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(54) **CONDENSATE-FREE OUTDOOR AIR COOLING UNIT**

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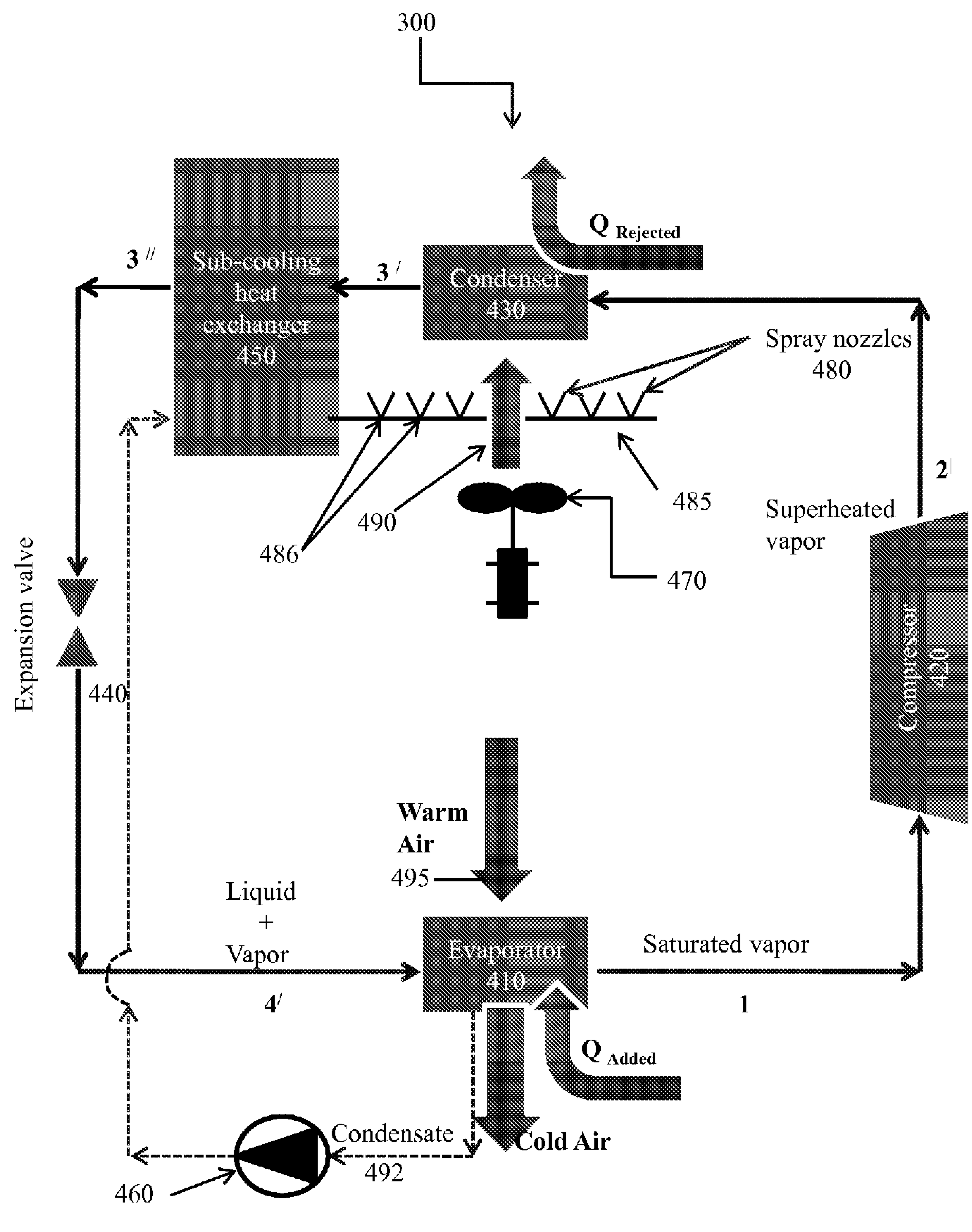
USPC **62/82**; 62/277

(57) **ABSTRACT**

A highly efficient condensate-free cooling unit functionally based on vapor-compression refrigeration cycle has been described. The condensate collected from the evaporator of the cooling unit is routed through a sub-cooling heat exchanger where it exchanges heat with the primary heat exchange medium emerging through the condenser of the cooling unit, thus, sub-cooling the primary heat exchange medium to a lower temperature before it enters the expansion valve. Emerging from the sub-cooling heat exchanger, the condensate flows through a condensate outlet pipe into multiple spray nozzles disposed over the condensate outlet pipe. The spray nozzles sprinkle the condensate over the hot air blown into the condenser to reduce its temperature. The cooling unit has a substantially higher coefficient of performance compared to the conventional cooling units utilizing vapor-compression refrigeration cycle, and eliminates the problems of condensate removal persistent in the art.



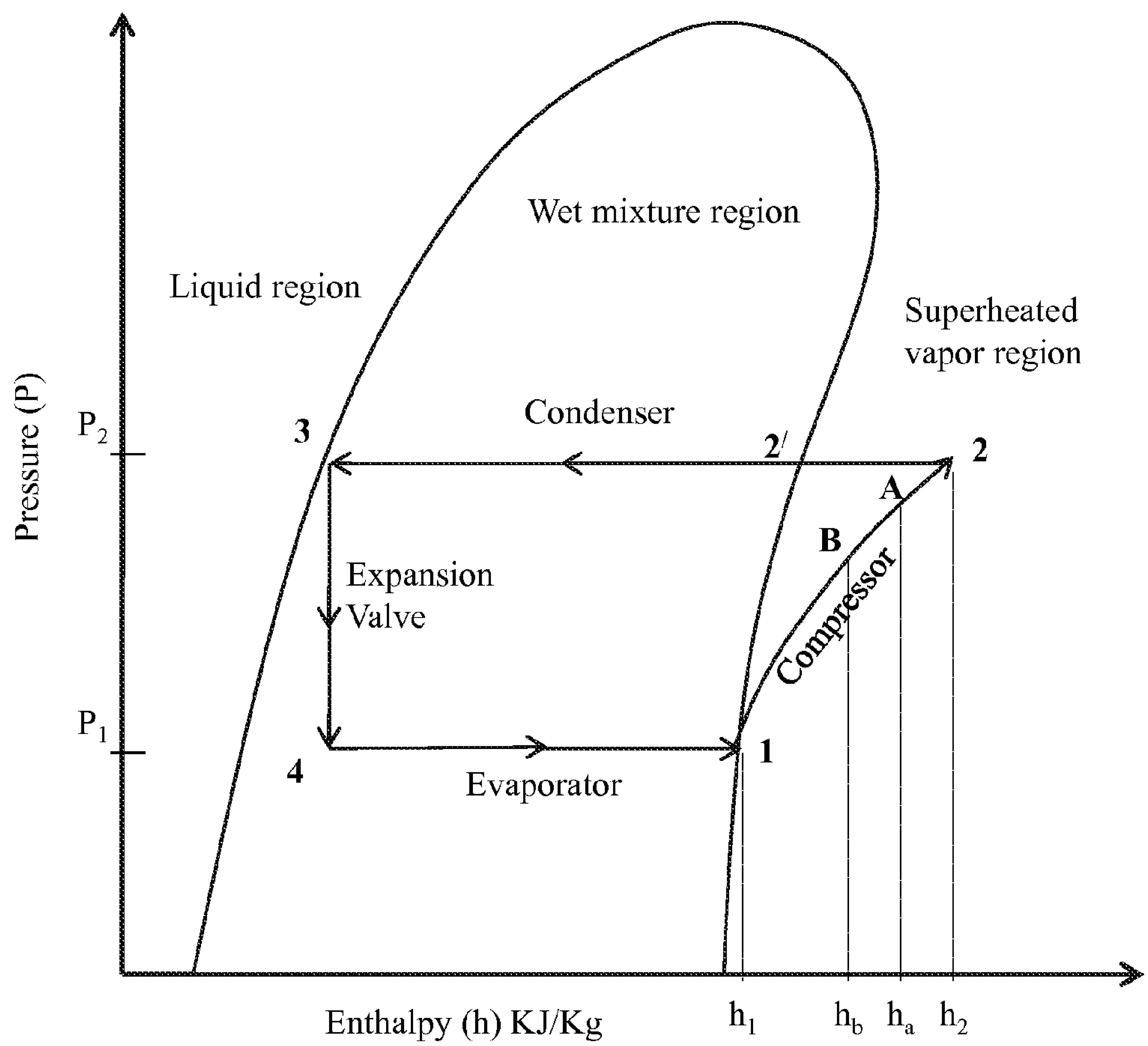

Wet mixture region
Liquid region
Superheated
vapor region
Pressure (P)
P_2
3
Condenser
2'
2
A
B
Expansion
Valve
Compressor
P_1
1
4
Evaporator
Enthalpy (h) KJ/Kg
h_1
h_b
h_a
h_2

Fig. 1

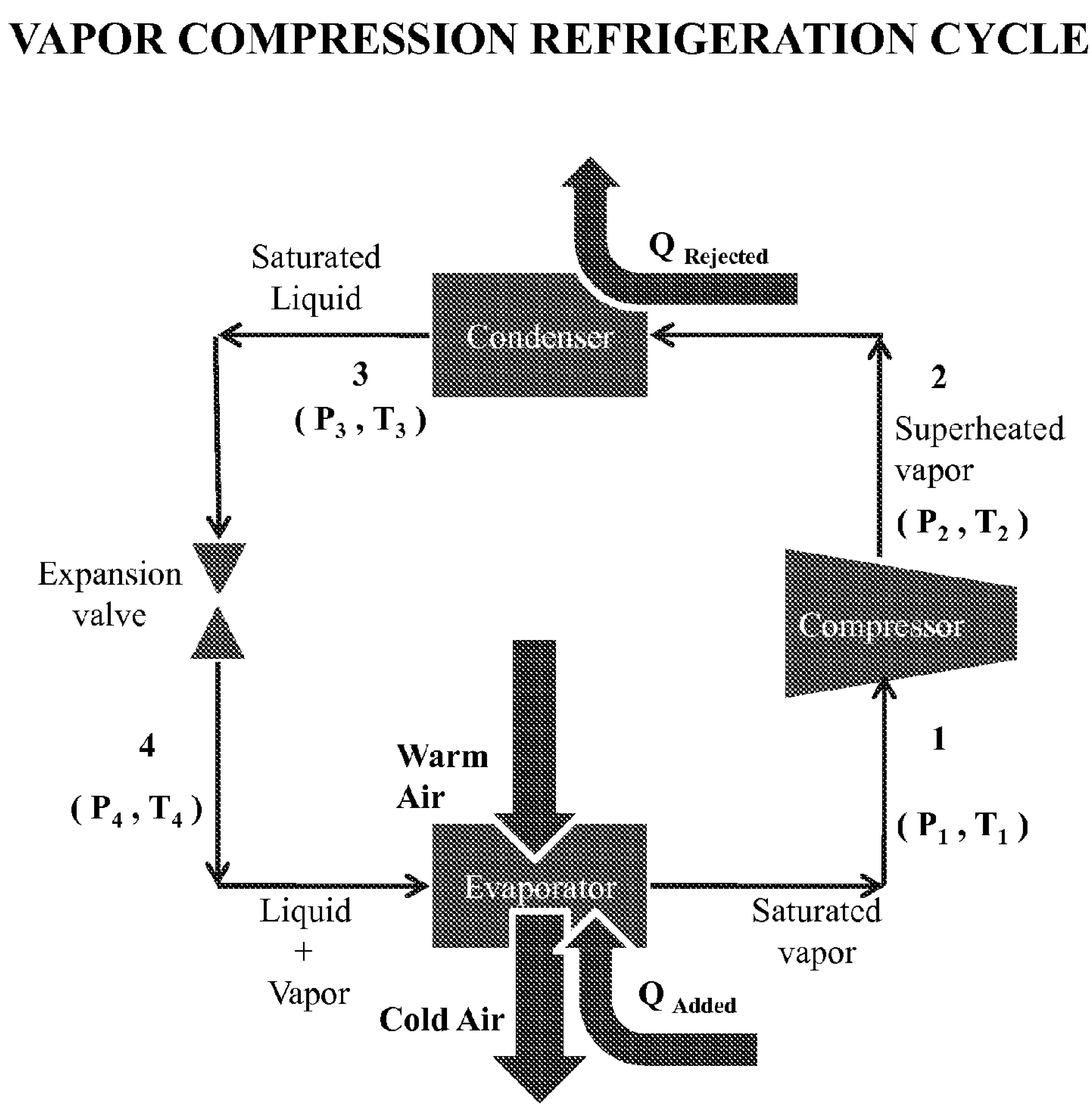


Fig. 2 (Prior Art)

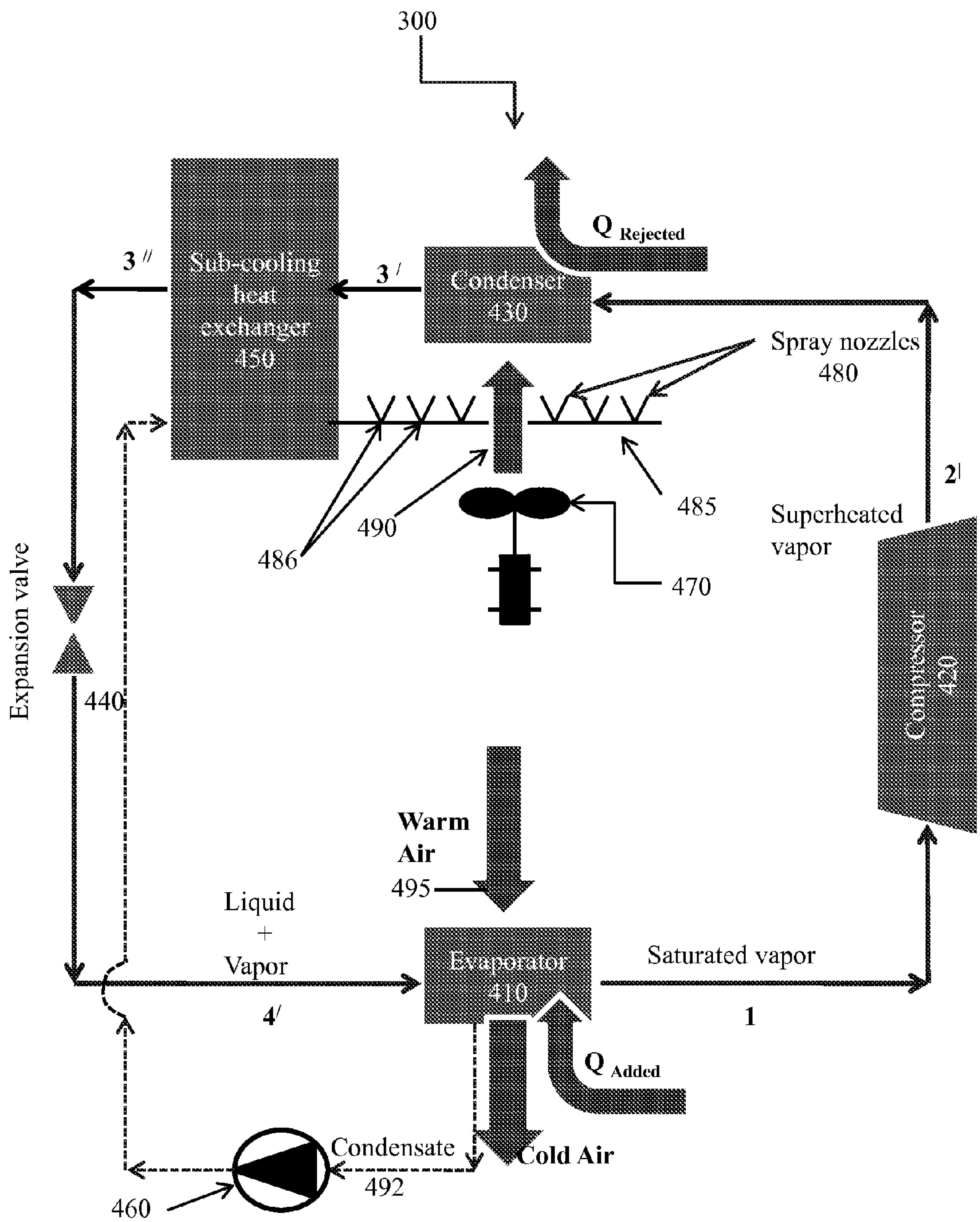

300
Q Rejected
3 //
Sub-cooling heat exchanger 450
3 /
Condenser 430
Spray nozzles 480
2
485
Superheated vapor
486
490
470
Expansion valve
440
Compressor 420
Warm Air
495
Liquid + Vapor
Evaporator 410
Saturated vapor
4/
1
Q Added
Condensate
Cold Air
492
460

Fig. 3

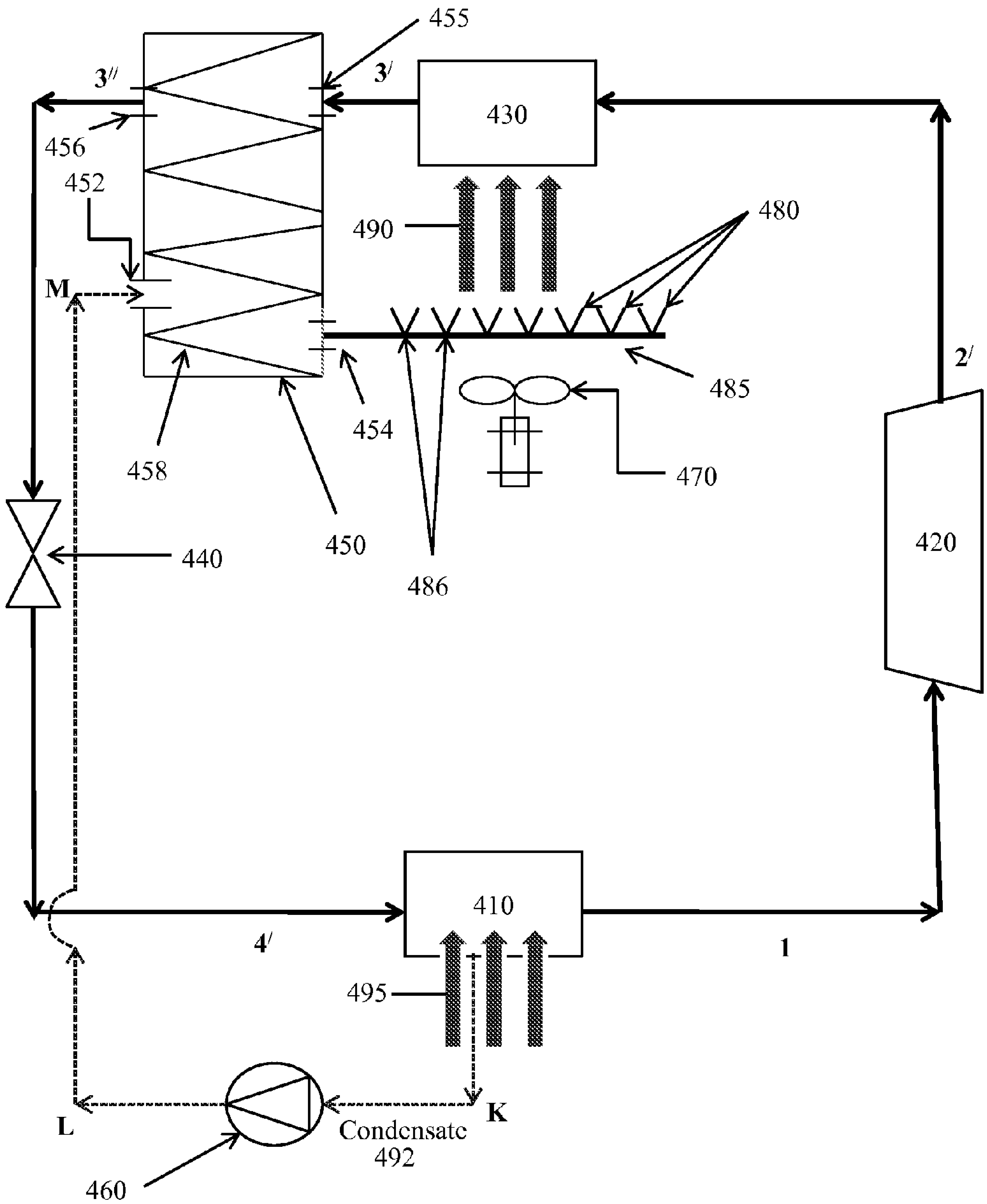

455
3''
3'
430
456
452
490
480
M
485
2'
454
458
470
450
440
486
420
410
4'
1
495
L
K
Condensate
492
460

Fig. 4

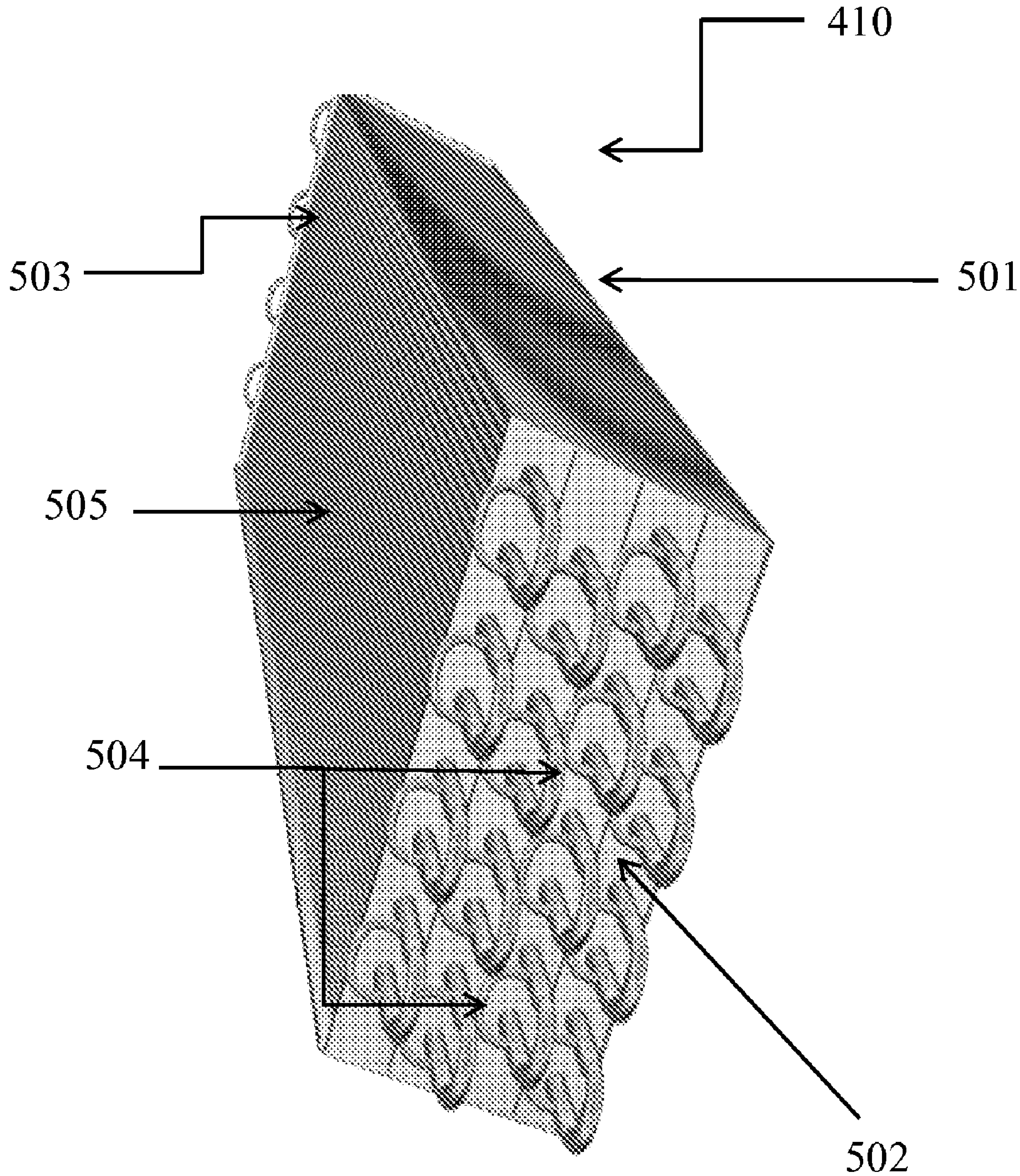


Fig. 5

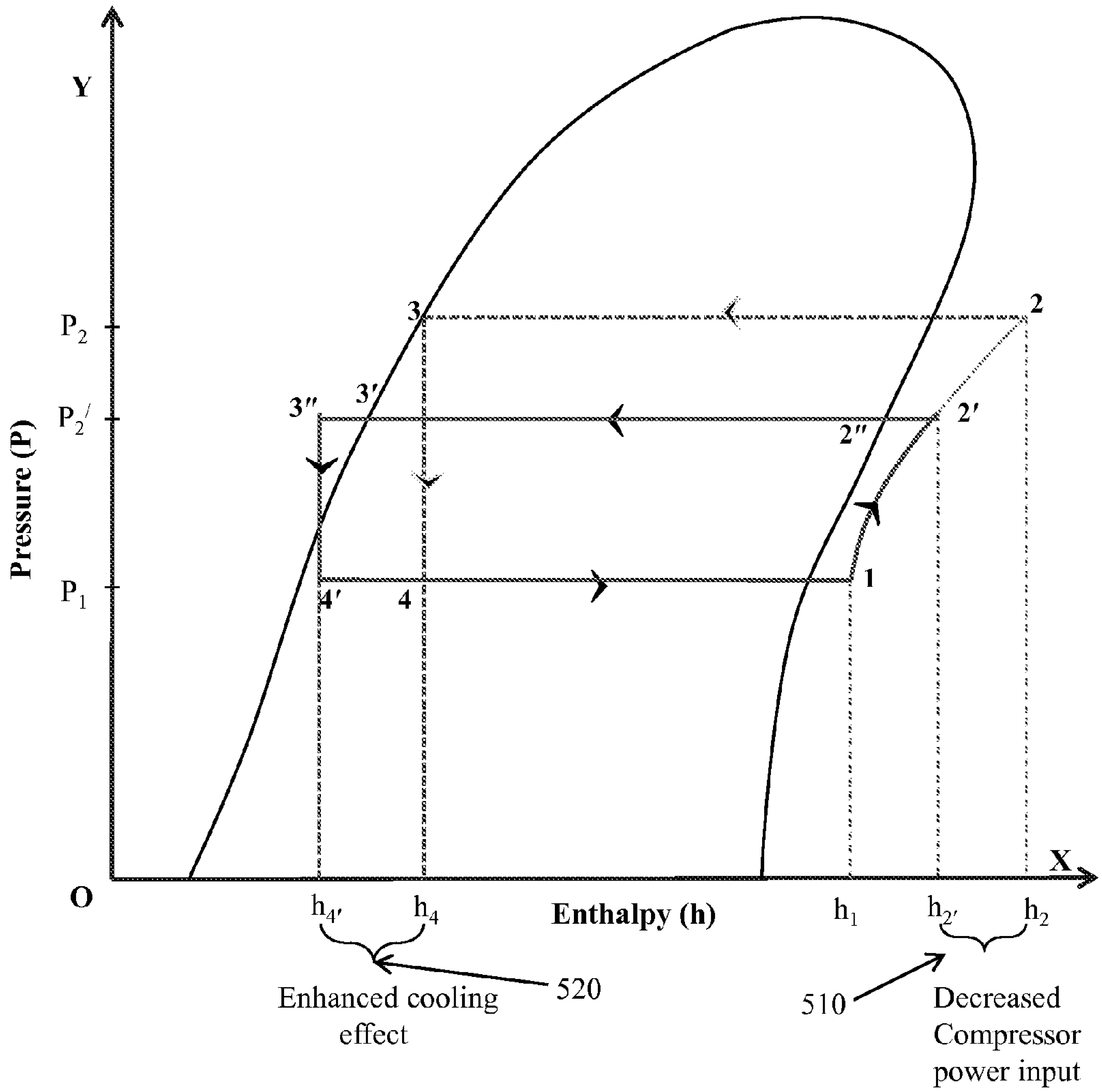

Y
Pressure (P)
P2
P2/
P1
O
X
Enthalpy (h)
3
3′
3″
2
2′
2″
1
4′
4
h4′
h4
h1
h2′
h2
520
Enhanced cooling effect
510
Decreased Compressor power input

Fig. 6

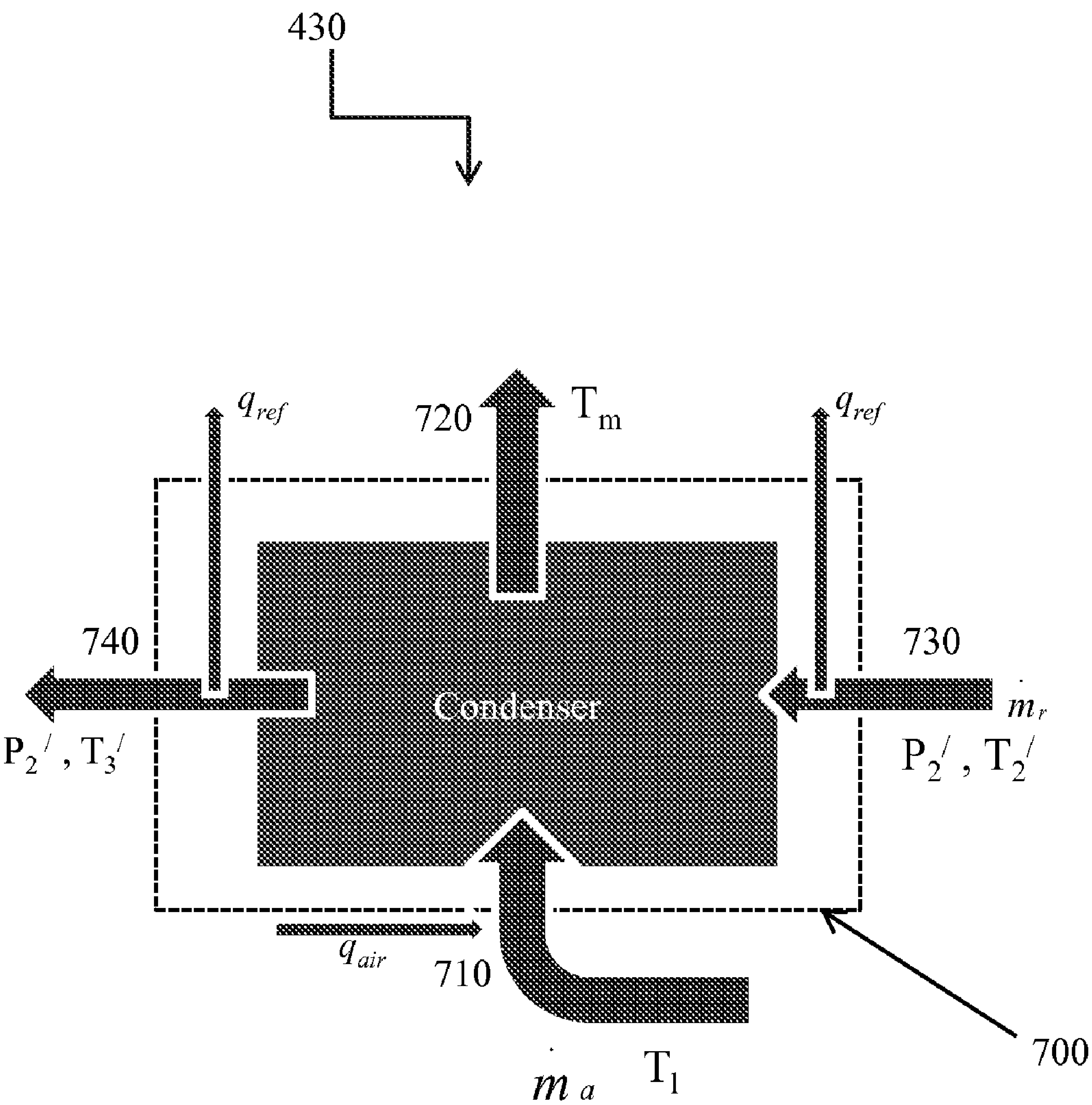

430
q_{ref}
720
T_m
q_{ref}
740
730
Condenser
$\dot{m}_r$
$P_2^{/}$, $T_3^{/}$
$P_2^{/}$, $T_2^{/}$
q_{air}
710
700
$\dot{m}_a$
T_1

Fig. 7

CONDENSATE-FREE OUTDOOR AIR COOLING UNIT

FIELD OF THE INVENTION

[0001] The present disclosure deals generally with heat exchange devices, and more specifically with cooling devices utilizing the vapor compression refrigeration cycle.

BACKGROUND OF THE INVENTION

[0002] Vapor compression refrigeration cycles are widely used in many cooling systems for achieving refrigerating effects, including refrigerators, air-conditioning systems, industrial and commercial refrigeration systems and the like. Typical applications of such cooling units include warehouses, offices, private and public residential buildings, hospitals, hotels, restaurants and cafeteria. Succinctly, the process of refrigeration refers to extracting heat from a space and rejecting it elsewhere, thus lowering the temperature of the space. Vapor compression refrigeration cycles use a refrigerant for this process. The cycle includes four elementary components: a compressor, a condenser, an expansion valve, and an evaporator. The circulating refrigerant enters the compressor in the form of saturated vapor and undergoes isentropic compression, thus increasing its pressure and temperature, and converting into a superheated vapor. The refrigerant then enters the condenser, where it comes into thermal contact with cold water or air and rejects heat to it. In the process, the refrigerant absorbs heat (sensible heat) and is converted from superheated vapor to saturated vapor. After further absorbing latent heat, the saturated vapor eventually is converted to saturated liquid.

[0003] At the condenser exit, the refrigerant is thermodynamically a saturated liquid. That liquid passes through an expansion valve, where it expands, undergoing a reduction in pressure and consequently a partial flash evaporation. This process converts the refrigerant into a mixed liquid and vapor and reduces its temperature to a level below that of the space to be refrigerated. The mixture then enters the evaporator where it extracts latent heat from the space and thus completely vaporizes to a saturated vapor. The saturated vapor re-enters the compressor to complete the refrigeration cycle. This description pertains to an ideal vapor-compression refrigeration cycle, neglecting practical real-world effects such as the frictional pressure drop in the system and the slight thermodynamic irreversibility.

[0004] In the evaporator, the air to be conditioned passes over heat exchange tubes, exchanging heat with the refrigerant and simultaneously lowering the air temperature and vaporizing the refrigerant. The ambient air blown over the evaporator's heat exchange tubes generally carries a certain amount of water vapor, and in the course of exchanging heat with the refrigerant, this vapor partially condenses, forming a water condensate which starts dripping off from the evaporator.

[0005] Appropriate disposal of cooling unit condensate poses a challenging problem. Providing a separate piping system or condensate collection containers can be burdensome and expensive. Dehumidifiers are employed in certain applications, to reduce the moisture content of the air entering the evaporator. Sometimes, excessive extraction of heat from the air even causes frost or ice to build-up in the evaporator and the surface of its heat exchange tubes, causing the dehumidifier to stop. Thermostats have been used in the art to detect the frost accumulation conditions, by identifying moments when the air temperature drops below a certain value. The compressor then shuts down until the system warms up, but those actions shut down the cooling system itself. Efforts have been also made in the art for routing the condensate out of the evaporator casing through pumps.

[0006] Another challenge is increasing the operating efficiency of the cooling units. A measure of the operating efficiency of such cycles is the coefficient of performance (COP), which is the ratio of the obtainable useful refrigeration effect to the power required to drive the cycle (including the compressor driving power). Either an increase in the useful refrigeration effect or a decrease in the power input to the compressor may increase the operating efficiency. Substantially successful attempts in that direction include disposing an intercooler between the condenser and the expansion valve, routing the high pressure/high temperature refrigerant through the intercooler, where it exchanges heat with the low temperature refrigerant entering the evaporator. This decreases the enthalpy of the high pressure refrigerant and thus increases the coefficient of performance. To increase efficiency, the added compressor power input must be less than the increase in the useful refrigeration effect. In some applications, the high pressure value of the refrigerant in the cycle is varied and a corresponding change in the value of the COP is observed to derive a correlation. This correlation is then used to identify the high pressure value at which the COP maximizes. Many times, the analysis and identification of this correlation may be time consuming and inaccurate.

[0007] At present, there exists a need for an efficient cooling unit that could have a considerably higher coefficient of performance compared to the conventionally used systems, and would simultaneously address the problem of condensate removal from cooling units, in an effective manner.

SUMMARY

[0008] The present invention is directed to a highly effective and advantageous cooling unit functionally based on vapor compression refrigeration cycle. The cooling unit has an operable efficiency (coefficient of performance) substantially greater than the coefficient of performance of the conventionally used cooling units, including refrigerators and air-conditioning devices. It further effectively eliminates problem of condensate removal persistent in the conventional systems.

[0009] In one aspect, a cooling unit for cooling an ambient medium is provided that includes a compressor, a condenser, an expansion valve, an evaporator, a sub-cooling heat exchanger and a condensate pump. The cooling unit functionally utilizes vapor compression refrigeration cycle for achieving refrigerating effect. The evaporator includes a number of heat exchange tubes through which a primary heat exchange medium flows. The heat exchange tubes remain in continuous thermal communication with the ambient medium. The condensate pump is connected to the evaporator to route the condensate collected within the evaporator through the sub-cooling heat exchanger. The sub-cooling heat exchanger has a condensate inlet meant to receive the routed condensate and a condensate outlet for delivering the condensate to a number of spray nozzles connected to it through a condensate outlet pipe. The condensate outlet pipe has a set of perforations provided on its surface through which the sub-cooling heat exchanger remains in continuous fluid communication with the spray nozzles. Effectively, the condensate is routed from the evaporator to the sub-cooling heat exchanger and further to the spray nozzles. The spray nozzles are directly coupled with hot air blowing into the condenser and are meant to sprinkle the condensate thereon to reduce its temperature. The sub-cooling heat exchanger further has a primary heat exchange medium inlet and a primary heat exchange medium outlet for a continuous influx and efflux of the primary heat exchange medium through it. During operations, the condensate and the primary heat exchange medium remain in continuous thermal communication within the sub-cooling heat exchanger and exchange heat with each other.

[0010] The cooling unit has a considerably high coefficient of performance compared to the conventionally used cooling systems. Further, the cooling unit is effectively dry and perfectly addresses the problems of condensate removal persistent in the art.

[0011] Additional features and advantages of the invention will be made apparent from the following detailed description of illustrative embodiments that proceed with reference to the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

[0012] The summary above, as well as the following detailed description of preferred embodiments, is better understood when read in conjunction with the appended drawings. For the purpose of illustrating the invention, exemplary constructions of the invention are shown in the drawings. The invention is not limited to the specific methods and instrumentalities disclosed however. Moreover, those in the art will understand that the drawings are not to scale. Where possible, like elements are indicated by identical numbers.

[0013] FIG. **1** is the pressure-enthalpy chart for a refrigerant undergoing a conventional vapor compression refrigeration cycle.

[0014] FIG. **2** is a schematic diagram showing the basic components of a conventional vapor compression refrigeration system.

[0015] FIG. **3** illustrates the different components of an exemplary cooling unit in accordance with the present disclosure

[0016] FIG. **4** is another schematic diagram representing the different components of the cooling unit of FIG. **3**

[0017] FIG. **5** shows the evaporator of the cooling unit of FIG. **4** in greater details.

[0018] FIG. **6** shows the pressure-enthalpy chart for the primary heat exchange medium circulating within the refrigeration cycle utilized by the exemplary cooling unit of FIG. **4** in accordance with the present disclosure.

[0019] FIG. **7** illustrates the heat exchange process within the condenser of the cooling unit of FIG. **4**.

DETAILED DESCRIPTION OF ILLUSTRATIVE EMBODIMENTS

[0020] The description below illustrates embodiments of the claimed invention. This description discloses aspects of the invention but does not define or limit the invention, such definition and limitation being contained solely in the claims appended hereto. Those of skill in the art will understand that the invention can be implemented in a number of ways different from those set out here, in conjunction with other present or future technologies.

[0021] As used herein, the following terms carry the had indicated meanings: "Ambient medium" is the medium targeted for conditioning by the disclosed cooling unit. In a building air-conditioning system, for example, the ambient medium would be the air inside the building. "Primary heat exchange medium" designates the refrigerant. In most circumstances, the primary heat exchange medium would be a refrigerant capable of exchanging heat with the ambient medium targeted for conditioning.

[0022] FIG. **1** shows the pressure-enthalpy diagram for a refrigerant undergoing a typical single-stage vapor compression refrigeration cycle. The description should be read in conjunction with FIG. **2**, which depicts the components of a conventional vapor-compression refrigeration system and the states corresponding to those marked in the pressure-enthalpy diagram for the refrigerant, as shown in FIG. **1**. The refrigerant, in the form of saturated vapor at point **1** (pressure P_1 and Temperature T_1), enters the compressor where it is compressed to a high pressure P_2 and high temperature T_2, which converts the refrigerant from the thermodynamic state of saturated vapor to superheated vapor at point **2**. At this point, the refrigerant exits the compressor. The increase in refrigerant pressure causes an increase in temperature, thus bringing it to the state **2** where it can be condensed by typically available air or water.

[0023] At state point **2**, the refrigerant enters the condenser and flows through the condenser coils, where it comes in thermal contact with the air or water flowing across the coils. The coils allow the refrigerant to reject sensible heat to the air or water, converting it from superheated vapor to saturated vapor in the course of traversing from point **2** to point **2**. The rejected heat is called sensible heat because its loss leads to a change in refrigerant temperature during the conversion from superheated to saturated vapor. The enthalpy of the refrigerant decreases in this process as heat is extracted, or equivalently, heat is rejected by it to the air or water in contact. The refrigerant absorbs further heat, and loses further enthalpy, until it is converted to saturated liquid at point **3**.

[0024] At point **3**, the refrigerant exits the condenser and enters the expansion valve. Here, saturated liquid refrigerant expands, with a sharp loss of pressure, to point **4** corresponding to the exit from the expansion valve. This sudden decrease in pressure from a value P_3 to P_4 causes a corresponding drop in the temperature of the liquid refrigerant from the corresponding value T_3 to T_4 and a portion of the liquid refrigerant flashes off into vapor, leaving the refrigerant in mixed liquid and vapor form at the exit **4** of the expansion valve. The enthalpy of the refrigerant however remains constant from point **3** to **4**, as shown in FIG. **1**, as no external heat is added or subtracted.

[0025] At point **4**, the refrigerant enters the evaporator where it comes into contact with the air to be cooled. This air flows across the evaporator coils and exchanges heat with the refrigerant flowing in the coils. By extracting the latent heat of vaporization from the air, the refrigerant is converted from a mixed state to saturated vapor at point **1**, where it again enters the compressor and completes the refrigeration cycle.

[0026] The coefficient of performance (COP) of a vapor compression refrigeration cycle measures the effectiveness of the cycle and is defined as the ratio of the useful refrigerating effect to the total power required to operate the cycle. For the case of a mechanical vapor compression refrigeration cycle, the total power required is usually consumed in driving the mechanical components including the compressor, the fans, the pumps etc. Mathematically:

$$COP = \frac{Q_{evaporator}}{W_{net}} \quad (1)$$

[0027] where, $Q_{evaporator}$=net rate of heat extracted by the evaporator from the ambient medium to be conditioned; and

[0028] W_{net}=Net power required to drive the cycle components, including the compressor, the fans and the pumps

[0029] As seen from equation (1) above, there are two possible ways to increase the COP of a vapor compression refrigeration cycle, one by increasing the rate of energy extraction from the ambient medium to be conditioned (Q_{evap}) and the other by decreasing the power input to the mechanical components, mainly the compressor (W_{net}).

[0030] As shown in FIG. **1**, if points **1** and **2** represent the states of the refrigerant during its entry into and exit from the compressor, respectively, and h_1 and h_2 represent the specific enthalpies (enthalpy per unit mass) of the refrigerant at states **1** and **2** respectively, and assuming the process **1→2** being reversible and adiabatic, then the amount of power required to drive the compressor is:

$$W_{net}=m(h_2-h_1) \quad (2)$$

[0031] where m=mass flow rate of the refrigerant at the inlet to the compressor

[0032] The quantity Q_{evap} in eq. (1) depends on the temperature of the ambient air flowing into the evaporator, which is uncontrollable to a certain extent, at a specific mass flow rate. Further, the magnitude of the higher refrigerant pressure value (P_2) corresponding to point **2** in FIG. **1**, depends upon the temperature of the air entering the condenser. The purpose of compressing the refrigerant is to increase its pressure, and hence its temperature, to a value where it can easily reject heat to the medium entering the condenser. In that sense, the refrigerant has to be compressed to an extent where the temperature of superheated vapor refrigerant (T_2) is greater than the temperature of the condenser inlet air. If somehow, this extent of compression is reduced (to a point somewhere between states **1** and **2**), to the point A or point B as represented in FIG. **1** for instance, the power required to drive the compressor would decrease consequently, leading to a tremendous increase in the coefficient of performance of the refrigeration cycle. Specifically, in a case where the refrigerant is compressed to a point A, instead of point **2**, the approximate power input to the compressor would be:

$$W'_{net}=m(h_a-h_1) \quad (3)$$

[0033] where h_a=specific enthalpy of refrigerant at A and $h_a<h_2, \Rightarrow W'_{net}<W_{net}$

[0034] The present disclosure uses this concept to substantially increase the coefficient of performance of a cooling unit while also effectively eliminating the problem of condensate removal.

[0035] FIG. **3** is a schematic diagram representing the components of cooling unit **300** in accordance with the present disclosure, as well as the heat exchange process at the evaporator and the condenser side. FIG. **4** further shows the different labeled components of the cooling unit shown in FIG. **3**. Explaining in conjunction with these figures, cooling unit **300** includes an evaporator **410**, a compressor **420**, a condenser **430**, a sub-cooling heat exchanger **450**, an expansion valve **440**, a condensate pump **460** and a number of spray nozzles **480**. The primary heat exchange medium flows along the circuit represented by solid thick lines shown in FIG. **3** and FIG. **4**, while condensate flows along the dotted-line circuit.

[0036] As illustrated, the primary heat exchange medium (refrigerant), in the form of saturated vapor, enters the compressor **420** at state **1** and converts into superheated vapor at point **2**′. The superheated vapor refrigerant rejects heat to air **490** blown into the condenser **430**, and as its temperature drops it converts first into saturated vapor and finally into saturated liquid at the exit **3**′ from the condenser **430**. The refrigerant is then routed through a sub-cooling heat exchanger **450**, where it exchanges heat with condensate **492** collected from the evaporator **410** through the condensate pump **460**. Thus, the refrigerant temperature further decreases and it converts from saturated liquid to sub-cooled liquid at point **3**″. The condensate handling system is discussed in detail below. Exiting from the sub-cooling heat exchanger, the refrigerant enters expansion valve **440**, where it expands and undergoes an abrupt decrease in pressure, thus lowering its temperature. A fraction of the liquid refrigerant undergoes partial flash evaporation, and it exits expansion valve **440** as a mixed liquid and vapor at point **4**′. Entering the evaporator **410**, the refrigerant, its temperature substantially lowered, extracts heat from ambient air **495**, and vaporizes completely before exiting the evaporator **410** at state **1**. The vaporized refrigerant reenters the compressor **420** to complete the refrigeration cycle.

[0037] The ambient air **495** generally has a certain absolute humidity, and the heat extraction process generally results in at least a portion of that moisture condensing as condensate **492** within the evaporator **410**. Condensate pump **460** routes this condensate **492** along a condensate flow path K→L→M as represented by the dotted lines in FIG. **4**. Following this path, the condensate **492** enters the sub-cooling heat exchanger **450** through a condensate inlet **452**, exchanges heat with the refrigerant flowing through the sub-cooling exchanger **450**, and finally exits through a condensate outlet **454**. Thereafter, the condensate **492** flows out through a condensate outlet pipe **485** connected to the condensate outlet **454**. A number of spray nozzles **480** are mounted spatially equidistant on the condensate outlet pipe **485**, and set of perforations **486** convey condensate **492** to the spray nozzles **480**.

[0038] The spray nozzles **480** are arranged to continuously spray condensate **492** into the air stream **490** flowing into condenser **430**. That flow can be assisted by a blower fan **470**, and the condensate spray acts on air stream **490** to reduce its temperature, as is clear to those of skill in the art. As a result, air stream **490** makes contact with the condenser coils at a reduced temperature, enhancing its ability to extract heat from the refrigerant.

[0039] Compressor **420** could be any suitable compressor known in the art usable for refrigerating units, and the selection of the exact compressor type is based on the operating conditions of the cooling unit **300** and certain associated design and maintenance criteria. To minimize losses in the refrigerant pressure and reduce the maintenance activities, and for cases where prolonged maintenance-free operations of the cooling unit **300** is desirable, a hermetically sealed or a semi-hermetic compressor would be preferred. An electric motor (not shown) is used a source of continuous power supply to the compressor **420** during operating conditions. Further, compressor **420** can be either a positive displacement or dynamic compressor depending upon the cooling load required and the environment of installation of the cooling unit **300**. In large buildings and warehouses demanding huge cooling capacity, a dynamic type single-stage centrifugal compression would be preferred. Cases wherein extremely high output pressures at the compressor outlet are required, a multi-staged centrifugal compressor would be preferably used. In certain embodiments, a variable displacement type compressor may be used wherein the mass flow rate of the refrigerant needs to be frequently controlled based on variations in the temperature of the air blown into the evaporator **410**.

[0040] Any suitable refrigerant known to those in the art can be used as the primary heat exchange medium for circulation within the cooling unit **300** for practicing the present disclosure. Common examples include the well-known family of refrigerants denoted by the 'R-number' system including R-11, R-22, R-134 (a) and others.

[0041] The sub-cooling heat exchanger **450** is be designed to provide sufficient capacity to incorporate high volume of condensate **492** generated during prolonged operations of the cooling unit **300**. Those skilled in the art would recognize that the size and capacity of the sub-cooling heat exchanger **450** can be varied based on the desired cooling capacity of the cooling unit **300**, as well as upon the expected conditions of ambient air. During inoperative conditions of the cooling unit **300**, the sub-cooling heat exchanger **450** would act as a reservoir for the condensate **492** collected during former operations. Further, the aperture of the condensate outlet **454** is preferably smaller than the aperture of the condensate inlet **452**, so that the volume of condensate entering the sub-cooling heat exchanger **450** is more than the volume leaving it. In that manner, a certain volume of condensate accumulates in the sub-cooling heat exchanger **450**, and thus the level of condensate rises during continuing operations of the cooling unit **300**. The condensate inlet **452** is provided at a higher elevation with respect to the condensate outlet **454** to ensure that there is a continuous influx of the condensate **492** within the sub-cooling heat exchanger **450**. Eventually, with the elevated level of the accumulated condensate **492** in the sub-cooling heat exchanger **450**, the condensate **492** flows out through the condensate outlet **454** into the condensate outlet pipe **485** by virtue of the achieved velocity of efflux. Further, the pressurized influx of the condensate **492** into the sub-cooling heat exchanger **450** by the condensate pump **460** provides additional expelling impulse for the volume of condensate **492** leaving the sub-cooling heat exchanger through the condensate outlet **454**.

[0042] Condensate pump **460** pressurizes the condensate **492** to flow through the sub-cooling heat exchanger **450** and on to spray nozzles **480**. The condensate velocity in that flow circuit should be sufficient to allow the spray nozzles **480** to discharge the condensate as a spray of fine droplets, maximizing heat exchange between the condensate and the air stream **490**. To minimize energy consumption, hydraulic spray nozzles are preferred. However, those skilled in the art would understand that any other type of spray nozzles conventionally known in the art may also be used, including gas atomized spray nozzles, thus, not limiting the scope of the invention.

[0043] The sub-cooling heat exchanger **450** further includes a primary heat exchange medium inlet **455** and a primary heat exchange medium outlet **456** for a continuous influx and discharge of refrigerant. Exiting the condenser **430**, the refrigerant enters the sub-cooling heat exchanger **450** inlet **455**, flows through a set of cooling coils **458**, and finally emerges through outlet **456**. That flow allows the refrigerant to reject heat to the condensate **492**, and to emerge at point $\mathbf{3''}$ as a sub-cooled liquid. This decreases the temperature of the refrigerant as it exits the sub-cooling heat exchanger **450**. This heat exchange is an isobaric process, occurring at the constant saturation pressure of the saturated liquid refrigerant exiting the condenser **430** at point $\mathbf{3'}$. The specific enthalpy of the refrigerant decreases as it moves from the state $\mathbf{3'}$ to the state $\mathbf{3''}$.

[0044] FIG. **5** illustrates the evaporator **410** to be used in the cooling unit **300** of FIG. **4**, in further detail. Evaporator **410** includes a casing **501** having an anterior surface **502** and a posterior surface **503**. A set of heat exchange tubes **504** are disposed in parallel within the casing **501**. The heat exchange tubes **504** extend along the entire length of the casing **501**, lying substantially perpendicular to the anterior and posterior surfaces **502** and **503** respectively. Further, the anterior surface **502** and posterior surface **503** have perforations through which the heat exchange tubes extend into and out of casing **501**. Fins **505** are equidistantly disposed within the casing over the heat exchange tubes **504**. Those skilled in the art would recognize that different evaporator structures could easily be substituted for the disclosed component without altering the performance of the disclosed embodiment.

[0045] FIG. **6** illustrates the pressure-enthalpy diagram for the refrigerant circulating in the cooling unit **300** of the present disclosure. The solid lined path $\mathbf{1 \rightarrow 2' \rightarrow 2'' \rightarrow 3' \rightarrow 3'' \rightarrow 4'} > \mathbf{1}$ represents the respective states marked in FIG. **3** and FIG. **4**, for the refrigerant flowing in the cooling unit **300** according to the present disclosure. Path $\mathbf{1 \rightarrow 2' \rightarrow 2 \rightarrow 3 \rightarrow 4 \rightarrow 1}$ (dotted lines) represents the pressure-enthalpy variation of the refrigerant if it flows through a conventional cooling unit based on the vapor-compression refrigeration cycle and not equipped with a sub-cooling heat exchanger and spray nozzles, as shown by FIG. **2**. The horizontal axis OX represents the specific enthalpy (enthalpy per unit mass) of the refrigerant at different thermodynamic state points, while the vertical axis OB represents the pressure of the refrigerant at different states. The two superimposed paths lay out a comparison between conventional cooling units and the inventive cooling unit **300** of FIG. **4**.

[0046] As shown, in conventional refrigerating units, the air at the inlet to the condenser is at a relatively higher temperature, and hence the refrigerant is compressed to a higher pressure P_2 and correspondingly a higher temperature T_2, so that it can easily reject heat to the air entering the condenser. In the inventive cooling unit **300**, the air blown into the condenser **430** is sprinkled with condensate **492** by the spray nozzles **480** (FIG. **4**), before that air stream enters the condenser **430**. Hence, the temperature of the air **490** entering the condenser **430** is relatively lower. Therefore, the refrigerant in the cooling unit **300** needs to be compressed to a lower pressure represented by a point $\mathbf{2'}$ lying somewhere on the curve joining the points **1** and **2** in FIG. **6**. The exact location of this point $\mathbf{2'}$ on the curved line $\mathbf{1 \rightarrow 2}$ depends on certain factors, the primary ones being the temperature of the air **490** entering the condenser **430**, its mass flow rate and the mass flow rate of the refrigerant circulating in the cooling unit **300**.

[0047] Further, during process $\mathbf{3' \rightarrow 3''}$, the refrigerant flows through the sub-cooling heat exchanger where it rejects heat to the condensate and emerges as a sub-cooled liquid at point $\mathbf{3''}$. Process $\mathbf{3'' \rightarrow 4'}$ represents the expansion of the refrigerant within the expansion valve **440**. This expansion abruptly decreases the pressure of the refrigerant from P_2' to P_1, though the enthalpy of the refrigerant remains at a constant value h_4' during the process. In comparison, the refrigerant in conventional cooling unit of FIG. **2** would undergo the path **2→3** in the condenser, emerge as a saturated liquid at point **3**, and finally expand in the expansion valve isobarically to the point **4** (process **3→4** in FIG. **6**), consequently decreasing its pressure from $P_{2\ to\ P1}$.

[0048] Portion **510** on axis OX in FIG. **6** represents the decrease in the required power input to the compressor achieved by the inventive cooling unit **300** of FIG. **4**, in comparison to a conventional cooling unit of FIG. **2**. As set out earlier in Eq. (2), the power required to drive the compressor of a conventional cooling unit of FIG. **2** is:

$$W_{net}=\dot{m}(h_2-h_1) \tag{2}$$

[0049] where, ṁ=mass flow rate of the refrigerant flowing in the vapor-compression refrigeration cycle.

[0050] Using the same refrigerant, at the same mass flow rate ṁ, the power required to drive the compressor of the inventive cooling unit of FIG. **4** is:

$$W'_{net}=\dot{m}(h_2'-h_1) \tag{3}$$

[0051] Since $h_2'<h_2$ (as seen in FIG. **5**), $W'_{net}<W_{net}$

[0052] The decreased power input to the compressor, as achieved by the cooling unit **300** of FIG. **4** is obtainable by subtracting Eq. (3) from Eq. (2), and is:

$$\Delta W=W_{net}-W'_{net}=\dot{m}(h_2-h_2') \tag{4}$$

[0053] The length of the portion **510** on the enthalpy axis OX of FIG. **5** actually represents this decreased compressor power input per unit mass flow rate of the refrigerant ($\Delta W/\dot{m}$), as achieved by the cooling unit **300** of FIG. **4**.

[0054] Further, portion **520** in FIG. **6** represents the enhanced cooling effect achieved by the cooling unit **300** of FIG. **4**. This enhanced cooling effect **520** is equivalent to the additional heat energy extracted by the evaporator **410** of cooling unit **300** (denoted by Q'_{evap}) from the ambient air **495** to be conditioned, in comparison to the heat actually extracted by the evaporator of a conventional cooling unit of FIG. **2** (denoted by Q_{evap}).

[0055] For a conventional cooling unit of FIG. **2**, the refrigerant undergoes the process **4→1** (shown in FIG. **6**) in the evaporator and hence, the heat gained by the refrigerant from the ambient air to be conditioned is:

$$Q_{evap}=\dot{m}(h_1-h_4) \tag{5}$$

[0056] In the inventive cooling unit **300**, the refrigerant is further sub-cooled to state **3''** and hence undergoes the process represented by **4'→1** within the evaporator. Therefore, the heat gained by the refrigerant is:

$$Q'_{evap}=\dot{m}(h_1-h_4') \tag{6}$$

[0057] Since, $h_4'<h_4$, we have $Q'_{evap}>Q_{evap}$

[0058] The approximate enhanced cooling effect gained by the cooling unit **300**, can be obtained from Eq. (5) and Eq. (6) above, and is:

$$\Delta Q_{evap}=Q'_{evap}-Q_{evap}=\dot{m}(h_4-h_4') \tag{7}$$

[0059] The length of the portion **520** on the enthalpy axis OX of FIG. **5** actually represents this enhanced cooling effect per unit mass flow rate of the refrigerant, i.e., $\Delta Q_{evap}/\dot{m}$.

[0060] The coefficient of performance of the cooling unit **300** of FIG. **4** is substantially increased in this manner, compared with the COP of the conventional cooling units utilizing vapor-compression refrigeration cycles. As stated in eq. (i) earlier, the COP of a refrigeration cycle is given by:

$$COP=\frac{Q_{evaporator}}{W_{net}} \tag{1}$$

[0061] Referring to FIG. **6**, the value of COP for a conventional unit of FIG. **3** would be:

$$COP=\frac{Q_{evaporator}}{W_{net}}=\frac{(h_1-h_4)}{(h_2-h_1)} \qquad (8)\ \text{[from eq (2) and (5)]}$$

[0062] For the inventive cooling unit **300** of FIG. **4**, the enhanced COP is given by:

$$COP'=\frac{(h_1-h_4')}{(h_2'-h_1)} \qquad (9)\ \text{[from eq (3) and (4)]}$$

[0063] Equations (8) and (9) clearly depict the increase in Coefficient of performance obtained by the cooling unit **300** of FIG. **4**. For the same refrigerant and the same value its mass flow rate in the refrigeration cycle, the cooling unit **300** obtains a substantially enhanced cooling capacity with a considerably lower value of the power required to drive the compressor **420**, as compared to the conventional cooling unit of FIG. **3**. Additionally, the problem of condensate removal from the cooling units based on vapor-compression refrigeration cycles is significantly alleviated by the inventive cooling unit **300**.

[0064] FIG. **7** further illustrates the heat transfer phenomenon within the condenser of the cooling unit **300** of FIG. **4** in accordance with the present disclosure. As shown, control volume **700** surrounds the condenser **430** and all the heat exchange processes at the condenser side occur within the control volume **700**. Since the condenser **430** is air-cooled, hot outdoor air **710** acquires an average temperature T_1 after the condensate **492** is sprinkled into the airstream, enters the control volume **700** and leaves the condenser **430** at a higher average temperature T_m after gaining heat from the refrigerant. Refrigerant **730** enters the condenser **430** at a pressure P_2' and temperature T_2' (corresponding to the state **2'** marked in FIG. **6**) and emerges at the same pressure but a lower temperature T_3' after rejecting heat to the hot air **710** blown into the condenser **430**. The temperature values T_1 and T_m for the hot air entering and exiting the control volume **700** are measurable through any suitable temperature detecting device known in the art, and conventionally used for the purpose, including a thermocouple, a thermostat etc. Neglecting the heat gained by the interiors of the condenser **430**, including the coil tubes through which the refrigerant **730** flows, and the minute heat losses to the surroundings, the major part of the heat lost by the refrigerant **730** goes into the hot air **710** entering the condenser **430**, thus resulting in an elevation in its temperature from an average inlet temperature value T_1 to the outlet value T_m.

[0065] Let $\dot{m}_{air}$=mass flow rate of the air blown into the condenser;

[0066] $\dot{m}_{ref}$=mass flow rate of the refrigerant through the vapor-compression cycle for the cooling unit **300**

[0067] C=average specific heat capacity of the hot air within temperature range T_1 to T_m

[0068] h_2', h_3'=enthalpies of the refrigerant at entry point **2'** and exit point **3'**, as marked in the pressure-enthalpy diagram of FIG. **6**

[0069] q_{ref}=average rate of heat lost by the refrigerant **730** within the control volume **700**

[0070] q_{air}=average rate of heat gained by the hot air **710** within the control volume **700**

[0071] We have the following approximations under above mentioned considerations:

$$q_{ref}=\dot{m}_{ref}(h_2'-h_3') \tag{10}$$

$$q_{air}=\dot{m}_{air}C(T_m-T_1) \tag{11}$$

[0072] Neglecting the minor losses within the condenser:

$$q_{ref}=q_{air}$$

$$\dot{m}_{ref}(h_2'-h_3')=\dot{m}_{air}C(T_m-T_1) \text{ [By equating eqns. (10) and (11)]}$$

$$(h_2'-h_3')=\dot{m}_{air}C(T_m-T_1)/\dot{m}_{ref} \tag{12}$$

[0073] This calculated value of the change in enthalpy of the refrigerant during its flow through the condenser is used as a reference for approximating the extent of the point **2'** (shown in FIG. **6**) until which the refrigerant needs to be compressed by the compressor **420** of the cooling unit **300**.

[0074] For a specific temperature (T_1) of the air **710** at the inlet to the condenser **700**, the value of the higher pressure P_2' to which the refrigerant should undergo compression in the compressor **420**, for maximizing the cooling capacity of the cooling unit, is obtainable by the standard pressure-enthalpy chart for the refrigerant used. With known values of the mass flow rate of the refrigerant ($\dot{m}_{ref}$) and the air ($\dot{m}_{air}$), and an approximate value of the specific heat capacity of the air in the temperature range T_1 to T_m, the corresponding difference between the enthalpies of the superheated vapor refrigerant (entering the condenser) and the saturated liquid refrigerant (exiting the condenser), i.e., ($h_2'-h_3'$), is calculated from Eq. (12).

[0075] The high pressure value of the refrigerant (P_2') is calibrated to be in conformity with the enthalpy difference calculated in eq. (xii) using the standard pressure-enthalpy chart for the refrigerant used, and this calibrated value P_2' is set for the compressor **420** of the cooling unit **300**. As an example, using R-134 (a) as the working refrigerant fluid, with the high pressure side saturated vapor temperature of 50° C. (considering hot summer outdoor air entering the condenser), the enthalpies of the saturated liquid refrigerant and the saturated vapor refrigerant are found to be 270 KJ/Kg. K and 420 KJ/Kg. K respectively, from the standard pressure-enthalpy chart for R-134 (a). Considering specific operating values of the mass flow rate of the hot air and the refrigerant in Eq. (12), if the enthalpy difference h_2'–hd **3'** is 200 KJ/Kg K (for instance), then the refrigerant needs to be compressed to a pressure of about 1.4 Mpa (P_2'), and to a temperature of about 90° C. (T_2'), for proper functioning of the cooling unit **300**.

[0076] Although the present invention has been described in considerable details with reference to certain preferred versions thereof, other versions are also possible.

[0077] The cooling unit as disclosed herein can be used in several circumstances where a refrigerating effect is desired. In an aspect, the cooling unit can be an integral part of a usual air-conditioning systems utilized in homes or other buildings. As another example, several such cooling units can be simultaneously used in collaboration for commercial and industrial applications where large scale air-conditioning is required, including residential buildings, factories etc. As a further example, the unit can also be used in conditioning the air in movie theatres, concert halls, restaurants, cafeteria etc. The appropriate method of use would be to install the evaporator of the cooling unit at a suitable location within the space where the refrigerating effect is desired such that it can extract heat from the space, condition the air and reject this heat elsewhere. These and other variations are well within the scope of those of ordinary skill in the art.

I claim:

1. A cooling unit employing vapor compression refrigeration cycle for cooling an ambient medium, the cooling unit comprising:

an evaporator, including a plurality of heat exchange tubes adapted to carry a primary heat exchange medium, and arranged to receive a flow of an ambient medium;

a condensate pump positioned to route condensate from the evaporator;

a sub-cooling heat exchanger in fluid communication with the condensate pump; and

a plurality of spray nozzles in simultaneous fluid communication with the sub-cooling heat exchanger and positioned to spray condensate into the flow of the ambient medium.

2. The cooling unit of claim **1**, wherein the sub-cooling heat exchanger has a condensate inlet for a continuous inflow of the condensate therein, and a condensate outlet for a continuous outflow of the condensate therefrom.

3. The cooling unit of claim **2**, wherein the condensate outlet has a cross-sectional area preferably smaller than the cross-sectional area of the condensate inlet.

4. The cooling unit of claim **2**, wherein the condensate inlet is positioned at a higher elevation with respect to the condensate outlet.

5. The cooling unit of claim **2**, wherein the sub-cooling heat exchanger is in fluid communication with a condensate outlet pipe, the condensate outlet pipe having the spray nozzles mounted.thereon.

6. The cooling unit of claim **5**, wherein the condensate outlet pipe is provided with a set of perforations to fluidly communicate with the plurality of spray nozzles.

7. The cooling unit of claim **1**, wherein the sub-cooling heat exchanger has a primary heat exchange medium inlet for a continuous inflow of the primary heat exchange medium therein, and a primary heat exchange medium outlet for a continuous outflow of the primary heat exchange medium therefrom.

8. The cooling unit of claim **1**, wherein the primary heat exchange medium and the condensate are in thermal communication within the sub-cooling heat exchanger.

9. The cooling unit of claim **1**, wherein the sub-cooling heat exchanger is in fluid communication with least one of the plurality of spray nozzles.

10. The cooling unit of claim **1**, wherein the plurality of spray nozzles are arranged in a spaced array.

11. The cooling unit of claim **1**, wherein the plurality of spray nozzles are coupled to a condenser, and configured to continuously spray the condensate on air blown into the condenser.

12. A method of increasing the coefficient of performance of a cooling unit utilizing vapor compression refrigeration cycle, the cooling unit including an evaporator, a compressor, a condenser and an expansion valve, the method comprising:

collecting condensate from the evaporator;

routing condensate to a sub-cooling heat exchanger;

exchanging heat in the sub-cooling heat exchanger between the condensate and a primary heat exchange medium;

directing the condensate from the sub-cooling heat exchanger to a plurality of spray nozzles; and

sprinkling the condensate through the plurality of spray nozzles over air blown into the condenser.

13. The method of claim **12** wherein the cooling unit includes a compressor, and a condenser configured to receive a flow of an ambient medium, the method further comprising compressing the primary heat exchange medium within the compressor to a high-pressure value based at least on a set of parameters corresponding to the ambient medium.

14. The method of claim **13**, wherein the set of parameters includes a mass-flow rate value of the ambient medium flowing through the condenser.

15. The method of claim **13**, wherein the set of parameters include the corresponding average temperature values of the ambient medium entering and leaving the condenser.

16. The method of claim **12**, wherein the condensate remains in continuous thermal communication with a primary heat exchange medium within the sub-cooling heat exchanger.

17. The method of claim **12** further comprising directing the condensate through a condensate outlet pipe from the sub-cooling heat exchanger to the spray nozzles.

18. The method of claim **17** further comprising providing a set of perforations on the condensate outlet pipe to enable it being in simultaneous thermal communication with the plurality of spray nozzles.

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