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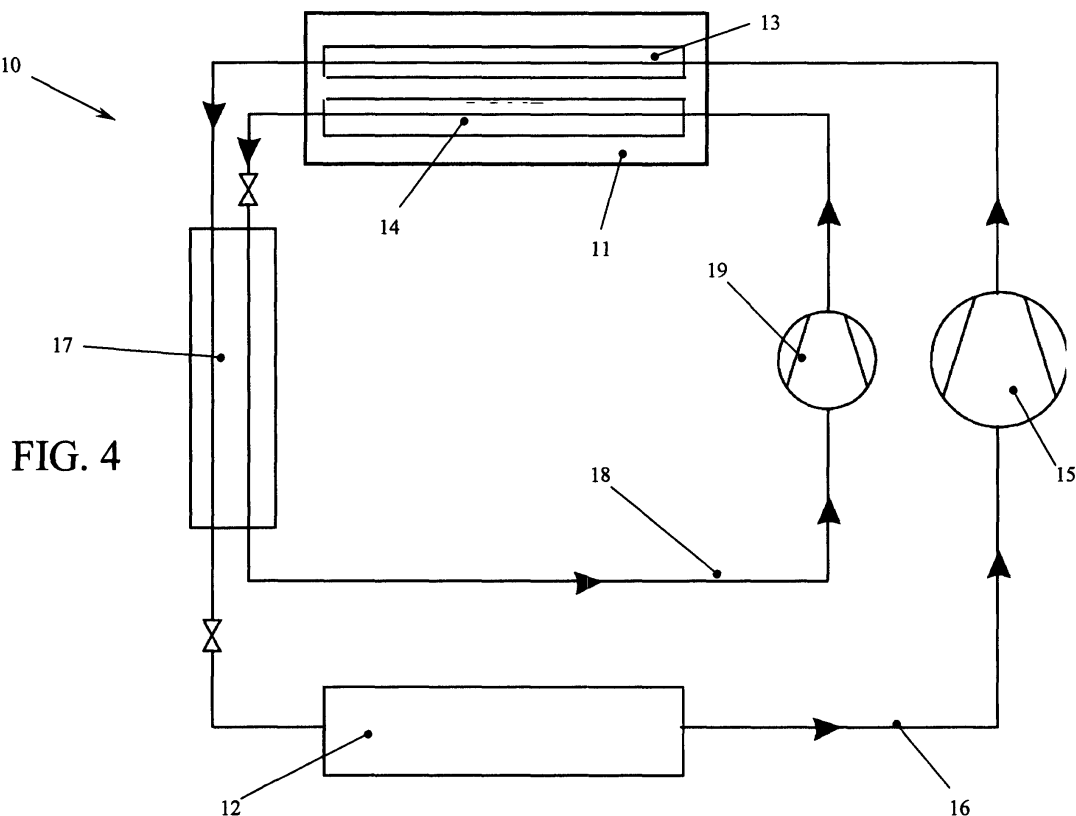
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(54) **Improved refrigerating system with fractioned expansion**

(57) An improved refrigerating system (10) with fractioned lamination is disclosed, comprising at least one main circuit (16) of a first cold-generating fluid and at least one auxiliary circuit (18) of a second cold-generating fluid, the main circuit (16) comprising in line at least

one evaporator (12), at least one compressor (15), at least one first condenser (13) and at least one under-cooling exchanger (17), and the auxiliary circuit (18) comprising in line at least one auxiliary compressor (19), at least the second condenser (14) and the under-cooling exchanger (17).



Description

[0001] The present invention refers to an improved refrigerating system with fractioned lamination.

[0002] As known, the refrigeration sector has a very wide use, which cannot now be done without, in numerous applications among which, as importance and capillary diffusion, the maintenance of the cold chain for foodstuff must be pointed out.

[0003] In particular, the commercial refrigeration field, particularly important given the high increase of companies linked thereto, is subjected, after years without changes, to a transition period due to the occurrence of known environmental problems, such as ozone layer reduction and greenhouse effect.

[0004] Currently, above all in the European Community, due to the prohibition of HCFC fluids, the majority of refrigerating systems for commercial food cold of the plug-in type use the zeotropic HFC mixture R404A or the similar azeotropic mixture R507, since this refrigerating agent has a good frigorific volumetric yield, which allows obtaining adequate frigorific powers at reduced costs (also with respect to R134a, another HFC refrigerating agent used in this field, but typically for small powers and with decreasing success).

[0005] However, due to the high GWP (Global Warming Potential), or direct greenhouse effect, of such HFC, the recent legal developments, above all at European Community level, have been addressed to limiting the use of such fluids even if, since there are no valid and recognised technologic alternatives, so far they have referred above all to mobile conditioning, imposing fluids with GWP less than 150, deeming such threshold an environmentally acceptable limit.

[0006] Among HFC, only R152a is acceptable. This however is a flammable gas, even if weakly so (A2 category according to Ashrae).

[0007] Among natural refrigerating agents, instead, which satisfy such condition, only ammonia, hydrocarbons and carbon dioxide are potentially usable in commercial refrigeration. However, if toxicity and/or flammability of the first two fluids is taken into account, it becomes necessary, in case of charges greater than 150 g allowed by laws, to adopt centralised systems of the indirect type, with all problems linked to installation costs and energy efficiency, above all in case of low-temperature or multiple-circuit systems. It must therefore be understood how the use of carbon dioxide is one of the few, if not the only way which can be used.

[0008] However, CO₂ characteristics are still currently under study and development and it has not been ascertained whether negative and positive aspects of such fluid allow its reasonable use in terms of indirect contribution meant as energy efficiency; CO₂ only solves the direct effect problem, while for the indirect effect, current technical opinions are rather controversial and it has still not been univocally defined whether the better efficiency in thermal exchange and compression processes compensates for the low thermodynamic efficiency of the basic cycle.

[0009] It is anyway certain that, currently, and it is provided also in the short term, components to be used in such technologic CO₂ applications have not yet reached an enough maturity degree.

[0010] Consequently, the efficiency improvement of traditional cycles can be the most rational medium - short term solution.

[0011] In fact, studies dealing with TEWI (Total Equivalent Warming Impact, index which takes into account direct and indirect effects) report that, for current systems, differently from centralised systems which give high fluid losses, the indirect effect appears as dominant, from which it is deduced that improving the energy efficiency does not mean only having a money gain, but instead mostly solving the environmental problem.

[0012] Substantially, it is possible to obtain a meaningful reduction of the greenhouse effect with the use (namely the replacement) of refrigerating fluids with low GWP only in systems characterised by high fluid leakages in the atmosphere, such as for example current air conditioning systems in car and in centralised industrial refrigeration. Vice versa, compact refrigerating machines used for commercial refrigeration of the plug-in type, in civil conditioning and in home conditioning have strong airtightness characteristics and low refrigerating charge. In this case, therefore, the refrigerating agent GWP is scarcely influent, while the electric energy consumption effect becomes preponderant. Under such condition, a big reduction of the greenhouse effect can be thereby obtained only by increasing the energy efficiency of the refrigerating machine.

[0013] Therefore, object of the present invention is providing an improved refrigerating system, able to improve the environmental compatibility with respect to current products.

[0014] Another object of the present invention is providing an improved refrigerating system which is able to use cold-generating mixtures R404A or R507 with a high efficiency increase and a reduction of the global frigorific power used with respect to similar systems belonging to the prior art.

[0015] The above and other objects and advantages of the invention, as will appear from the following description, are obtained with an improved refrigerating system as disclosed in claim 1. Preferred embodiments and non-trivial variations of the present invention are the subject matter of the dependent claims.

[0016] The present invention will be better described by some preferred embodiments thereof, provided as a non-limiting example, with reference to the enclosed drawings, in which:

- FIG. 1a and 1b show thermal cycle diagrams related to fluids with different specific heat value;
- FIG. 2 shows a block diagram related to a refrigerating system belonging to the prior art;
- FIG. 3 shows a block diagram related to another refrigerating system belonging to the prior art; and
- FIG. 4 shows a block diagram related to a preferred embodiment of the improved refrigerating system with fractioned lamination according to the present invention.

[0017] As can be appreciated in the following Table 1, in which main thermodynamic characteristics and COP (Coefficient of Performance) values of the theoretical thermodynamic cycle - 30/+30°C of some cold-generating mixtures are listed, mixture R404A, commonly used in commercial refrigeration, is characterised by COP values which are appreciably lower than those obtained with commonly-used fluids in air conditioning or in home refrigeration.

[0018] It must further be taken into account that penalties further increase if the range between extreme cycle temperatures is widened. It must be noted that, when there are "glides" in Table 1, condensation temperatures and p.e.n. are the arithmetic mean of bubble and dew temperatures, while the evaporation temperature is the arithmetic mean between end-of-lamination point and dry saturated steam point.

Table 1

Fluid	Mol. mass	Crit. T [°C]	p.e.n. (glide) [°C]	Cond. P.(*) [bar]	Evap. P. (*) [bar]	COP
R22	86,47	96,2	-40,8	11,9	1,64	3,08
R717	17,03	133,0	-33,3	11,7	1,19	3,14
R407C R32/125/134a (23/25/52)	86,20	86,7	-40,2 (6,9)	12,7	1,54	2,98
R410A R32/125 (50/50)	72,58	72,5	-51,4 (<0,1)	18,9	2,73	2,95
R290	44,10	96,8	-42,1	10,8	1,68	2,99
R744	44,01	31,06	-87,9	72,05	14,26	1,77
R12	120,91	112,0	-29,8	7,47	1,01	3,07
R134a	102,03	101,1	-26,1	7,70	0,84	3,02
R600a	58,12	135,0	-11,8	4,04	0,46	3,04
R11	137,37	198,0	23,7	1,26	0,09	3,31
R502 R22/115 (48,8/51,2)	111,63	82,2	-45,4	13,22	1,98	2,80
R404A R125/143a/134a (44/52/4)	97,60	72,1	-46,2 (0,8)	14,5	2,08	2,69
R507 R125/143a (50/50)	98,86	70,8	-47,1	14,82	2,17	2,67

[0019] COP data shown in Table 1 refer to a simple ideal thermodynamic cycle, characterised by reversible transformations, apart from iso-enthalpic lamination, which is typically irreversible, in addition to absence of overheating in steam sucked by the compressor and absence of under-cooling in liquid going out of the condenser; anyway, it can be deemed that in a real system, energy efficiency is lower for all fluids, but in about the same ratio which exists for the ideal cycle.

[0020] The main cause of lower performance in fluids R404A and R507 is the particular shape of the limit curve which can be seen in the diagram in FIG. 1b, which is slanted forwards and asymmetrical, different from the fluid curve shown in the diagram in FIG. 1a: such shape derives from a rather high ratio between fluid specific heat and latent vaporisation heat, and implies a low overheating value as regards compression. This is the main reason why the mentioned fluids are typically used in commercial refrigeration, but such positive characteristic is accompanied by an increase of energy loss in lamination process, graphically shown by the trapezoid-like area which contains the iso-enthalpic lamination transformation.

[0021] The classic method for reducing lamination losses is the fractioned lamination, characteristic of the two-stage

refrigerating cycle, which is typically realised by the system 20, schematically shown in FIG 2, equipped, as known, of an intermediate separator ("open flash tank") 25 arranged between a first lamination stage, comprising in line an evaporator 21 and a low-pressure compressor 23, and a second lamination stage, comprising in line a high-pressure compressor 27 and a condenser 29. The energy advantage deriving from a fractioned lamination is evident without resorting to energy analysis. In fact, deriving from the intermediate separator 25 the steam developed in the first lamination stage and compressing it at the condensation pressure with the high-pressure compressor 27 means avoiding the further expansion of this amount of steam up to the evaporation pressure, such operation not adding a refrigerating effect, but increasing the low-pressure compressor 23 work.

[0022] A two-stage refrigerating system 30, as schematically shown in FIG. 3, is preferred to such system 20, due to its simple arrangement and lower costs, such system 30 being equipped with a "compound" compressor 33 and a under-cooling exchanger 35 arranged in parallel between evaporator 31 and condenser 37; as known, a "compound" compressor is a single compressor using part of the cylinders as low pressure stage and part as high pressure stage. This variation, generally used in systems operating at low temperature with alternative compressors, offers the same thermodynamic advantages of the classic scheme of system 20 and approximately the same energy efficiency, and provides for the use of the "compound" compressor 33 integrated with the under-cooling exchanger 35 of the condensate, replacing the intermediate separator 25 being present in system 20; the compressed steam at intermediate pressure is de-overheated for mixing with the two-phase liquid going out of the valve 39.

[0023] Such system diagram enables a very high COP increase, but it cannot be used in practice in small-sized systems, in which it is not possible to arrange two-stage compressors of the "compound" type and the use of separate compressors could create serious problems for oil return, being placed between the two carters at different pressures.

[0024] Instead, with reference to FIG. 4, it is possible to note a schematic representation of a preferred embodiment of the improved refrigerating system 10 with fractioned lamination according to the present invention. In particular, the system 10 comprises at least one main circuit 16 of a first cold-generating fluid and at least one auxiliary circuit 18 of a second cold-generating fluid, the main circuit 16 comprising in line at least one evaporator 12, at least one compressor 15, at least one first condenser 13 and at least one under-cooling exchanger 17, and the auxiliary circuit 18 comprising in line at least one auxiliary compressor 19, at least one second condenser 14 and at least the under-cooling exchanger 17, operating as evaporator for the auxiliary circuit 18; as realised, the system 10 advantageously allows performing the fractioned lamination with reliable and low-cost technologic solutions, also for systems with small potentiality. In fact, a lamination loss reduction equivalent to the one obtained with known systems shown in FIG. 2 and 3 is obtained with the system 10 of the present invention by under-cooling the condensate, using the refrigerating effect of the auxiliary circuit 18 completely separated from the main circuit 16. The principle is the same one used in the cycle known as "economizer", but, differently therefrom, the auxiliary compressor 19 is used, instead of a compressor equipped with the economiser tap: such possibility is in fact reserved almost exclusively to screw compressors. It must be noted that, for a simple representation, the under-cooling exchanger 17 is shown in FIG. 4 as equi-current, while in reality the counter-current arrangement is preferable.

[0025] The following Table 2 includes the main performance data of the system 10 according to the present invention when Evap. T. of the auxiliary circuit coincides with Und. Liq. T. and percentage COP increases obtained for an ideal cycle (defined similar to the one in FIG. 1a and 1b) depending on different operating conditions, together with the percentage reduction of the sum of volumes sucked by compressors 15 and 19, with respect to sucked volume in traditional cycles.

Table 2

Condensation Temperature = 30°C				
Evaporation T. [°C]	-10.00	-20.00	-30.00	-40.00
Under-Cooled Liquid T. [°C]	10.00	5.00	0.00	-5.00
Auxiliary Compressor Cycle COP	5.67	4.27	3.35	2.69
Conventional Cycle COP	5.00	3.63	2.73	2.10
COP Increase [%]	13.43	17.70	22.54	28.06
Sucked Volume Reduction [%]	-10.45	-15.23	-20.49	-26.05
Sucked Vol. Ratio. (aux/main)	0.13	0.13	0.13	0.12
Condensation Temperature = 40°C				
Evaporation T. [°C]	-10.00	-20.00	-30.00	-40.00

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(continued)

Condensation Temperature = 40°C				
Under-Cooled Liquid T. [°C]	15.00	10.00	5.00	0.00
Auxiliary Compressor Cycle COP	4.30	3.37	2.71	2.22
Conventional Cycle COP	3.58	2.68	2.05	1.59
COP Increase [%]	20.26	25.79	32.13	39.53
Sucked Volume Reduction [%]	-16.52	-22.11	-28.03	-34.11
Sucked Vol. Ratio (aux/main)	0.16	0.16	0.15	0.13
Condensation Temperature = 50°C				
Evaporation T. [°C]	-10.00	-20.00	-30.00	-40.00
Under-Cooled Liquid T. [°C]	20.00	15.00	10.00	5.00
Auxiliary Compressor Cycle COP	3.34	2.69	2.20	1.83
Conventional Cycle COP	2.56	1.95	1.50	1.16
COP Increase [%]	30.59	38.18	47.14	57.90
Sucked Volume Reduction [%]	-24.55	-30.93	-37.48	-44.05
Sucked Vol. Ratio (aux/main)	0.20	0.19	0.17	0.15

[0026] Such data have been obtained assuming an ideal under-cooling exchanger, namely such as to cool the cold-generating fluid till the bubble temperature of the fluid evaporating in the auxiliary circuit.

[0027] This assumption represents a limit condition: for a more realistic estimate, the following Table 3 includes the values obtained assuming a difference of 4°C between fluid temperature and auxiliary cycle evaporation temperature, which can be deemed realistic for the type of exchanger taken into account and its operating conditions.

[0028] In both cases, the under-cooled liquid temperature has been fixed as arithmetic mean of extreme cycle temperatures, according to a classic approximate optimisation criterion; it has been anyway verified that, in any case, the performance data computed thereby are practically coincident with those corresponding to the actual optimisation.

Table 3

Condensation Temperature = 30°C				
Evaporation T. [°C]	-10.00	-20.00	-30.00	-40.00
Under-Cooled Liquid T. [°C]	10.00	5.00	0.00	-5.00
Auxiliary Compressor Cycle COP	5.54	4.18	3.27	2.63
Conventional Cycle COP	5.00	3.63	2.73	2.10
COP Increase [%]	10.84	15.01	19.72	25.10
Sucked Volume Reduction [%]	-8.99	-13.76	-19.09	-24.80
Sucked Vol. Ratio (aux/main)	0.14	0.15	0.15	0.14
Condensation Temperature = 40°C				
Evaporation T. [°C]	-10.00	-20.00	-30.00	-40.00
Under-Cooled Liquid T. [°C]	15.00	10.00	5.00	0.00
Auxiliary Compressor Cycle COP	4.19	3.28	2.64	2.16
Conventional Cycle COP	3.58	2.68	2.05	1.59
COP Increase [%]	17.14	22.51	28.67	35.85
Sucked Volume Reduction [%]	-14.88	-20.55	-26.61	-32.88

(continued)

Condensation Temperature = 40°C				
Sucked Volume Ratio (aux/main)	0.18	0.18	0.17	0.15
Condensation Temperature = 50°C				
Evaporation T. [°C]	-10.00	-20.00	-30.00	-40.00
Under-Cooled Liquid T. [°C]	20.00	15.00	10.00	5.00
Auxiliary Compressor Cycle COP	3.24	2.61	2.14	1.77
Conventional Cycle COP	2.56	1.95	1.50	1.16
COP Increase [%]	26.63	33.97	42.64	53.03
Sucked Volume Reduction [%]	-22.76	-29.30	-36.06	-42.87
Sucked Volume Ratio (aux/main)	0.23	0.22	0.20	0.18

[0029] Data in Table 3 must therefore be deemed a reliable estimate of what can be obtained in practice: since COP gains and global displacement reduction (more exactly reduction of the steam volume sucked per refrigerating effect unit) are expressed in percentage form, they do not depend on efficiency characteristics of considered compressors 15 and 19, provided that both compression iso-entropic efficiency and volumetric efficiency are the same for all.

[0030] In general, the first cold-generating fluid and the second cold-generating fluid could both have mixture HFC R404A. The reference to R404A is due to peculiarities of the commercial sector towards which the system 10 is mainly aimed and the obtained benefits. However, other refrigerating agents can be used, with different benefits depending on fluid and application type.

[0031] In order to provide examples about differences among fluid with the same working conditions, the following Table 4 includes performance data of the system 10 operating with other refrigerating agents used for the commercial refrigeration (for Auxiliary Circuit Evaporation T. equal to Under-Cooled Liquid T.).

Table 4

Condensation Temperature = 40°C				
Fluids	R404A	R507	R134a	R290
Evaporation T. [°C]	-30.00	-30.00	-30.00	-30.00
Under-Cooled Liquid T. [°C]	5.00	5.00	5.00	5.00
Auxiliary Compressor Cycle COP	2.71	2.68	2.91	2.88
Conventional Cycle COP	2.05	2.01	2.38	2.35
COP Increase [%]	32.13	33.35	22.24	22.58
Sucked Volume Reduction [%]	-28.03	-28.79	-22.30	-21.05
Sucked Volume Ratio (aux/main)	0.15	0.15	0.09	0.11

[0032] Using the system 10 according to the present invention, it is therefore possible to obtain high energy efficiency increases with modest increases in system costs; in fact, the first and second condensers 13 and 14 could be integrated in a same battery 11, possibly winged, composed of separate circuits respectively for main circuit 16 and auxiliary circuit 18, cooling side; with the same refrigerating power, its exchange surface is anyway less than the one of the reference system, since heat to be globally discharges to the outside is less, due to the COP increase effect.

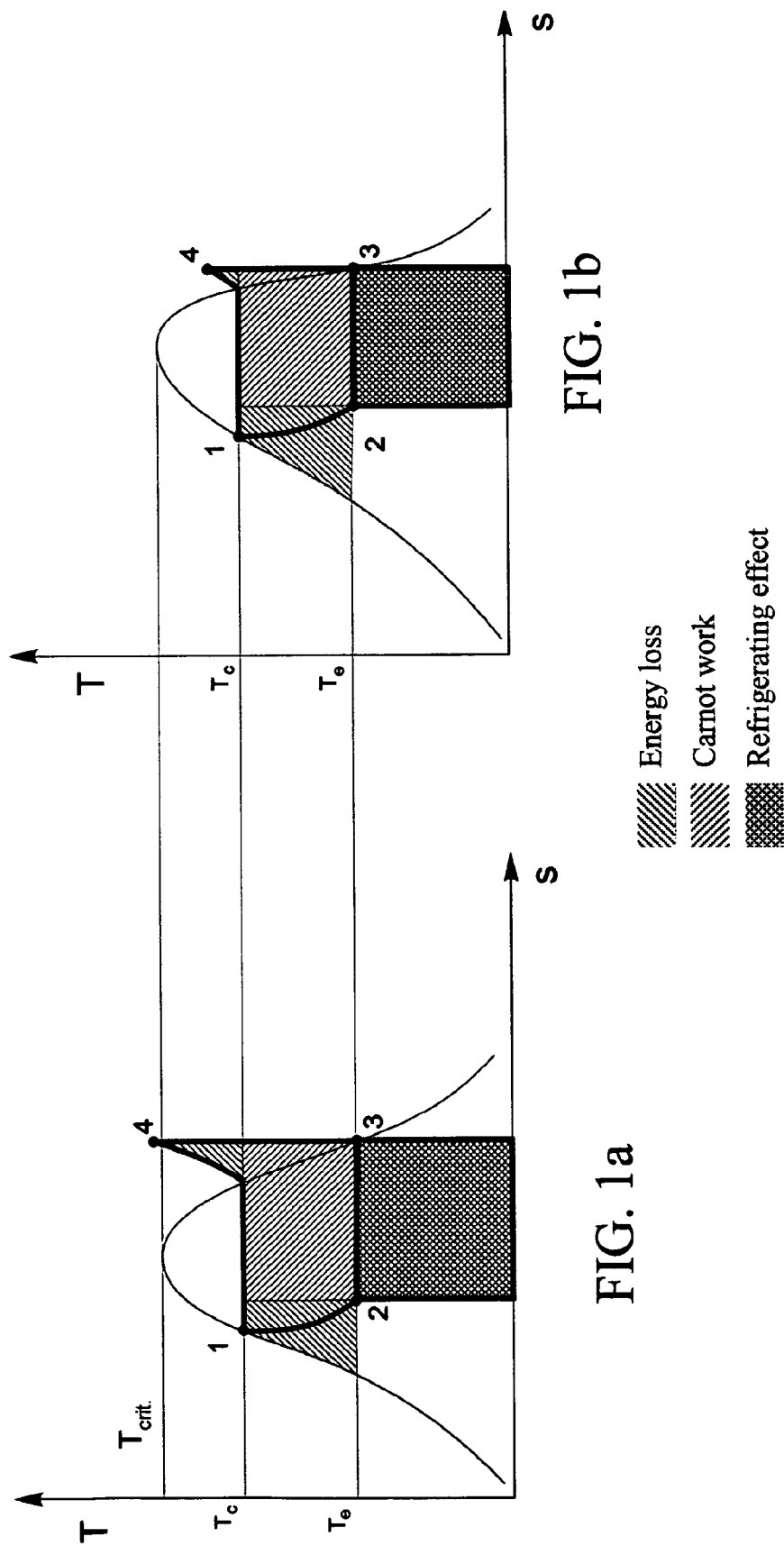
[0033] Moreover, with the same refrigerating power, the cost of the two compressors 15, 19 can be less than the cost of the single compressor used in the conventional cycle; the auxiliary compressor 19 in fact is a compressor for medium temperature service and implies a reduced cost, above all when mass-produced components are used; for such compressor, it could be convenient, both from the energy efficiency and from the costs points of view, to use a different fluid from 404A, for example R134a. The high reduction of volume flowrate of the steam sucked by compressors, with the same refrigerating power, must also be taken into account: for such purpose, the displacement of the auxiliary compressor 19 is much greater than what appears from the relationship between volume flow-rates as pointed out in Tables 2 and 3, since such compressor works with much higher volume efficiencies than those of the main compressor. In addition,

the cost of the under-cooling exchanger 17, probably of the coaxial type, is low, being it a very simple element. Alternatively, it is wholly clear that the under-cooling exchanger 17 can be of another type, generally with surface, such as for example with mantle or plates, without therefore departing from the scope of the present invention.

[0034] A further advantage of the thermodynamic cycle realised as an example with the system 10 according to the present invention is its applicability independently from the efficiency of basic components, which compose the system 10 itself. Therefore, developments in the components, as normally occurs for example for compressors, do not imply a smaller advantage for the proposed cycle, in percentage terms with respect to the basic cycle.

Claims

1. Refrigerating system (10) with fractioned lamination **characterised in that** it comprises at least one main circuit (16) of a first cold-generating fluid and at least one auxiliary circuit (18) of a second cold-generating fluid, said main circuit (16) comprising in line at least one evaporator (12), at least one compressor (15), at least one first condenser (13) and at least one under-cooling exchanger (17), and said auxiliary circuit (18) comprising in line at least one auxiliary compressor (19), at least one second condenser (14) and at least said under-cooling exchanger (17).
2. System (10) according to claim 1, **characterised in that** said first cold-generating fluid is a zeotropic HFC mixture R404A.
3. System (10) according to claim 1, **characterised in that** said first cold-generating fluid is an azeotropic HFC mixture R507.
4. System (10) according to claim 1, **characterised in that** said first cold-generating fluid is a HFC mixture HFC R290.
5. System (10) according to claim 1, **characterised in that** said first cold-generating fluid is a HFC mixture R134a.
6. System (10) according to claim 1, **characterised in that** said second cold-generating fluid is a zeotropic HFC mixture R404A.
7. System (10) according to claim 1, **characterised in that** said second cold-generating fluid is an azeotropic HFC mixture R507.
8. System (10) according to claim 1, **characterised in that** said second cold-generating fluid is a HFC mixture R290.
9. System (10) according to claim 1, **characterised in that** said second cold-generating fluid is a HFC mixture R134a.
10. System (10) according to claim 1, **characterised in that** said auxiliary compressor (19) is a compressor for medium temperature service.
11. System (10) according to claim 1, **characterised in that** said under-cooling exchanger (17) is of a surface type.
12. System (10) according to claim 1, **characterised in that** said under-cooling exchanger (17) is of a coaxial type.
13. System (10) according to claim 1, **characterised in that** said under-cooling exchanger (17) is of a mantle type.
14. System (10) according to claim 1, **characterised in that** said under-cooling exchanger (17) is of a plate type.
15. System (10) according to claim 1, **characterised in that** said main circuit (16) and said auxiliary circuit (18) are in equi-current in said under-cooling exchanger (17).
16. System (10) according to claim 1, **characterised in that** said main circuit (16) and said auxiliary circuit (18) are in counter-current in said under-cooling exchanger (17).
17. System (10) according to claim 1, **characterised in that** said first and second condensers (13, 14) are integrated in a single battery (11) composed of separate circuits respectively for said main circuit (16) and for said auxiliary circuit (18), cooling side.



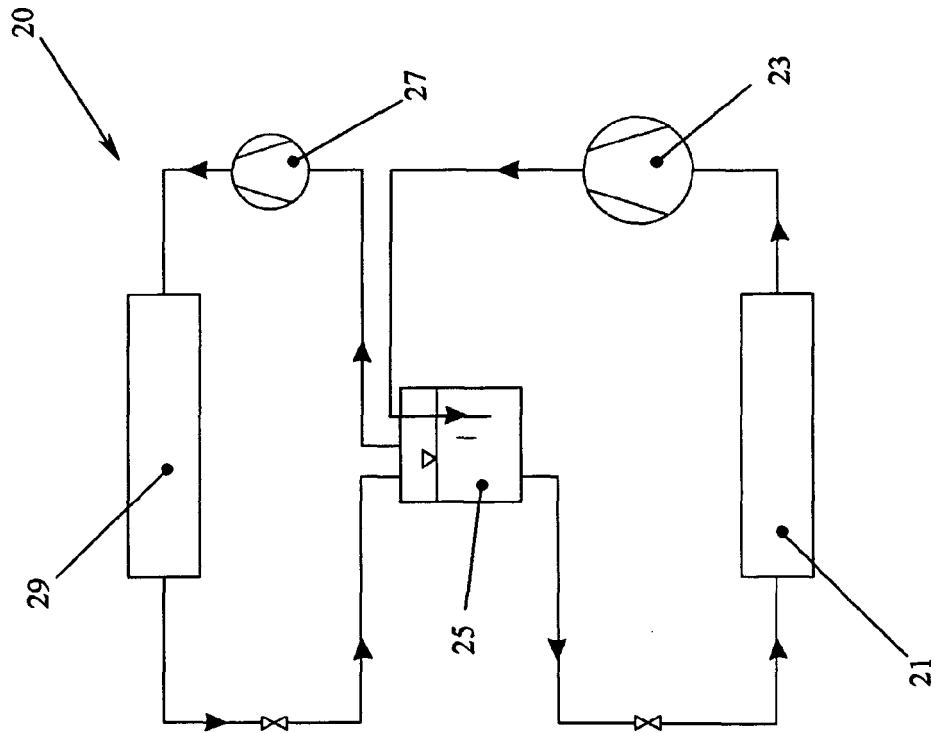


FIG. 2

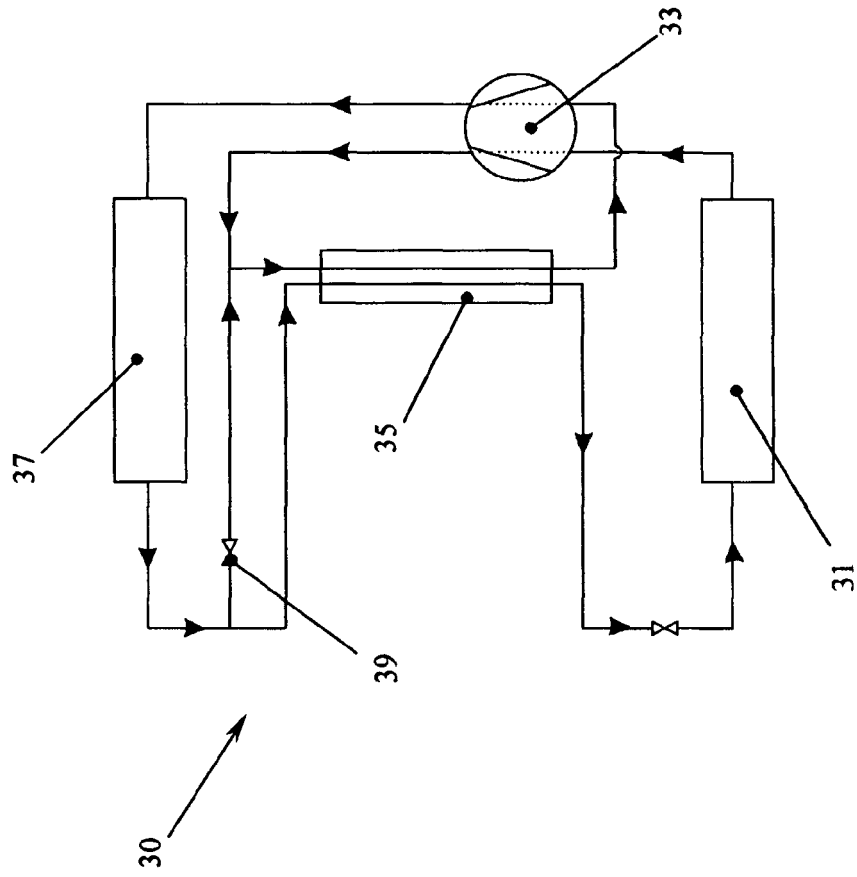


FIG. 3

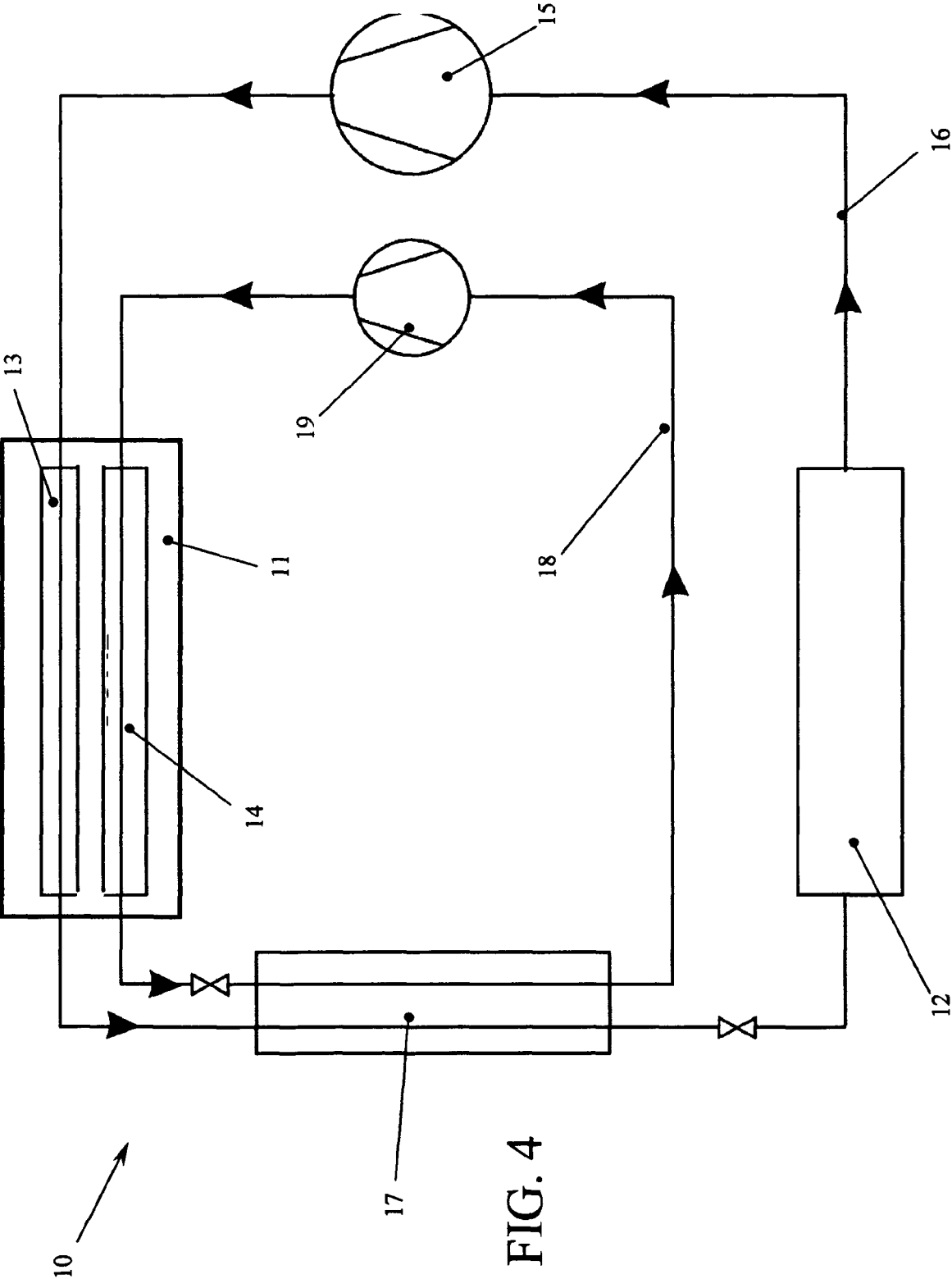


FIG. 4