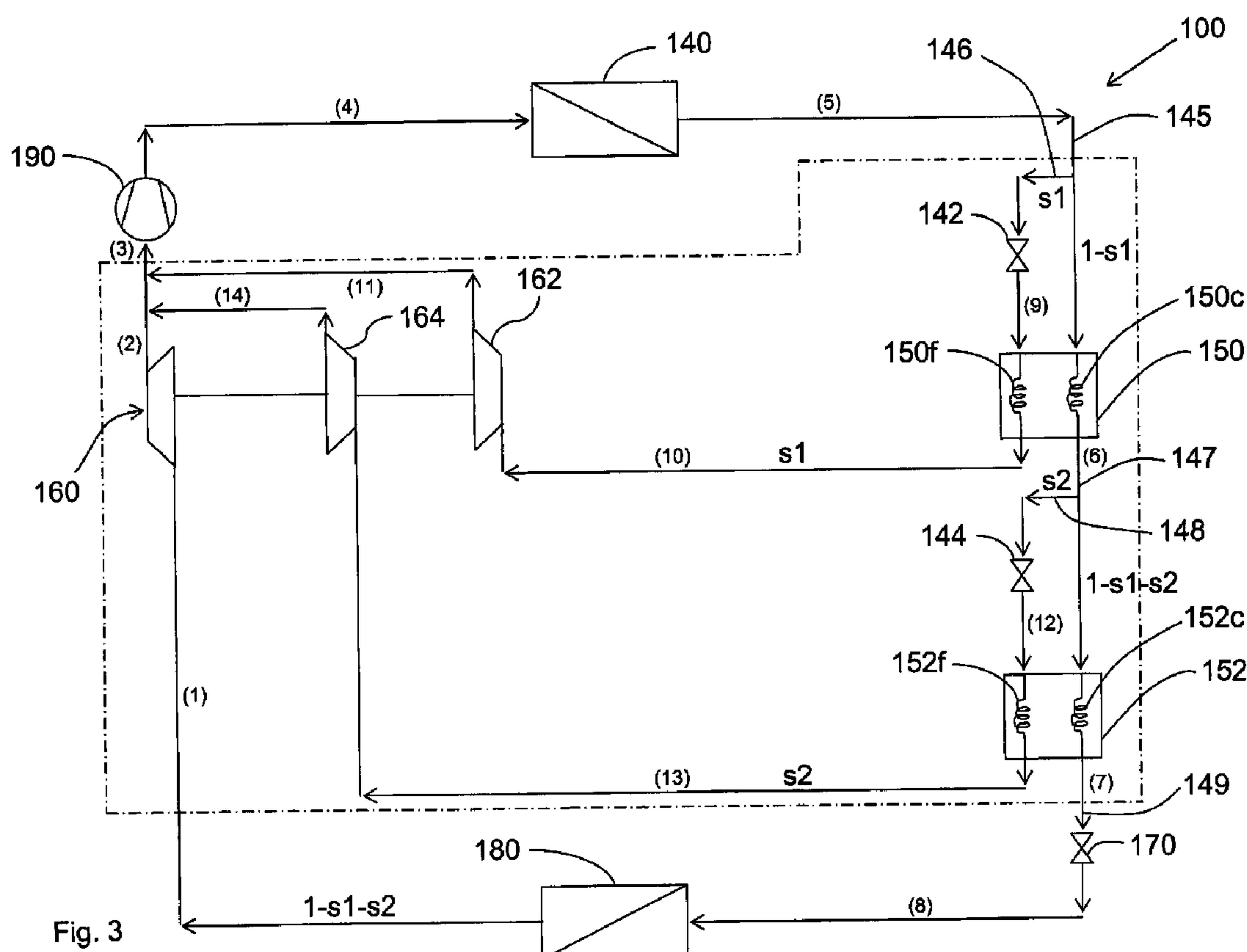




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(54) Titre : DISPOSITIF DE REFRIGERATION ET PROCEDE POUR FAIRE CIRCULER UN FLUIDE DE
REFRIGERATION ASSOCIE A CELUI-CI
(54) Title: REFRIGERATING DEVICE AND METHOD FOR CIRCULATING A REFRIGERATING FLUID ASSOCIATED
WITH IT



(57) Abrégé/Abstract:

Refrigerating device formed by a main compressor (190), a condenser (140) downstream of and in fluid communication with the main compressor (190), main expansion means (170) downstream of the condenser (140) and an evaporator (180) downstream

(57) **Abrégé(suite)/Abstract(continued):**

of and in fluid communication with the main expansion means (170), which also comprises a turbocompressor unit (160) in fluid communication between the evaporator (180) and the main compressor (190) and a heat exchanger (150, 152) having a hot branch (150c) connected upstream, via an inlet line (145), to the condenser (140) and downstream, via an outlet line (149), to the main expansion means (170) and a cold branch (150f) connected, upstream, to an expansion means (142, 144) mounted on a branch (146) of the line (145) and, downstream, to a turbine portion (162) of the turbocompressor unit (160). The invention also relates to a method for circulating a refrigerating fluid inside the abovementioned device.

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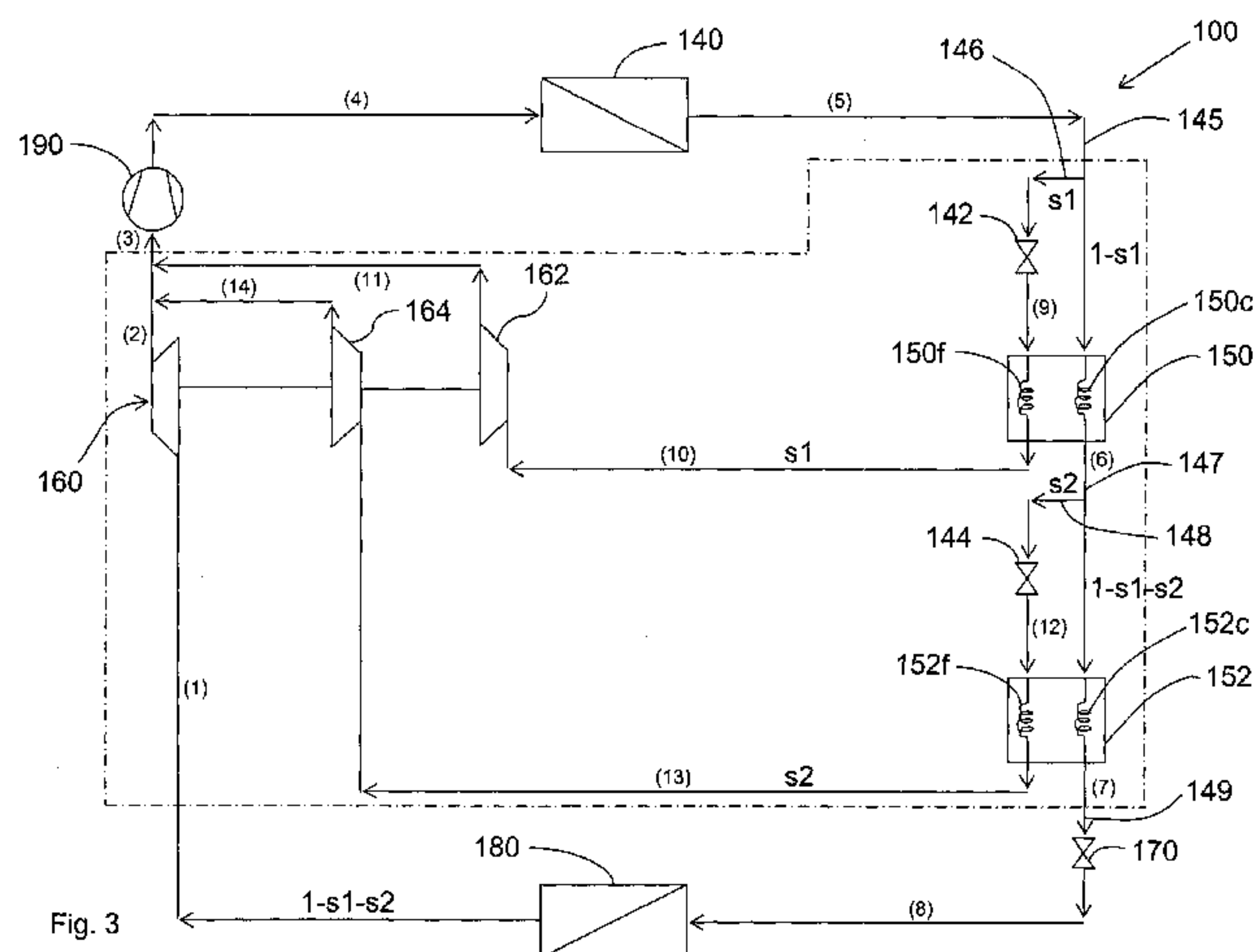


Fig. 3

(57) Abstract: Refrigerating device formed by a main compressor (190), a condenser (140) downstream of and in fluid communication with the main compressor (190), main expansion means (170) downstream of the condenser (140) and an evaporator (180) downstream of and in fluid communication with the main expansion means (170), which also comprises a turbocompressor unit (160) in fluid communication between the evaporator (180) and the main compressor (190) and a heat exchanger (150, 152) having a hot branch (150c) connected upstream, via an inlet line (145), to the condenser (140) and downstream, via an outlet line (149), to the main expansion means (170) and a cold branch (150f) connected, upstream, to an expansion means (142, 144) mounted on a branch (146) of the line (145) and, downstream, to a turbine portion (162) of the turbocompressor unit (160). The invention also relates to a method for circulating a refrigerating fluid inside the abovementioned device.

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REFRIGERATING DEVICE AND METHOD FOR CIRCULATING A
REFRIGERATING FLUID ASSOCIATED WITH IT

DESCRIPTION

Technical field of the invention

5 The present invention relates to a refrigerating device, in particular suitable for circulating a fluid in industrial refrigerating plants as well as in household air-conditioning systems, and to a method for circulating a refrigerating fluid associated with it.

10 Description of the prior art

 In general, a device for circulating a refrigerating fluid includes a compressor designed to compress the refrigerant in the gaseous state, giving it a higher temperature and pressure value; a condenser
15 able to condense the compressed gaseous refrigerant with consequent conversion thereof into the liquid state and release of heat to the external environment; an expansion unit, for example a capillary tube or an isoenthalpic throttling valve, intended to lower the
20 temperature and the pressure of the refrigerant; and an evaporator, which absorbs heat from the external environment, cooling it, and transfers it to the refrigerating fluid at a low temperature and pressure received from the expansion unit, said fluid passing
25 from the liquid state into the vapour state.

 During recent years many attempts have been made to increase the performance of the refrigerating devices. Some have encountered obstacles of a technological nature, which have prejudiced the
30 feasibility thereof, while others have brought advantages in terms of increased efficiency, while significantly complicating, however, the plant. An

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example in this connection consists of dual-stage
compression plants where the existence of two
independent compressors causes problems of balancing of
the loads and more complex management of the entire
5 plant.

The object of the present invention is to
eliminate, or at least reduce, the drawbacks mentioned
above, by providing a refrigerating device and a method
for circulating refrigerating fluid associated with it,
10 which are improved in terms of efficiency.

According to a first aspect of the present
invention, a refrigerating device comprising a main
compressor, a condenser downstream of and in fluid
communication with said main compressor, main expansion
15 means downstream of said condenser and an evaporator
downstream of and in fluid communication with said main
expansion means is provided,

characterized in that it comprises a
turbocompressor unit connected between said evaporator
20 and said main compressor and at least one heat
exchanger having a hot branch connected upstream, via
an inlet line, to said condenser and downstream, via an
outlet line, to said main expansion means and a cold
branch connected, upstream, to an expansion means
25 mounted on a branch of said inlet line and, downstream,
to a turbine portion of said turbocompressor unit.

According to another aspect of the present
invention a method for circulating a refrigerating
fluid inside a device according to the invention is
30 provided, said method comprising the stages of:

- compressing the refrigerating fluid in a main
compressor;

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- condensing the fluid in a condenser downstream of and in fluid communication with said main compressor;

5 - expanding the fluid in main expansion means downstream of said condenser;

- evaporating the fluid in an evaporator downstream of and in fluid communication with said main expansion means;

characterized in that it comprises

10 - between said condensation stage and said expansion stage at least one stage involving heat exchange stage, inside at least one heat exchanger, between the compressed refrigerating fluid, which flows inside a hot branch of the heat exchanger, and an
15 associated amount of compressed refrigerating fluid withdrawn upstream of the heat exchanger, cooled inside an expansion means and flowing inside a cold branch of the heat exchanger; and

- between said main expansion stage and said main
20 compression stage, a stage involving pre-compression of the refrigerating fluid inside a turbocompressor unit, said pre-compression stage comprising at least one stage involving expansion, inside at least one turbine portion of the turbocompressor unit, of the bled-off
25 refrigerating fluid leaving the cold branch of the heat exchanger.

Brief description of the drawings

Characteristic features and advantages of the present invention will emerge more clearly from the
30 following detailed description of a currently preferred example of embodiment thereof, provided solely by way of a non-limiting example, with reference to the

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accompanying drawings, in which:

Figure 1 is a schematic view, which shows a refrigerating device according to the prior art;

Figure 2 shows the pressure-enthalpy diagram for the refrigerating fluid circulating inside the device of Figure 1;

Figure 3 is a schematic view of a refrigerating device according to the present invention; and

Figure 4 shows the pressure-enthalpy diagram for the refrigerating fluid circulating inside the device of Figure 3.

In the accompanying drawings, identical or similar parts and components are indicated by the same reference numbers.

Detailed description of the preferred embodiments

Figures 1 and 2 show, respectively, a refrigerating device 10 of the conventional type, which is particularly suitable for freezing alimentary products, and the p-h (pressure-enthalpy) diagram for the fluid circulating inside it. As shown, the device 10 is formed by a compressor 12, by a condenser 14 in fluid communication with the compressor 12, by an isoenthalpic throttling valve 16 in fluid communication with the condenser 14 and by an evaporator in fluid communication with the throttling valve 16, upstream, and with the compressor 12 downstream.

The refrigerating fluid, for example freon, enters into the compressor 12 in the form of superheated vapour at a low temperature and pressure, for example - 35 °C and 1.33 bar (point 1* in p-h diagram), is compressed and enters into the condenser 14 at a high pressure and temperature, for example +65 °C and 16 bar

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(point 2* in p-h diagram). Inside the condenser 14 the refrigerating fluid undergoes cooling, passing from the superheated vapour state (point 2*) into the liquid state (point 3* in p-h diagram) and releasing a quantity of heat q_{out} to the external environment. The refrigerating fluid in the liquid state, leaving the condenser 14, expands passing through the isoenthalpic throttling valve 16 and undergoing a reduction in pressure without exchanging heat with the external environment (isoenthalpic conversion). The fluid leaving the throttling member (point 4* in p-h diagram) enters into the evaporator, where it passes from the liquid state into the superheated vapour state (point 1* in p-h diagram) absorbing a quantity of heat q_{in} from the external environment.

With reference to Figure 3, which shows a preferred embodiment of the present invention, a device for circulating a refrigerating fluid, denoted generally by the reference number 100, is formed by the components of a conventional refrigerating device, namely a main condenser 140, main expansion means such as a main isoenthalpic throttling valve 170, an evaporator 180 and a main compressor 190.

The aforementioned conventional device is supplemented with certain components, enclosed ideally within a block - defined by broken lines in Figure 3 - which comprises a first and a second heat exchanger, 150, 152, respectively, for example heat exchangers of the plate or tube-bundle type, commonly used in the refrigerating sector, arranged in series between the condenser 140 and the main throttling valve 170, and a turbocompressor unit 160, inserted between the main

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compressor 190 and the evaporator 180 and provided with a compressor portion 166 and a first and second turbine portion 162, 164, which are respectively supplied by an outlet of each heat exchanger 150, 152.

5 More particularly the condenser 140 is connected, via an inlet line 145, to a circuit for refrigerating fluid at a higher temperature, referred to below as "hot branch" 150c, of the first heat exchanger 150. The inlet line 145 has, branched off it, a line 146 which
10 incorporates first expansion means, for example a first throttling valve 142, which leads into a circuit for a refrigerating fluid at a lower temperature, referred to below as "cold branch" 150f, of the first heat exchanger 150. The outlet of the hot branch 150c of
15 the first heat exchanger 150 is linked, via a connection line 147, to the inlet of a circuit for refrigerating fluid at a higher temperature, referred to below as "hot branch" 152c, of the second heat exchanger 152, while the outlet of the cold branch 150f
20 of the first heat exchanger 150 is connected to the inlet of the first turbine portion 162 of the turbocompressor unit 160.

 The line 147 connecting together the first and the second heat exchanger 150, 152 has a branch 148
25 provided with second expansion means, for example a second throttling valve 144, which leads into a circuit for refrigerating fluid at a lower temperature, referred to below as "cold branch" 152f, of the second heat exchanger 152. The outlet of the hot branch 152c
30 of the second heat exchanger is connected, via an outlet line 149, to the main throttling valve 170, while the outlet of the cold branch 152f is connected

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to the inlet of the second turbine portion 164 of the turbocompressor unit 160.

The outlet of the evaporator 180 is connected to the inlet of the compressor portion 166 of the
5 turbocompressor unit 160, the outlet of which is in fluid communication with the main compressor 190.

Below the operating principle of the device according to Figure 3 will be described with reference to the p-h diagram relating to the refrigerating fluid
10 circulating through it, shown in Figure 4. In the particular example in question, the refrigerating device is used for rapid freezing of alimentary products. For this purpose, the temperatures of the fluid circulating inside the device vary between a
15 value $T_{\min} = -40\text{ }^{\circ}\text{C}$ and a value $T_{\max} = 63.7\text{ }^{\circ}\text{C}$ and the refrigerating fluid chosen is freon. It is understood that the refrigerating device according to the present invention is suitable for many applications, for example the air-conditioning of domestic premises, so
20 that, depending on the intended use, the pressure and temperature values of the physical states 1-14, as well as the type of refrigerating fluid circulating inside the device, will vary correspondingly.

Refrigerating fluid, typically freon, at a
25 temperature $T_5 = 35\text{ }^{\circ}\text{C}$ and pressure $p_5 = 16.1\text{ bar}$ (point 5 in p-h diagram), namely in a liquid/vapour equilibrium state, flows out from the condenser 140. A portion of the refrigerating fluid flowing out from the condenser 140, referred to below as first bleed-off s1,
30 is conveyed, via the branch 146 of the line 145 into the first isoenthalpic throttling valve 142, where it is cooled down to a temperature ranging between the

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maximum temperature ($T_{\max} = 35\text{ }^{\circ}\text{C}$) and the minimum temperature ($T_{\min} = -35\text{ }^{\circ}\text{C}$) of the cycle, preferably a temperature $T_9 = 7\text{ }^{\circ}\text{C}$ (point 9 in p-h diagram; $p_9 = 7.48\text{ bar}$) and then into the cold branch 150f of the first heat exchanger 150, while the remaining portion 1-s1 of refrigerating fluid enters directly into the cold branch 150c of the heat exchanger 150 at the temperature T_5 and at the pressure p_5 .

Inside the first heat exchanger 150, the refrigerating fluid portion contained in the hot branch 150c transfers heat to the refrigerating fluid portion contained in the cold branch 150f, being cooled from $T_5 = 35\text{ }^{\circ}\text{C}$ to a temperature $T_6 = 12\text{ }^{\circ}\text{C}$, and entering the subcooled liquid zone of the p-h diagram (point 6; $p_6 = 16.1\text{ bar}$), while the refrigerating fluid portion contained in the cold branch 150f absorbs heat from the refrigerating fluid portion contained in the hot branch 150c, being heated from $T_9 = 7\text{ }^{\circ}\text{C}$ to a temperature $T_{10} = 12\text{ }^{\circ}\text{C}$ and entering the superheated vapour zone of the p-h diagram (point 10; $p_{10} = 7.48\text{ bar}$).

Downstream of the first heat exchanger 150 a second amount of refrigerating fluid is bled off, so that a portion s2 of the subcooled liquid leaving the hot branch 150c passes through the second isenthalpic throttling valve 144, where it is further cooled from the temperature $T_6 = 12\text{ }^{\circ}\text{C}$ to a temperature $T_{12} = -17\text{ }^{\circ}\text{C}$ (point 12 in p-h diagram; $p_{12} = 3.38\text{ bar}$) and then into the cold branch 152f of the second heat exchanger 152, while the remaining portion 1-s1-s2 of the refrigerating fluid leaving the heat exchanger 150 enters into the hot branch 152c of the second heat exchanger 152 at the temperature T_6 and pressure p_6 .

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Inside the second heat exchanger 152, the portion of refrigerating fluid contained in the hot branch 152c releases heat to the refrigerating fluid portion contained in the cold branch 152f, cooling from $T_6 = 12$ °C to a temperature $T_7 = -12$ °C and moving further to the left, in the diagram of Figure 4, into the subcooled liquid zone (point 7 in p-h diagram; $p_7 = 16.1$ bar), while the refrigerating fluid portion contained in the cold branch 152f absorbs heat from the refrigerating fluid portion contained in the hot branch 152c, being heated from $T_{12} = -17$ °C to a temperature $T_{13} = -12$ °C and entering the superheated vapour zone of the p-h diagram (point 13; $p_{13} = 3.38$ bar).

The first and second bleed-offs of refrigerating fluid s_1 , s_2 leaving each heat exchanger 150, 152 in the form of refrigerating fluid in the superheated vapour state are introduced, respectively, into the first and second turbine portion 162, 164 of the turbocompressor unit 160. Inside the first turbine portion 162, the refrigerating fluid undergoes expansion, passing from a pressure $p_{10} = 7.48$ bar ($T_{10} = 12$ °C) to a pressure $p_{11} = 2.03$ bar ($T_{11} = -25$ °C); similarly, inside the second turbine portion 164 the refrigerating fluid will undergo expansion passing from a pressure $p_{13} = 3.38$ bar ($T_{13} = -12$ °C) to a pressure $p_{14} = 2.3$ bar ($T_{14} = -25.6$ °C).

The portion of refrigerating fluid 1- s_1 - s_2 leaving the hot branch 152c of the second heat exchanger 152 (point 7 in p-h diagram) enters into the main throttling valve 170, cooling from $T_7 = -12$ °C to a temperature $T_8 = -40$ °C (point 8 in p-h diagram; $p_8 = 1.33$ bar) and then into the evaporator 180, where it

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passes from the liquid+vapour state to the superheated vapour state (point 1 in p-h diagram), absorbing a quantity of heat Q_{in} from the external environment. The refrigerating fluid in the superheated vapour state leaving the evaporator 180 enters into the compressor portion 166 of the turbocompressor unit 160.

The compressor 166, operated by the turbines 162, 164 hosting, inside them, the conversion, into mechanical energy, of the kinetic energy contained in the bled-off refrigerating fluid s1 and s2 in the superheated vapour state supplied by the first and second heat exchanger 150, 152, performs pre-compression of the refrigerating fluid supplied by the evaporator 180 (point 3 in p-h diagram; $T_3 = -22.1\text{ }^{\circ}\text{C}$, $p_3 = 2.03\text{ bar}$), before its entry into the main compressor 190.

This pre-compression stage offers considerable advantages. Firstly, since the mechanical energy is supplied by the bleed-offs s1, s2 which expand inside the turbines 162, 164, it is not required to use an external energy source. Secondly, the turbocompressor unit 160 compresses the refrigerating fluid, performing the work L_{TC} (Figure 4), when it is in the maximum specific volume condition, so that the main compressor 190 does not perform that part of the work which, in view of its constructional characteristics, penalizes its efficiency and in particular its processable mass flow, with a consequent reduction in the electric energy supplying the compressor itself. Again, the turbocompressor unit 160 has a fluid/dynamic connection with the main compressor 190 with the possibility of being able to adapt independently to the different load

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conditions without the aid of external control. Finally, it is important to mention the fact that cooling of the refrigerating fluid produced in the heat exchangers 150, 152 causes an increase in the performance of the evaporator 180, despite the fact that, following the bleed-offs s1, s2 there is, at the same time, a simultaneous reduction in the flow of refrigerating fluid into the evaporator 180.

The refrigerating fluid pre-compressed in turbocompressor unit 160 enters into the main compressor 190, where it is compressed to a pressure $p_4 = 16.1$ bar (point 4 in p-h diagram; $T_4 = 63.7$), and then conveyed to the inlet of the condenser 140.

It has been found that, with a device for circulating refrigerating fluid according to the present invention, namely comprising a pre-compression stage performed by a turbocompressor unit, it is possible to achieve a coefficient of performance (COP), defined as the ratio between the heat Q drawn from the lower temperature source, which constitutes the "amount of cold" produced and the work L expended in order to cause operation of the device for circulating a refrigerating fluid, which is greater than that of a conventional device of the type illustrated in Figures 1 and 2.

In particular, assuming the pressures of the bleed-offs s1 and s2 to be, respectively, of $p_9 = 7.48$ bar and $p_{12} = 3.38$ bar, a minimum temperature gradient $\Delta T_{\min} = 5$ °C in the heat exchangers 150, 152, an efficiency $\eta_T = 0.85$ of the first and second turbine portion 162, 164, an efficiency $\eta_c = 0.80$ of the compressor portion 166 and an efficiency $\eta_{CP} = 0.75$ of

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the main compressor 190, the pressure values ($\hat{p}$), temperature values (T) and enthalpy values (h) are obtained for the physical states 1-14 of the p-h diagram according to Figure 4, shown in the following

5 Table 1:

Table 1

Physical State	p [bar]	T [°C]	h [Kj/Kg]
1	1.33	-35	347.6
2	2.03	-20	358.1
3	2.03	-22.1	356.6
4	16.1	63.7	415.0
5	16.1	35	254.8
6	16.1	12	217.5
7	16.1	-12	183.4
8	1.33	-40	183.4
9	7.48	7	254.8
10	7.48	12	376.7
11	2.03	-25	354.3
12	3.38	-17	217.5
13	3.38	-12	362.5
14	2.03	-25.6	353.8

The coefficient of performance COP is defined, in
 10 general, as the ratio between the heat Q subtracted
 from the lower temperature source, which constitutes
 the "amount of cold" produced, and the work L expended
 to cause operation of the refrigerating fluid
 circulation device. In particular, the COP is defined
 15 by the ratio between the heat Q_{in} subtracted from the
 external environment by the evaporator 180 and the work

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L_{CP} performed by the main compressor 190, namely:

$$\text{and} \quad Q_{in} = (1 - s1 - s2) \times (h1 - h7)$$

$$L_{CP} = h4 - h2$$

5 From which, based on the values shown in Table 1, the following is obtained:

$$COP = \frac{Q_{in}}{L_{CP}} = 1,74$$

10 Table 2 below summarises the typical pressure, temperature and enthalpy values of a refrigerating fluid circulating inside a conventional refrigeration device of the type illustrated in Figures 1 and 2.

Table 2

Physical State	p [bar]	T [°C]	h [Kj/Kg]
1	1.33	-35	347.6
2	16.1	65.3	416.9
3	16.1	35	254.8
4	1.33	-40	254.8

15

This gives:

$$\text{and} \quad q_{in} = (h1 - h4)$$

$$L_{CP} = h2 - h1$$

20 from which, based on the values shown in Table 2, the following is obtained:

$$COP_{ST} = \frac{q_{in}}{L_{CP}} = 1,34$$

25 The percentage benefit Δ of the novel refrigerating device compared to a refrigerating device of the conventional type is:

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$$\Delta = \frac{COP - COP_{st}}{COP_{st}} \cong 30\%$$

From the description provided hitherto it is possible to state that a refrigerating device according to the present invention, owing to the presence of the turbocompressor unit 160 and the consequent pre-compression of the refrigerating fluid circulating inside the device upstream of the main compressor 190, allows an increase in performance equal to about 30% to be obtained, all of which without the need for power supplied externally, but advantageously using the mechanical energy provided by one or more turbine portions 162, 164 of the turbocompressor unit 160, obtained by causing the expansion of one or more amounts s1, s2 of refrigerating fluid bled-off downstream of the condenser 140.

Although the invention has been described with reference to a preferred example thereof, persons skilled in the art will understand that it is possible to apply numerous modifications and variations thereto, all of which fall within the scope of protection defined by the accompanying claims. For example, instead of two heat exchangers and turbocompressor unit with two turbines, it is possible to use a single heat exchanger and a turbocompressor unit with a single turbine. In this specific case, the single heat exchanger will have the hot branch connected between the condenser and the main throttling valve and the cold branch in fluid communication with the inlet of the single turbine portion of the turbocompressor. Moreover, instead of a turbocompressor unit having

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multiple turbine portions, it is possible to envisage a plurality of turbocompressors each with a single turbine portion.

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CLAIMS

1. Refrigerating device comprising a main compressor (190), a condenser (140) downstream of and in fluid communication with said main compressor (190), main expansion means (170) downstream of said condenser (140) and an evaporator (180) downstream of and in fluid communication with said main expansion means (170),
- characterized in that it comprises a turbocompressor unit (160) in fluid communication between said evaporator (180) and said main compressor (190) and at least one heat exchanger (150, 152) having a hot branch (150c) connected upstream, via an inlet line (145), to said condenser (140) and downstream, via an outlet line (149), to said main expansion means (170) and a cold branch (150f) connected, upstream, to an expansion means (142, 144) mounted on a branch (146) of said line (145) and, downstream, to a turbine portion (162) of said turbocompressor unit (160).
2. Device according to Claim 1, characterized in that said at least one heat exchanger (150, 152) is a tube-bundle heat exchanger.
3. Device according to Claim 1, characterized in that said at least one heat exchanger (150, 152) is a plate-type heat exchanger.
4. Device according to Claim 1, characterized in that said expansion means (142, 144) is an isoenthalpic throttling valve.
5. Device according to any one of Claims 1 to 4, characterized in that it comprises a first and a second heat exchanger (150, 152) arranged in series between said heat exchanger (140) and said main expansion means

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(170) and in that said turbocompressor unit (160) comprises a first and a second turbine portion (162, 164), said second heat exchanger (152) having a hot branch (152c) in fluid communication, via a connection
5 line (147), with the hot branch (150c) of said first heat exchanger and a cold branch (152f) connected, upstream, to an expansion means (144) mounted on a branch (148) of said line (147) and, downstream, to said second turbine portion (166) of said
10 turbocompressor unit (160).

6. Method for circulating a refrigerating fluid comprising the stages of:

- compressing the refrigerating fluid in a main compressor (190);

15 - condensing the fluid in a condenser (140) downstream of and in fluid communication with said main compressor (190);

- expanding the fluid in main expansion means (170) downstream of said condenser (140);

20 - evaporating the fluid in an evaporator (180) downstream of and in fluid communication with said main expansion means (180);

characterized in that it comprises:

- between said condensation stage and said expansion
25 stage at least one stage involving heat exchange, inside at least one heat exchanger (150, 152), between the compressed refrigerating fluid circulating inside a hot branch (150c, 152c) of the heat exchanger (150, 152) and an associated amount (s1, s2) of the
30 compressed refrigerating fluid bled-off upstream of the heat exchanger (150, 152), cooled inside an expansion means (142, 144) and flowing inside a cold branch

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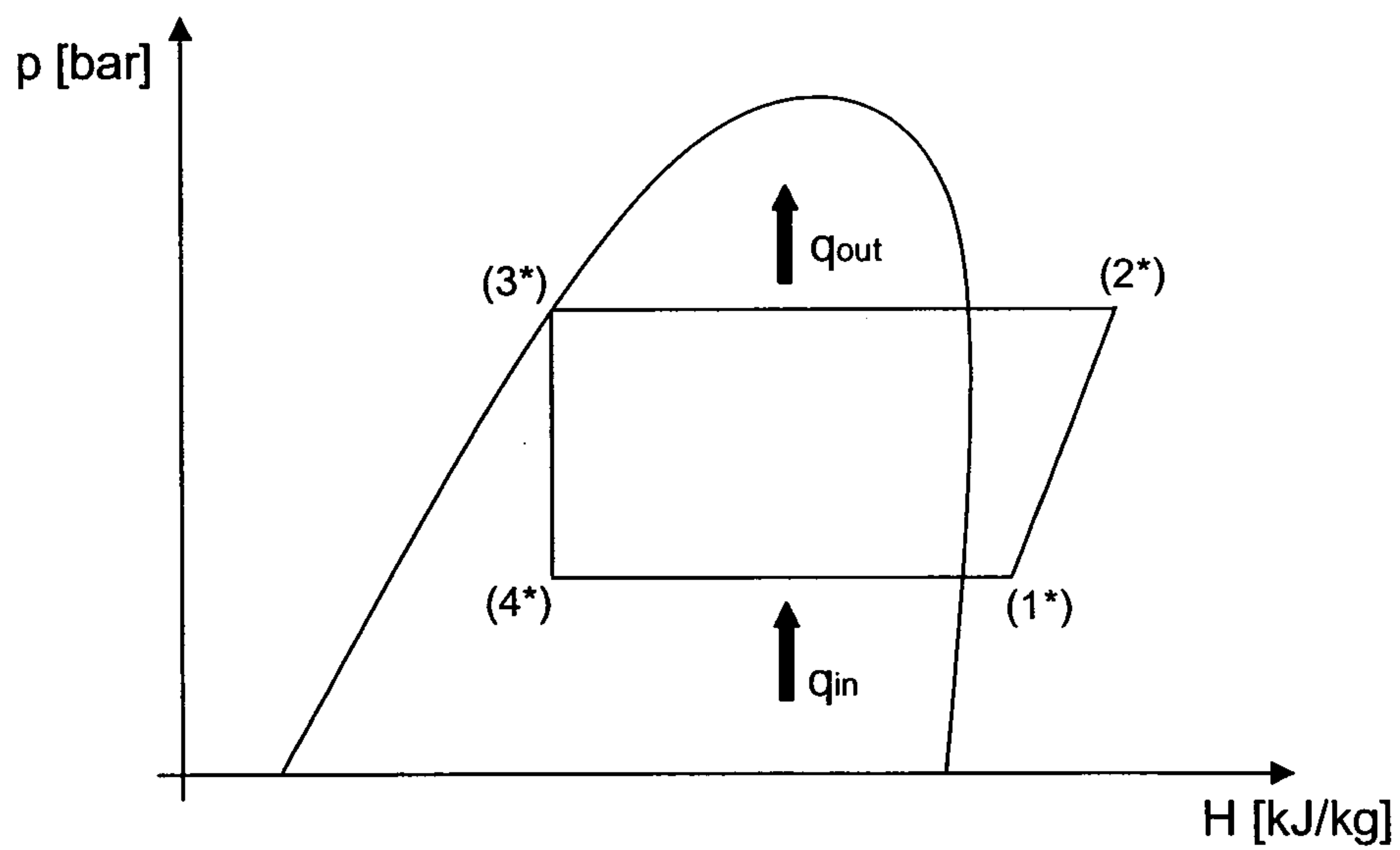
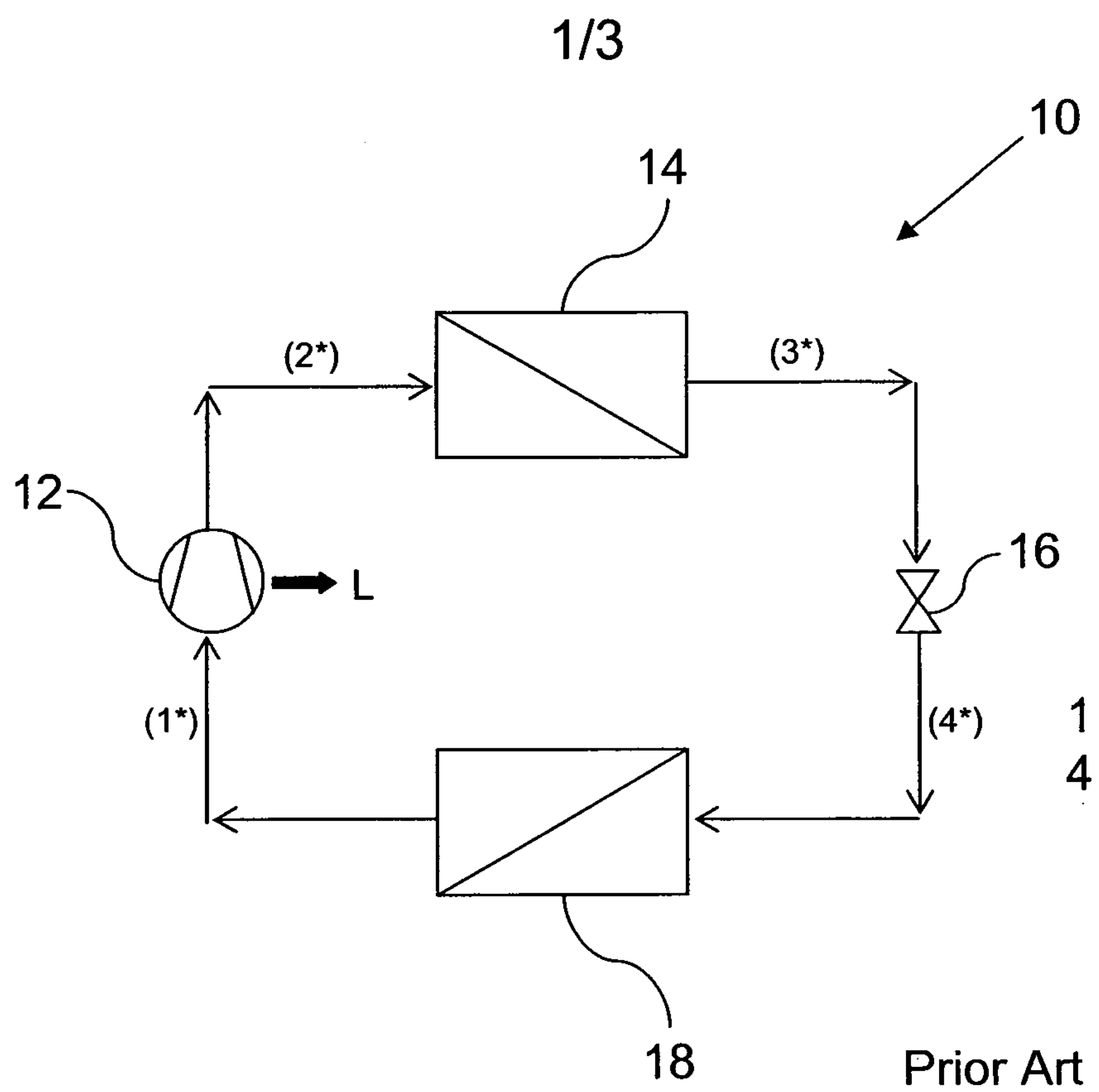
(150f, 152f) of the heat exchanger (150, 152); and

- between said main expansion stage and said main compression stage, a stage involving pre-compression of the refrigerating fluid inside a turbocompressor unit (160), said pre-compression stage comprising at least one stage involving expansion, inside at least one turbine portion (162, 166) of the turbocompressor unit, of the bled-off amount (s1, s2) of refrigerating fluid, leaving the cold branch (150f, 152f) of the heat exchanger (150, 152).

7. Method according to Claim 6, characterized in that it comprises, downstream of said at least one heat exchange stage between said condensation stage and said expansion stage:

- a second stage involving heat exchange in a second heat exchanger (150) arranged in series with the at least one exchanger (150) between the refrigerating fluid leaving the hot branch (150c) of the at least one heat exchanger (150) and circulating inside the hot branch (152c) of the second exchanger (152) and an associated amount (s2) of the refrigerating fluid bled-off upstream of the heat exchanger (152), cooled inside an expansion means (144) and circulating in the cold branch;

and in that said pre-compression stage between said main expansion stage and main compression stage is powered by expansion, in a first and second turbine portion (162, 164) of said turbocompressor unit (160), of the bleed-offs from each heat exchanger (150, 152).



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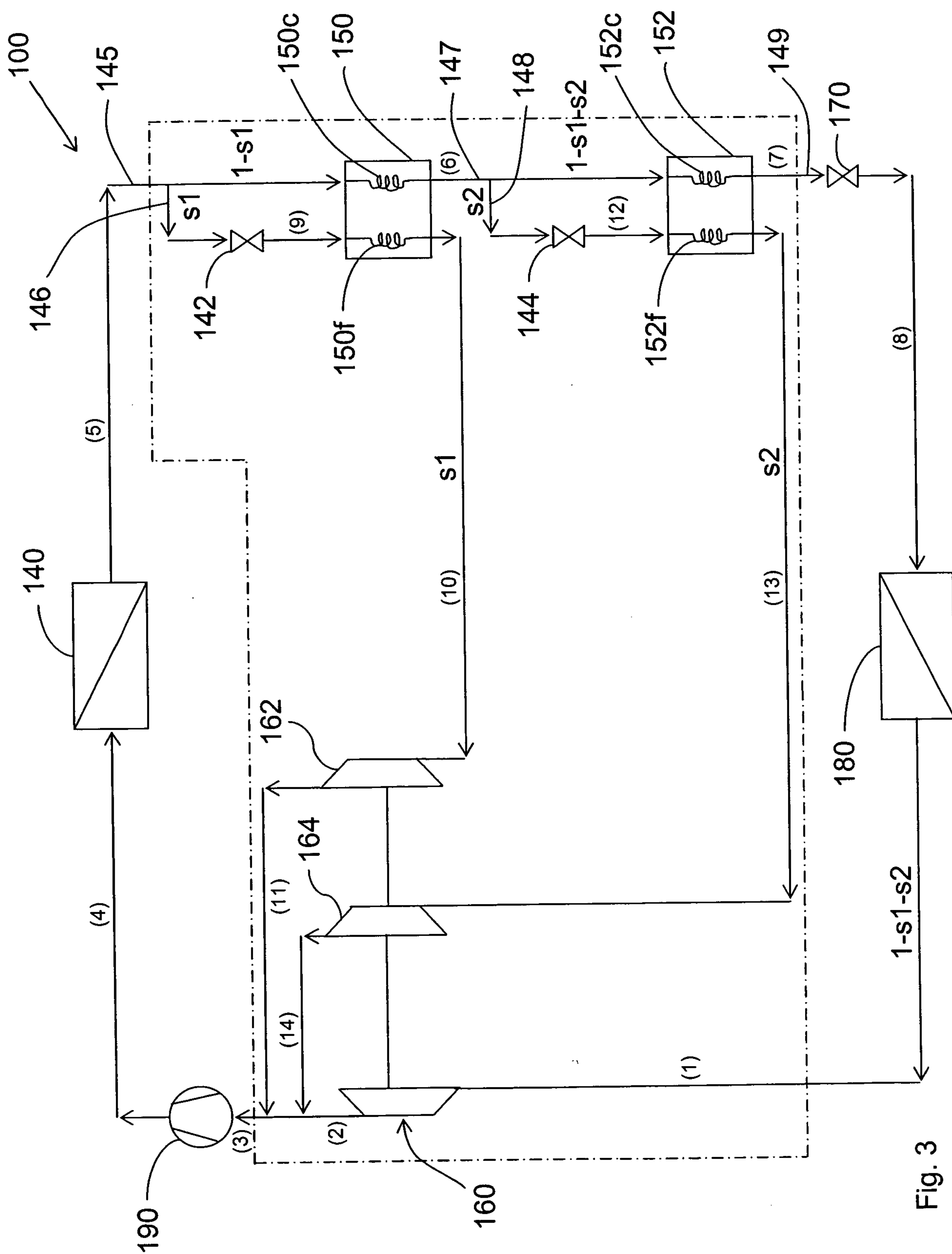


Fig. 3

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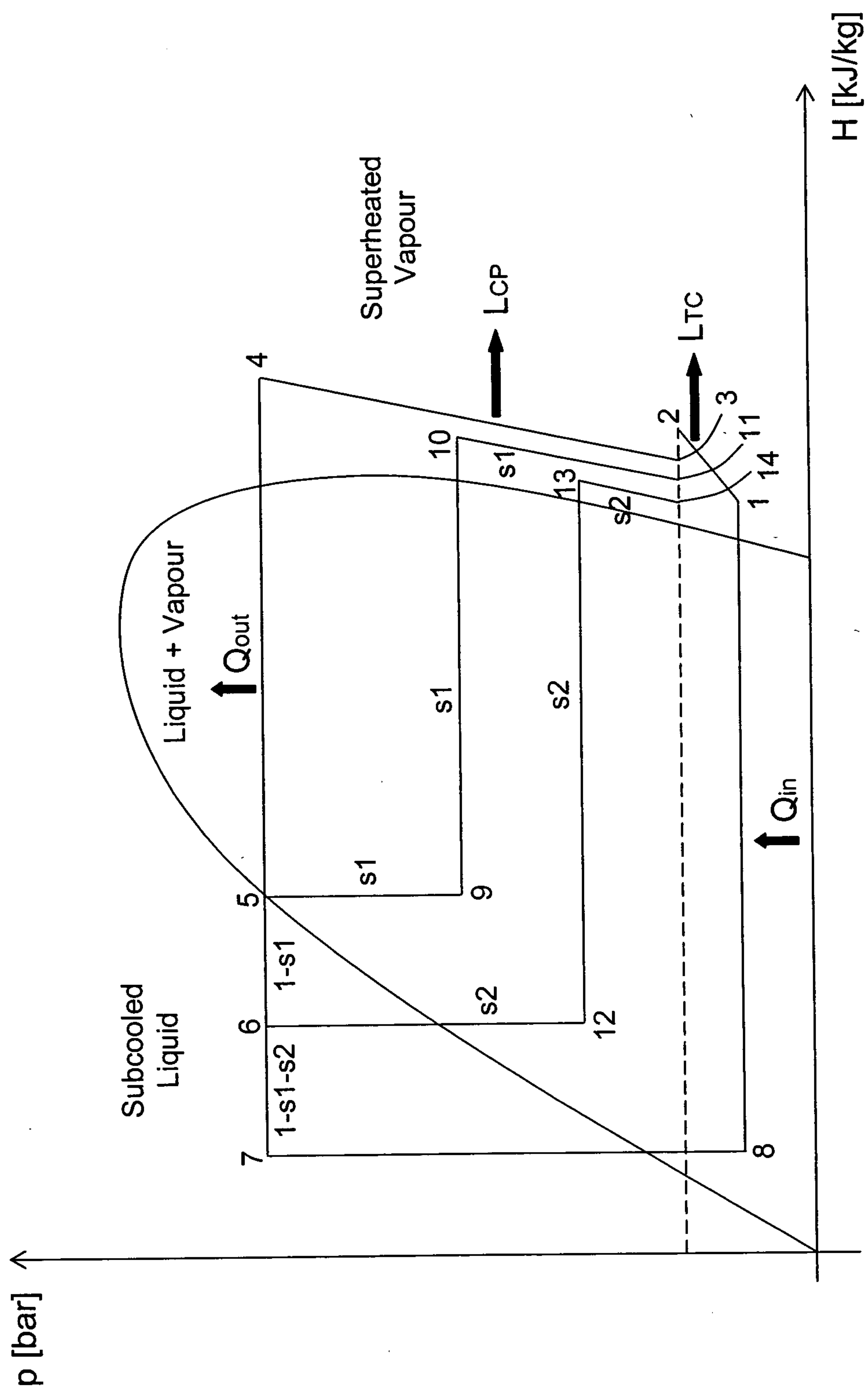


Fig. 4

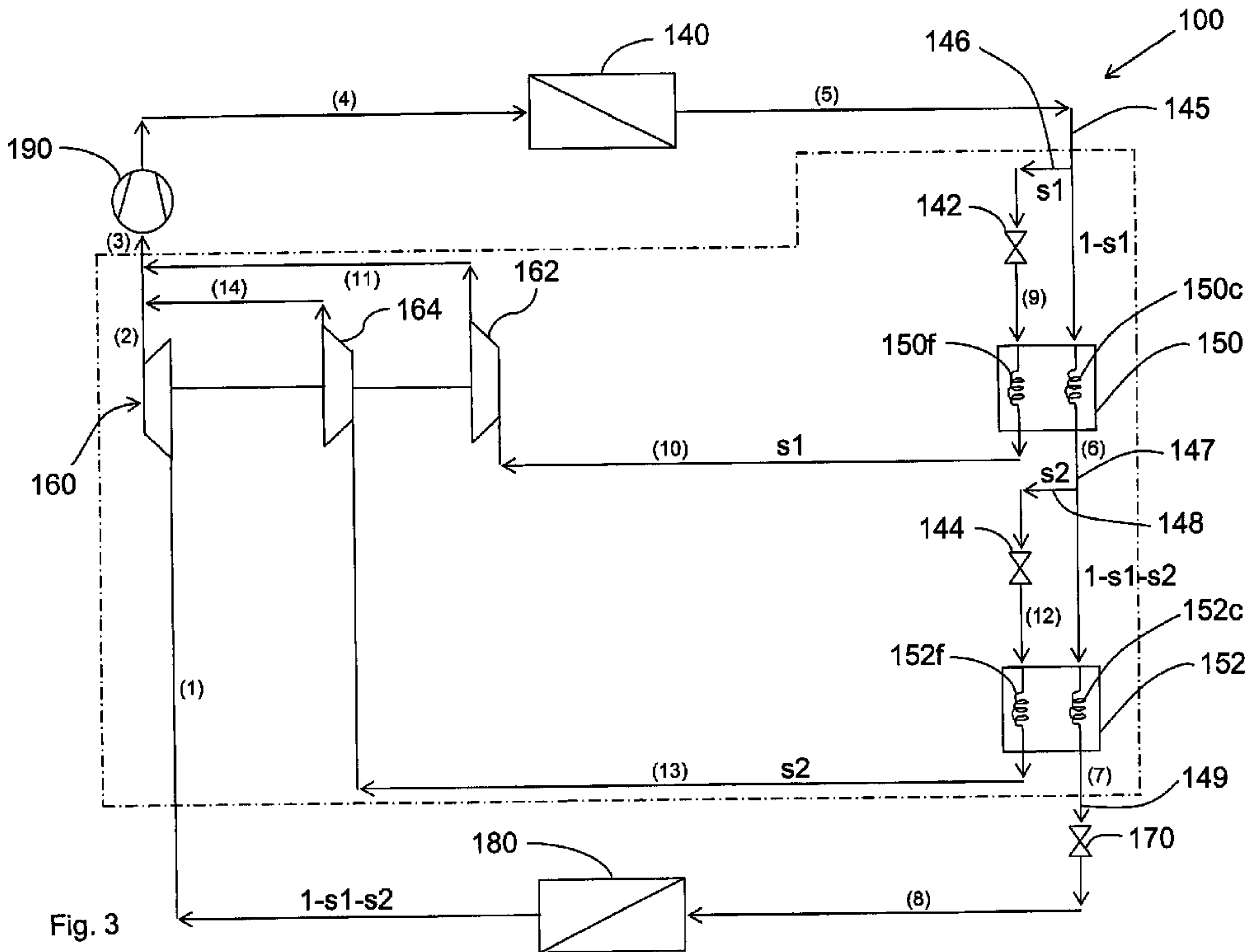


Fig. 3