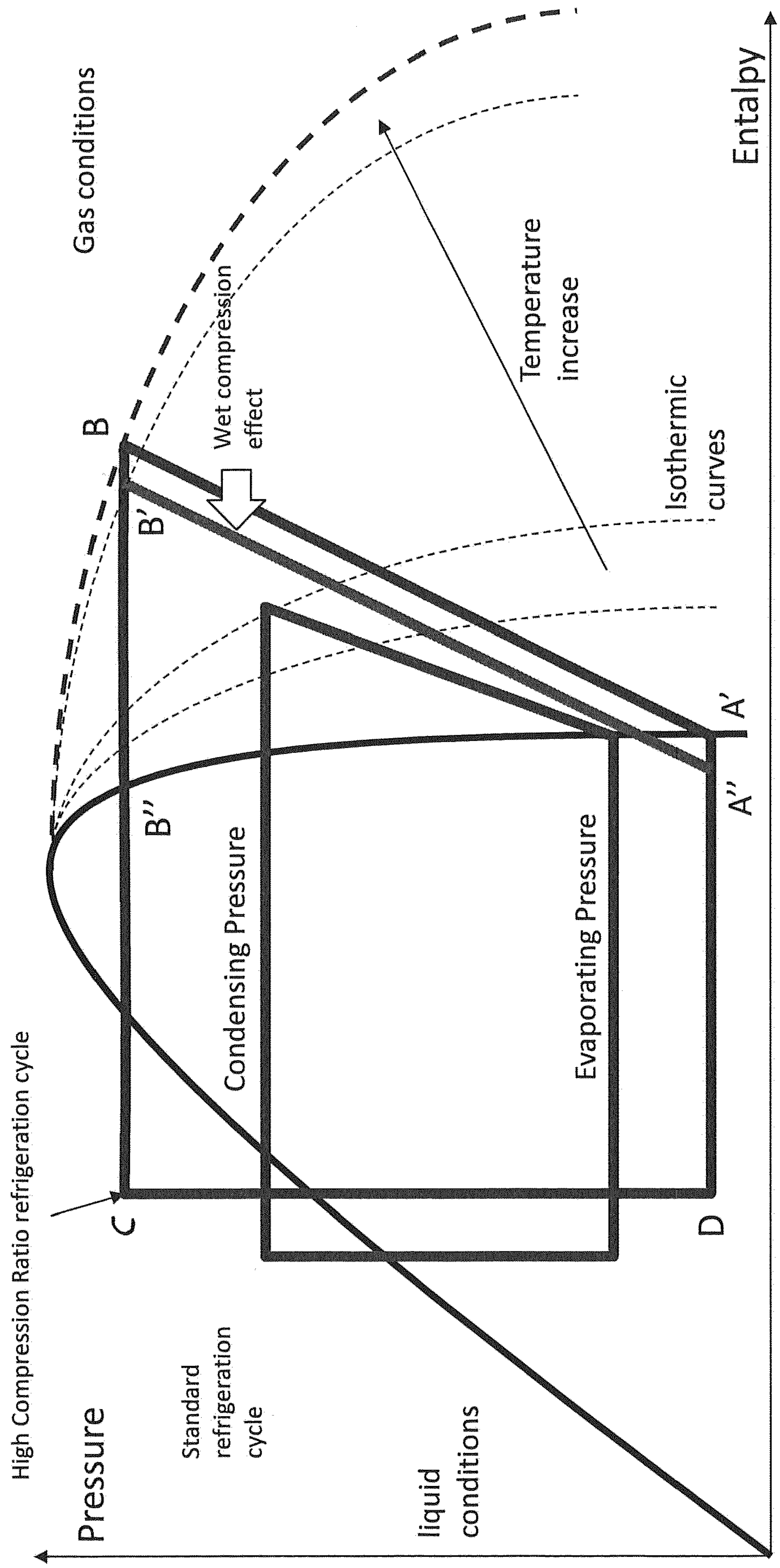

Gas conditions
Entalpy
Wet compression effect
Temperature increase
Isothermic curves
B
B'
B''
A'
A''
C
D
Condensing Pressure
Evaporating Pressure
High Compression Ratio refrigeration cycle
Pressure
Standard refrigeration cycle
liquid conditions

Fig. 3

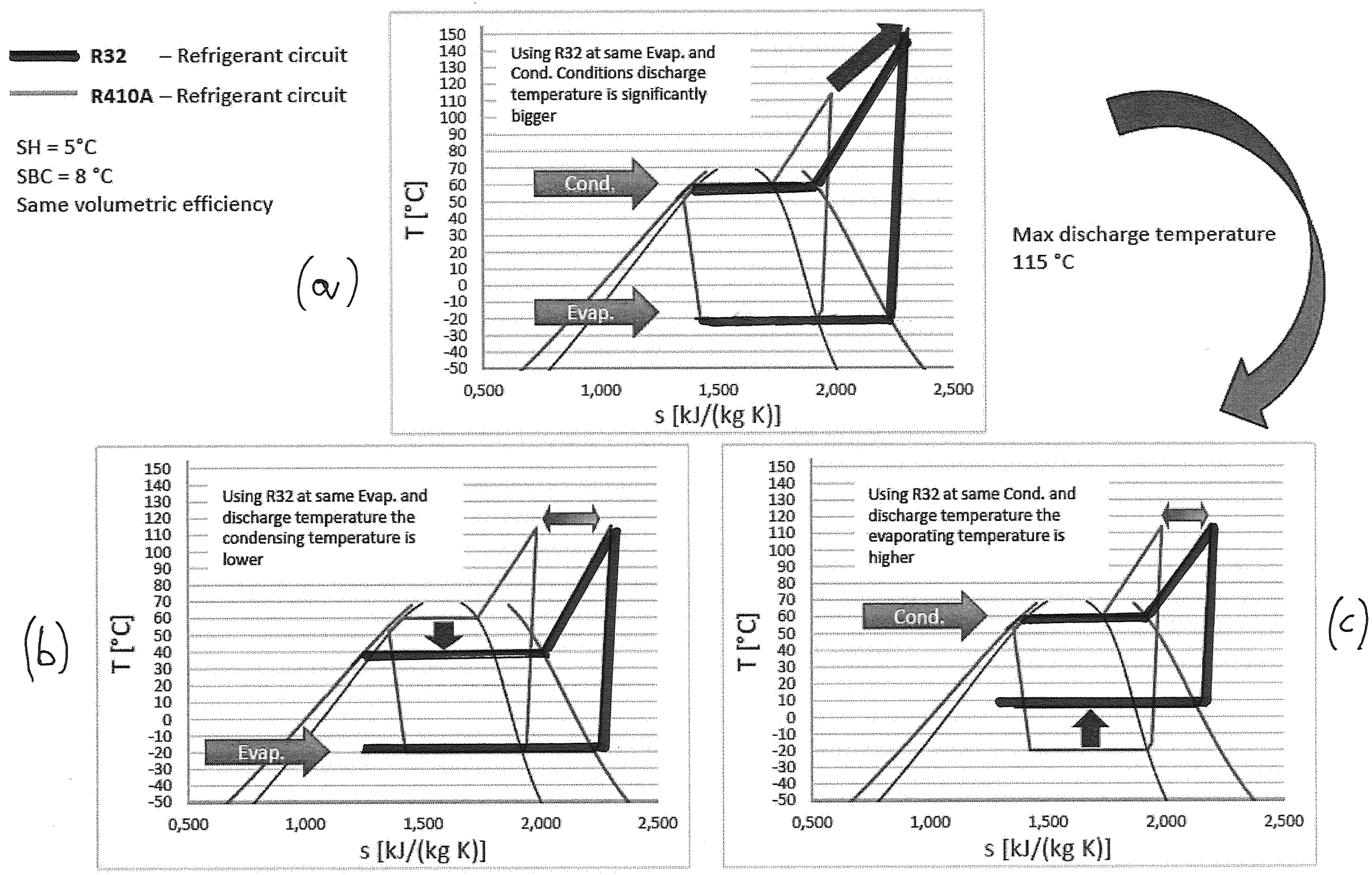
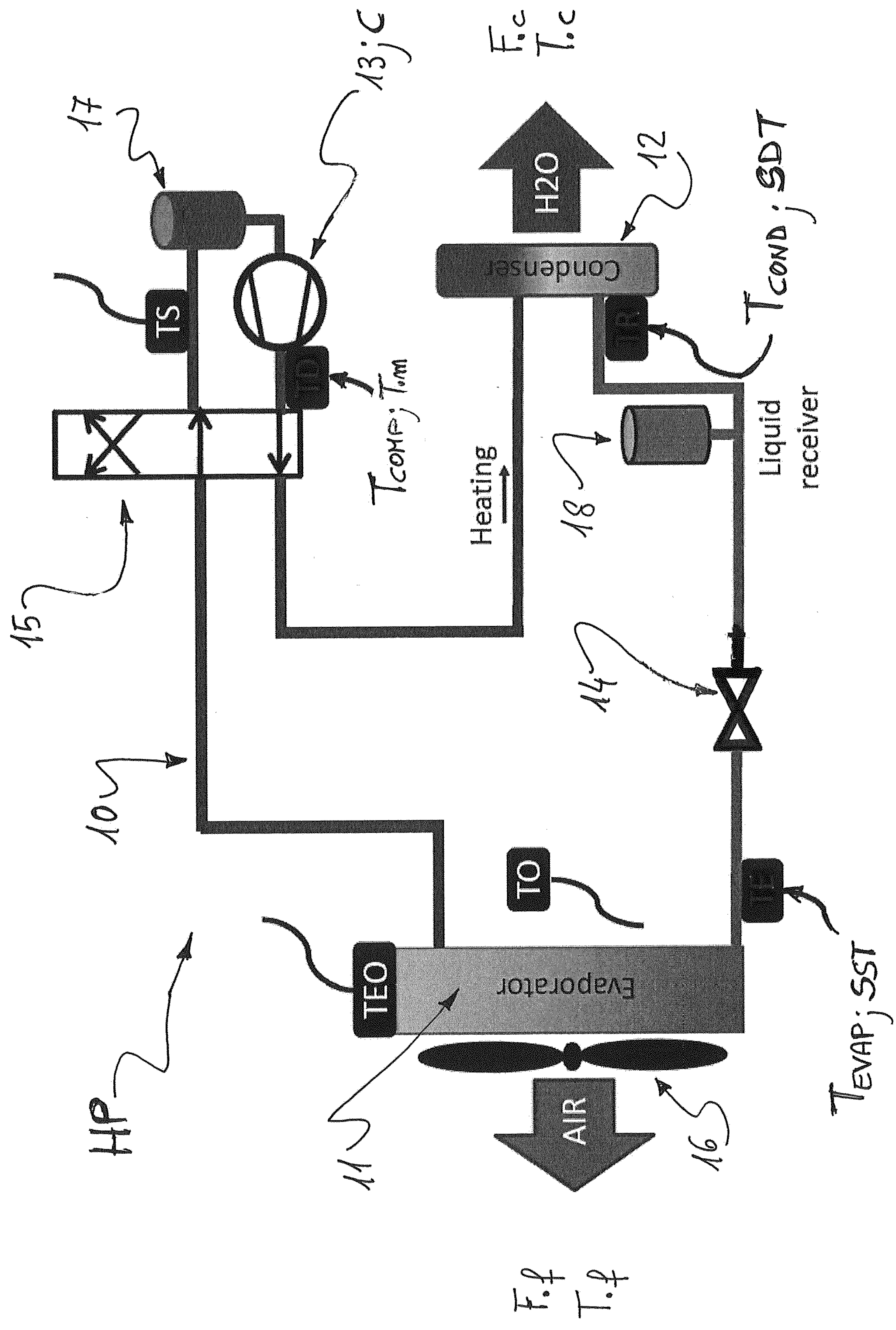

R32 – Refrigerant circuit
R410A – Refrigerant circuit
SH = 5°C
SBC = 8 °C
Same volumetric efficiency
(a)
Using R32 at same Evap. and Cond. Conditions discharge temperature is significantly bigger
Cond.
Evap.
T [°C]
s [kJ/(kg K)]
Max discharge temperature 115 °C
(b)
Using R32 at same Evap. and discharge temperature the condensing temperature is lower
Evap.
T [°C]
s [kJ/(kg K)]
(c)
Using R32 at same Cond. and discharge temperature the evaporating temperature is higher
Cond.
T [°C]
s [kJ/(kg K)]

Fig. 4



HP
10
11
12
13; C
14
15
16
17
18
TS
TD
TR
TO
TEO
TE
Condenser
Evaporator
H2O
AIR
Heating
Liquid receiver
T_{COMP}; T.m
T_{COND}; SDT
T_{EVAP}; SST
F.c
T.c
F.f
T.f

Fig. 5

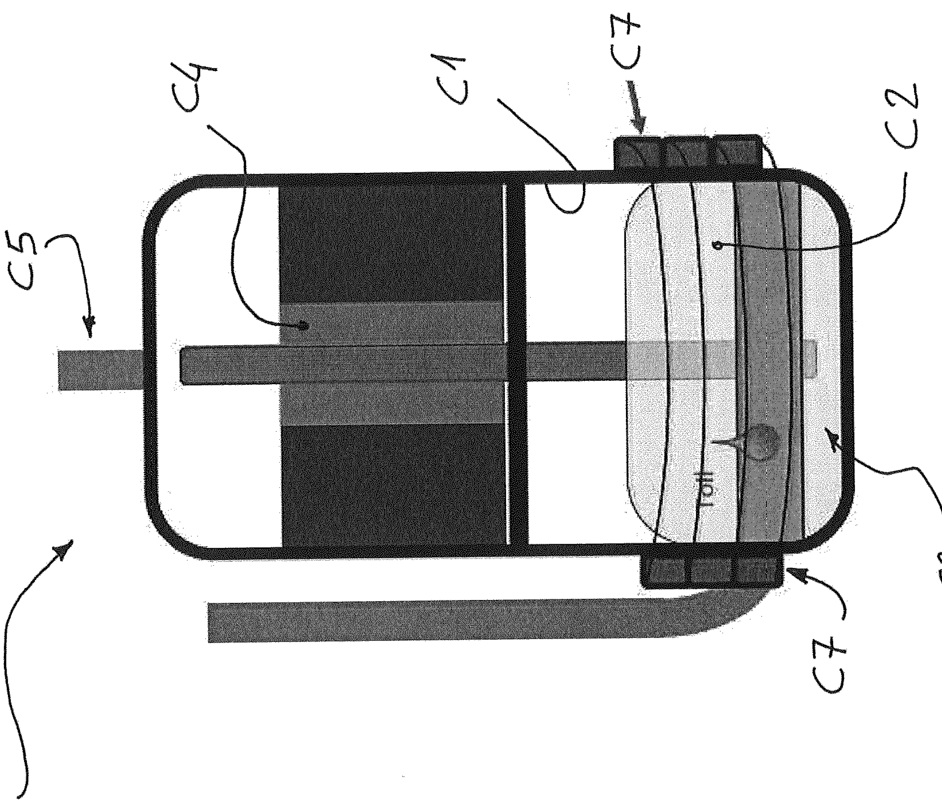


Fig. 7

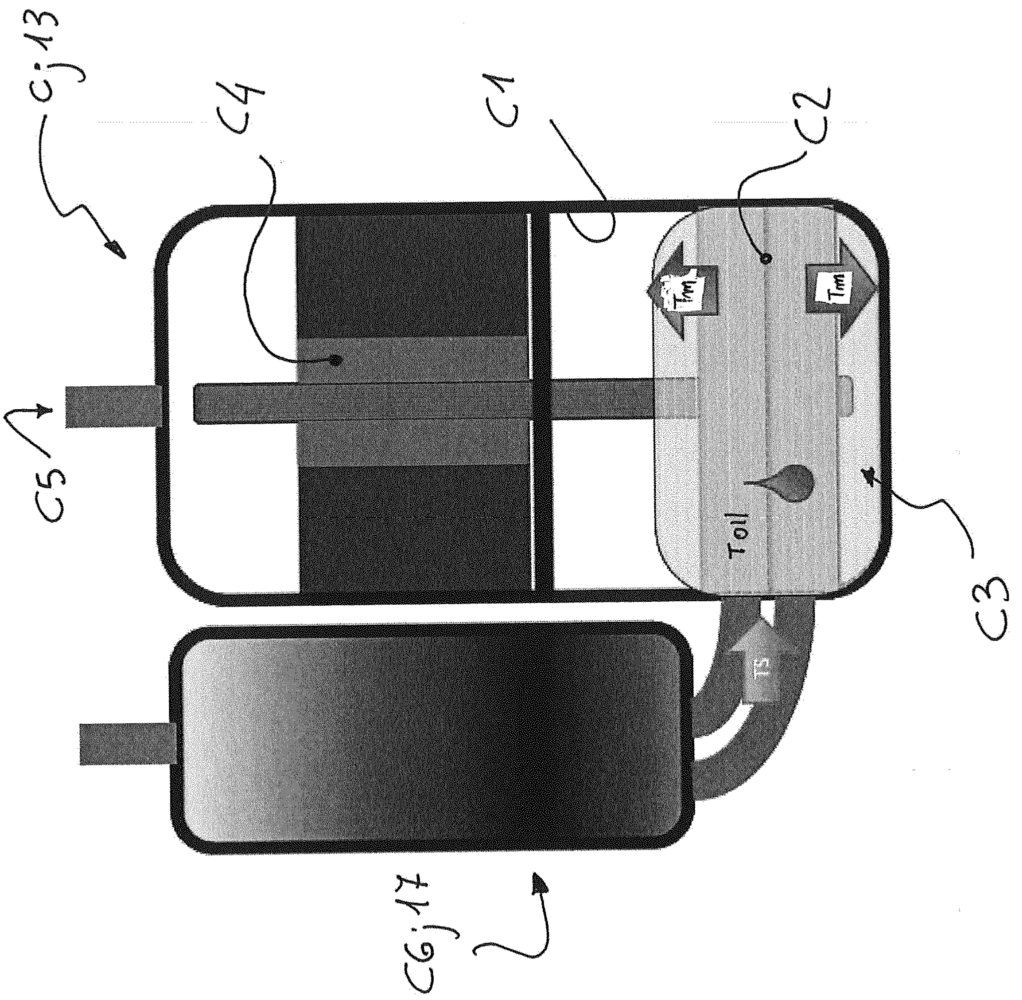


Fig. 6

REFERENCES CITED IN THE DESCRIPTION

Patent documents cited in the description

• JP 6594698 B **[0030]**