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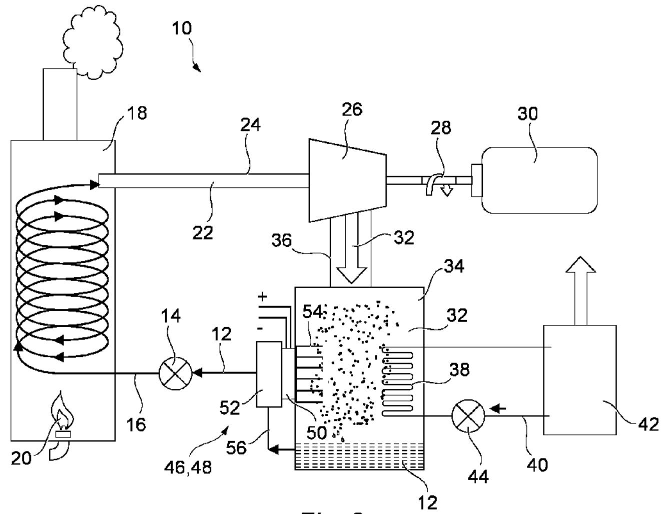

Fig. 2

(57) Abstract: The present disclosure relates to condensers for use in condensing a working fluid in a thermal cycle apparatus prior to its re-entry into a working fluid line and onto a boiler, the condenser including a heat exchanger with a cold surface portion and a hot surface portion, the cold surface portion being disposed within a condensing chamber of the condenser, and the hot surface portion being deployed such that, in use, it transfers heat to condensed working fluid in a working fluid line prior to said condensed working fluid entering a boiler. It also covers thermal power stations and thermal cycles using such an apparatus, methods of improving thermal power stations and thermal cycles using such an apparatus. A temperature sink is also provided, which may be used in the condenser, other thermal power stations and thermal cycles using such an apparatus.

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Method & Apparatus

Description

Field of the Invention

The present invention relates to the improvement of efficiency of thermal cycles, especially those used in the generation of electricity from thermal sources, and apparatus relating to it.

Background of the Invention

Thermal power stations use a heat source in order to raise the temperature of a working fluid, most commonly water, turning it into a vapour at elevated pressure and temperature, usually therefore superheated steam, and extracting useful work by expanding the vapour through a series of turbines. These turbines effectively turn the heat generated by the fuel into mechanical work, and this mechanical work is in turn converted to electrical energy. The heat source may be of many different types, from traditional coal-fired, nuclear, gas-fired, etc onto more advanced types such as solar, solar-fossil fuel hybrids, etc.

The fluid in a thermal power station will be in a closed loop i.e. the same fluid will be used continuously. The main reason for this is that it must be sufficiently pure such that damage to the various components within the loop is mitigated.

The thermal cycle most commonly used is the Rankine cycle – a practical variant of the theoretical Carnot cycle.

Fig. 1 is a schematic depiction of the typical components within a simplified Rankine cycle power station.

Feed water FW (the common expression for water as working fluid within a power station) is pressurised by a pump P and forced towards a boiler B through pipe work PW. The boiler B is heated by a heat source H, which may be fuelled by, for example, coal.

The feed water FW has its pressure and temperature raised in the boiler B until changing phase into a vapour or gas, often into superheated or dry steam S.

The steam S is fed into a turbine TB which extracts a portion of the thermal energy of the steam S and converts that to mechanical work W. The mechanical work W will then be converted to electrical energy by powering an electrical generator EG.

Often, for efficiency, several turbines will be used in series: high pressure, intermediate pressure and low pressure, the use of which increases the overall efficiency of the process. For simplicity, Fig. 1 depicts a single turbine TB.

Once the steam S has exited the turbine TB (or last low pressure turbine in a multi-turbine arrangement) the steam will still be in a vapour phase VP, and will still possess some latent heat.

The efficiency of the overall process and the laws of thermodynamics mean that not all heat added may be extracted in the form of mechanical work.

A condenser C is required to return the wet vapour steam VP into the saturated liquid form of the feed water FW. The condenser C may be of various designs, but is commonly a shell and tube design. In a shell and tube design, an outer shell includes a plurality of tubes T within it, through which liquid water (not shown) is pumped. The wet steam is fed into the shell, and as it passes over the tubes T, the heat flows through the tubes and into the liquid water within them. The wet vapour eventually reaches the dew point, and it changes phase back to a saturated liquid. The thermal energy extracted from the wet vapour will generally be vented to the atmosphere usually via a cooling tower CT. Importantly, heat must be extracted from the wet vapour VP for condensation to occur.

This saturated liquid form of the feed water FW exiting the condenser then passes back through pump P before repeating the cycle.

Summary of the Invention

According to a first aspect of the present invention there is provided a condenser for use in condensing a working fluid in a thermal cycle apparatus prior to its re-entry into a working fluid line and onto a boiler, the condenser including a heat exchanger with a cold surface portion and a hot surface portion, the cold surface portion being disposed within a condensing chamber of the condenser, and the hot surface portion being deployed such that, in use, it transfers heat to condensed working fluid in a working fluid line prior to said condensed working fluid entering a boiler.

By effectively “scavenging” some of the lost heat in the system by using it to preheat the working fluid, efficiency, energy extracted and/or the overall process efficiency may improve.

It will be appreciated that “cold” and “hot” in this context are terms relative to one another, and mean simply that, in use, the hot surface portion should be at a greater temperature than the cold surface portion, be that by fractions of a degree, or a far greater temperature difference.

Preferably the heat exchanger includes a heat pump. More preferably the heat exchanger includes a thermo-electric module.

A thermo-electric module can produce electricity from a heat difference, but may also operate in reverse (a function of the Seebeck effect) i.e. a heat difference may be created by applying a voltage to it (a function of the Peltier effect).

By locating the cold surface portion of the heat pump/exchanger within the condenser and said cold surface portion being at a lower temperature than the condenser generally, and particularly below the dew point of the working fluid when in its gaseous phase, a preferential site for the working fluid to condense upon is provided.

Preferably condensate channels are provided on or adjacent to said cold surface portion.

Preferably the part of the working fluid line to which the hot surface portion is adjacent is located upstream of the boiler, more preferably upstream of a pressurisation pump used to pressurise working fluid prior to its entry into the boiler.

Preferably the cold surface portion includes a cold sink. Preferably the cold sink comprises one or more cold sink fins, the cold sink fins preferably tapering from a first thickness adjacent the cold surface to a second thickness at a distal end, the first thickness being greater than the second thickness.

Preferably the hot surface portion includes a heat sink. Preferably the heat sink comprises one or more heat sink fins, the heat sink fins preferably tapering from a first thickness adjacent the hot side to a second thickness at a distal end, the first thickness being greater than the second thickness.

The hot side of the heat sink may optionally include a labyrinth pipe design as a component thereof.

Preferably the cold sink fin(s) comprise a material that results in a temperature gradient forming from a tip of the cold sink fin(s) to a root of the cold sink fin(s) when in use; preferably said temperature gradient is from a higher temperature at the fin tip to a lower temperature at the fin root.

More preferably a portion of said cold sink fin(s) is hydrophobic. The hydrophobia may be provided either by a hydrophobic coating or an intrinsic property of the material from which the portion and/or fin is constructed.

Preferably the heat sink fin(s) have a portion which is hydrophobic. The hydrophobia may be provided either by a hydrophobic coating or an intrinsic property of the material from which the portion and/or fin is constructed.

The hydrophobic property means that as working fluid condenses onto the cold sink, it is repelled from the surface and frees up surface area to allow further condensation. By “hydrophobic” it will be understood by the skilled addressee that it is intended to cover more than simply materials that repel water, but materials which will tend to repel the particular working fluid used e.g. a CO_2 repellent material if that is the working fluid etc.

Preferably the hydrophobic portion of said fins is limited to less than the total surface of said fin(s). This provides non-hydrophobic areas where condensate may be retained for longer, thus being more susceptible to having its temperature raised by the ambient temperature of the condenser environment.

Preferably said hydrophobic portion is disposed at the fin root and the non-hydrophobic portion toward the fin tip.

Preferably the tapering of said fins is generally hyperbolic, or may alternatively be generally exponential. The expressions “hyperbolic” and “exponential” used here in the sense of being generally descriptive of the relationship between the fin thickness and its length.

Optionally, the fins may be of uniform cross-sectional shape such that the distal end of the section when extruded forms a line (single dimensioned). Alternatively, the cross-section may be axially rotated such that the resultant volume comes to a point (zero dimensional).

Preferably the cold sink fins are orientated downwardly in the condenser, thus gravity aids in removing working fluid condensate from the cold sink.

According to a second aspect of the present invention there is provided a thermal power station employing the condenser of the first aspect of the present invention.

According to a third aspect of the present invention there is provided a method of operating a thermal cycle involving a working fluid comprising the steps of pressurising the working fluid in a substantially liquid phase, applying heat to the working fluid to transfer it to a substantially gaseous phase, extracting mechanical energy from the gaseous phase, condensing the working fluid to a liquid phase and therefore extracting heat, using said extracted heat to preheat the working fluid prior to applying further heat to the working fluid.

Preferably said extracted heat is applied prior to the working fluid being re-pressurised in a liquid phase.

According to a fourth aspect of the present invention there is provided a thermal power station employing the method of the third aspect of the present invention.

According to a fifth aspect there is provided a method of modifying a thermal power station including the steps of fitting a condenser according to a first aspect of the present invention.

According to a sixth aspect of the present invention there is provided a temperature sink comprising a portion of hydrophobic material. This may be in the form of a coating or as an intrinsic property of the material from which the temperature sink is made. By temperature sink it will be taken to be the collective term for a heat sink or a cold sink.

Preferably, the temperature sink comprises one or more fins, the fins preferably tapering from a first thickness adjacent the cold side to a second thickness at a distal end, the first thickness being greater than the second thickness.

Preferably the hydrophobic portion of said fins is limited to less than the total surface of said fin(s).

Preferably said hydrophobic portion is disposed at the fin root and the non-hydrophobic portion toward the fin tip.

Preferably the tapering of said fins is generally hyperbolic, or may alternatively be generally exponential. The expressions “hyperbolic” and “exponential” used in this context as being generally descriptive of the relationship between the fin thickness and its length.

According to a seventh aspect of the present invention there is provided a heat exchanger including at least one temperature sink according to a sixth aspect of the present invention.

According to an eighth aspect of the present invention there is provided a heat pump including at least one temperature sink according to a sixth aspect of the present invention.

According to a ninth aspect of the present invention there is provided a thermal power station employing at least one temperature sink according to a sixth aspect of the present invention or a heat exchanger according to a seventh aspect of the present invention or a heat pump according to an eighth aspect of the present invention.

According to a tenth aspect of the present invention there is provided a condenser suitable for use in a thermal cycle including at least one thermo-electric module.

According to an eleventh aspect of the present invention there is provided apparatus for operating a thermal cycle including at least one condenser according to the tenth aspect.

By “thermal” it will be understood that this may be of many different types, from traditional coal-fired, nuclear, gas-fired, etc onto more advanced types such as solar, solar-fossil fuel hybrids, etc.

Brief Description of the Drawings

An embodiment of the present invention will now be described with reference to the following drawings in which:

Fig. 2 is a schematic view of a thermal cycle apparatus and method according to the first and second aspects;

Fig. 3 is a diagrammatic side view of a cold sink according to the fifth aspect of the present invention and the thermal cycle apparatus of Fig. 2;

Fig. 4 is a diagrammatic side view of a heat sink according to the fifth aspect of the present invention and the thermal cycle apparatus of Fig. 2;

Fig. 5 is diagrammatic front view of an arrangement of cold sinks according to the fifth aspect of the present invention and the thermal cycle apparatus of Fig. 2;

Fig. 6 is diagrammatic front view of an alternative arrangement of a condenser including cold sinks according to the fifth aspect of the present invention and compatible with the thermal cycle apparatus of Fig. 2;

Fig. 7 is a diagrammatic detail view of Fig. 6 showing details of heat flow;

Fig. 8 is diagrammatic front view of a further alternative arrangement of a condenser including cold sinks according to the fifth aspect of the present invention and compatible with the thermal cycle apparatus of Fig. 2;

Fig. 9 is a plan view of the base of a condenser compatible with the thermal cycle apparatus of Fig. 2 which includes a thermal equaliser;

Fig. 10 is an isometric view of the base of a condenser of Fig. 7;

Fig. 11 is a diagrammatic view of a further embodiment of apparatus according to the present invention.

Turning first to Fig. 2, thermal cycle apparatus 10 is depicted. This is a simplified schematic representation of the apparatus as it may be provided in a power station.

Feed water 12 is pressurised by pump 14 through pipe work 16 and into a boiler 18. The boiler 18 includes a heat source 20. The heat source 20 may be of any suitable type, such as coal-fired, gas-fired, nuclear, solar, etc. The feed water 12 is raised in temperature and pressure until passing to a dry steam phase 22. Pipe work 24 delivers the dry steam phase 22 to a turbine 26. The dry steam phase 22 is expanded in the turbine 26 and mechanical work 28 is therefore extracted. The mechanical work 28 is used to drive an electrical generator 30 to produce electrical power.

The dry steam phase 22 becomes a wet vapour phase 32 and exits turbine 26 to be delivered to a condenser 34 by pipe work 36.

The condenser 34 is of a shell and tube design and includes tubes 38 through which cooling water 40 flows. The tubes 38 form part of a cycle which also includes a cooling tower 42 and a cooling water pump 44.

A heat exchanger 46 which includes a heat pump 48 is also provided on a sidewall of the condenser 34. The heat pump 48 in this case comprises a thermo-electric module (or “TEM”) 50 which is being operated in reverse i.e. instead of using a temperature difference to create a voltage; a voltage is applied to the TEM 50 to create a temperature difference.

This produces a hot side 52 and a cold side 54; the cold side 54 being located within and facing into the condenser 34.

Whereas the ambient condenser temperature may be about 30° Celsius, the cold side 54 in the present embodiment will be reduced to a temperature of about 20° Celsius. This provides a preferential site for the wet vapour phase 32 to condense back into saturated liquid feed water 12.

The saturated liquid feed water 12 falls to the base of the condenser 34 and is removed by pipe work 56. Pipe work 56 delivers the feed water 12 past or through the hot side 52 of the heat exchanger 46.

In the present embodiment, the hot side 52 of the heat exchanger 46 will be at a temperature of about 50° Celsius. This will provide a preheating of the feed water 12 prior to it being pressurised by pump 14 through pipe work 16 and into the boiler 18.

Importantly, the coefficient of performance of the heat pump 48 is a measure of the efficiency at which the heat exchange takes place. A heat pump with a coefficient of performance (“COP”) of 3 uses 1 unit of electrical energy to “move” 2 units of thermal energy. For example, every 1kWh of electrical energy used to drive the heat pump 48 results in 3kWh of thermal energy being transferred to the hot side 52. The cold side has 2kWh of energy removed from it and therefore the COP (cooling) is 2.

The electrical energy may be provided by an external source, or may more likely be drawn from the electrical generator 30.

The preheating effect provides an increase in overall thermal efficiency of the thermal cycle. Since much of the design is of a known type, it is possible to modify existing thermal cycle apparatus by the addition of the heat pump etc to prior art condensers, or indeed to replace a prior art condenser with one of a similar design to the present embodiment.

Turning to Figs. 3 and 4, detail of a cold sink 58 and a heat sink 60 associated with the cold side 54 and hot side 52 are depicted. Both are of a similar design.

The cold sink 58 comprises a cold sink back plate 58a and a plurality of cold sink fins 58b projecting from the cold sink back plate 58a. In use, the cold sink back plate 58a would attach on its first face 58c to the cold side of the heat exchanger 46, with the cold sink fins 58 projecting into the condenser 34.

The cold sink fins 58b generally taper in thickness from a first and greater thickness at their first end which is attached to the cold sink back plate 58a, to a second and lesser thickness at their distal end; effectively tapering to a narrow edge. The tapering profile is of a generally hyperbolic or exponential nature i.e. the relative thickness of the cold sink fins 58b compared to the distance from the cold sink back plate 58a decreases sharply initially, before becoming more gradual.

An initial portion 58d (the shaded area of Fig. 3) of the cold sink fins 58b is coated with a hydrophobic coating.

The design of the cold sink 58 is such that it provides a large surface area to maximise the conversion of the steam of the wet vapour phase 32 to feed water droplets 12. These feed water droplets 12 need to be removed from the cold sink 58 quickly: their presence reduces the surface area available for condensation. Some condensation will occur on the feed water droplets which are cooler than the ambient temperature of the condenser 34, but this is less efficient than directly on the surface of the cold sink 58. Channels 58e are provided between the cold sink fins 58a to move the feed water droplets 12 without significantly reducing the area for condensation.

It should be noted that since the cold side 54 is less than the ambient temperature of the condenser 34, the feed water 12 will condense on the cold side 54 at this temperature i.e. about 20° Celsius. Since it is preferable for the hot side 52 to raise the temperature of the feed water 12, it is preferable if the feed water 12 leaving the condenser is at ambient condenser temperature rather than cold side temperature i.e. it is preferable if the feed water 12 exits the condenser at closer to 30° Celsius than 20° Celsius.

The hydrophobic coating reduces wetting of the fins to minimise chilling of the condensate feed water 12.

The hydrophobic coating is selectively applied to regions of the cold sink 58 where efficient condensate feed water 12 removal is required. In other areas, i.e. towards the tips of the fins 58b, the condensate feed water 12 tends to be retained longer such that the steam environment in the condenser 34 heats the condensate feed water 12 before it drops to the bottom of the condenser 34 for subsequent pumping.

The thermal conductivity of the material used for the cold sink 58 is such that the fin tips are close to ambient temperature of the condenser and a gradient exists along the fin back to the root. Condensate feed water 12 flowing across the fin surface tends to the temperature of the fin at that point and hence “cold” condensate feed water 12 tends to rise in temperature before leaving the fin tips.

In the condenser 34, the cold sink will be preferably orientated such that the fins 58b point downwards, such that gravity assists in moving condensate feed water 12 from the channels 58e across the fins from root to tip away from the hydrophobic coating area 58d and, importantly, across the temperature gradient from “cold” (about 20° Celsius in the present embodiment) to “hot” (about 50° Celsius in the present embodiment).

The heat sink 60 likewise comprises a heat sink back plate 60a and a plurality of heat sink fins 60b projecting from the heat sink back plate 60a. In use, the heat sink back plate 60a would attach on its first face 60c to the hot side of the heat exchanger 46, with the heat sink fins 60 projecting away from it and the condenser 34. Through the heat sink back plate 60a is a feed water channel 60d through which feed water 12 is fed.

The heat sink fins 60b generally taper in thickness from a first and greater thickness at their first end which is attached to the heat sink back plate 60a, to a second and lesser thickness at their distal end; effectively tapering to a narrow edge. The tapering profile is of a generally hyperbolic or exponential nature i.e. the relative thickness of the heat sink fins 60b compared to the distance from the heat sink back plate 60a decreases sharply initially, before becoming more gradual.

An initial portion 60d (the shaded area of Fig. 4) of the heat sink fins 60b is coated with a hydrophobic coating.

The designs are similar such that they may be interchanged and effectively manufactured as one item.

Turning to Fig. 5, an arrangement of several cold sinks 58 within a condenser is shown. This grid-like arrangement includes nine cold sinks, arranged in a grid comprising three row and three columns of cold sinks 58.

Between the columns of cold sinks 58 are inter-sink condensate channels 62, effectively indented channels running vertically between the cold sinks. Below the cold sinks, and substantially in fluid communication with the inter-sink condensate channels 62 are collection channels 64. As condensate feed water 12 flows downwards off the cold sinks 58. Should it simply flow onto a cold sink 58 beneath the one on which it initially condensed, much of the disadvantages described above may occur. However, the collection channels 64 will tend to channel the condensate feed water 12 toward the inter-sink condensate channels 62 and onto the base of the condenser 34, avoiding needlessly wetting the cold sinks 58. The collection channels 64 are suitably designed projections from the side wall 34a of the condenser 34.

A schematic side view of a similar arrangement is shown in Figs. 6 & 7. Three rows of cold sinks 58 are shown: the bottom-most row (i.e. nearest the condenser 34 base) is labelled 58', the middle row 58" and finally the top-most row 58'''. The cold sinks 58', 58" & 58''' are attached to corresponding heat exchangers 46', 46" & 46''', and they in turn are attached to corresponding heat sinks 60', 60" & 60'''.

In this particular arrangement, the three separate rows of cold sinks, heat exchangers and heat sinks work over different temperature ranges. All of the cold sinks 58', 58" & 58''' are about 20° Celsius. However, the three rows of heat exchangers 46', 46" & 46''' works at different temperature differentials ("ΔT"), and thus the three rows of heat sinks are at different temperatures.

In this embodiment, the bottom-most row of heat exchangers 46' exhibits a ΔT of approximately 20° Celsius leading to a bottom heat sink temperature 60' of about 40° Celsius; the middle row of heat exchangers 46" exhibits a

ΔT of approximately 30° Celsius leading to a middle heat sink temperature 60'' of about 50° Celsius and lastly the top most row of heat exchangers 46''' exhibits a ΔT of approximately 40° Celsius leading to a top heat sink temperature 60''' of about 60° Celsius.

The lower the ΔT of the relevant heat exchanger arrangement, the greater the COP rating. Thus, the bottom-most arrangement would have a COP of, for example, 5, whereas the middle arrangement would have a COP of, for example, 4 and the top-most arrangement would have a COP of, for example, 3. The advantage of this multi-staged pre-heating is that overall efficiency of the thermal cycle may be improved.

Since thermo-electric modules are used as the heat exchangers, it is possibly to regulate the ΔT of each row and therefore the COP of each row simply by control over the applied voltage. Thus, optimisation of each may be undertaken to maximise overall efficiency.

Fig. 7 shows detail of the heat transfer taking place at heat exchanger 58''', although it will be appreciated that this is simply exemplary of all heat exchangers within the system.

The total heat transferred will be a summation of two heat transfer mechanisms: the heat being forced by the heat exchanger 46''' towards heat sink 60''' and being shown diagrammatically as the arrow marked H_{PUMP} (with a corresponding temperature drop of the cold sink 58''' being shown diagrammatically as the arrow marked C_{PUMP}) and a natural conduction mechanism flowing through from condenser 34 through the cold sink 58''', through the heat exchanger 46''' and into the hot sink 60''', shown diagrammatically with the arrow marked $H_{COND.}$

Fig. 8 shows a slightly modified arrangement from that depicted above in Figs. 6 & 7. In this embodiment, features similar or identical to those of previous embodiments are prefixed with a “1”.

In this embodiment, tubes 138 from cooling tower (not shown) are disposed such that they connect directly with the cold sinks 58’, 58” & 58’”. The tubes 138 may also form an open loop cooling system i.e. they may be fed by a continuous and non-recycled feed of coolant water.

Figs. 9 and 10 show a thermal equaliser 66 disposed in the base of the condenser 34, extending from the base and being surrounded at its lower extent by a pool of liquid feed water 12. The thermal equaliser 66 comprises a plurality of fins 66a extending radially from a substantially cylindrical central core 66b.

The thermal equaliser 66 has a very large surface area and good thermal conductivity. The top half of the thermal equaliser 66 is in the wet vapour 32 and is heated by it. The lower half is in the pool of liquid feed water 12 and there is therefore a tendency for a transfer of thermal energy from the wet vapour 32 to the liquid feed water 12. The thermal equaliser 66 tends to further increase the temperature of the liquid feed water 12 before it leaves the condenser 34 for subsequent pre-heating. The thermal equaliser 66 will also assist in the condensation of wet vapour 32 providing it is below the dew point for the conditions prevailing in the condenser 34.

Fig. 11 shows a further embodiment of the present invention. Features similar or identical to those of previous embodiments are prefixed with a “2”. The apparatus allows for more straightforward adaption of prior art thermal systems to take advantage of the present invention.

Thermal cycle apparatus 210 includes a standard prior art design condenser 234, with feed water 212 being pressurised by pump 114 through pipe work 216 and into a boiler (not shown).

A wet vapour phase 232 exits a turbine (not shown) to be delivered to the condenser 234 by pipe work 236.

In this modification, a pipe branch 236a has been added downstream of the turbine (not shown) and upstream of the condenser 234. This branches off into secondary pipe work 236b, and the flow of wet vapour phase 232 into the secondary pipe work 236b may be controlled with valves 236c.

In use, the secondary pipe work 236b diverts wet vapour phase 232 to a secondary condenser module 234a, which has been attached effectively in a parallel arrangement to the original condenser 234.

The wet vapour phase 232 entering the secondary condenser module 234a is delivered adjacent a cold side 254 of a heat exchanger 250. The specific design and function of the heat exchanger is similar to that described above, including the use a heat pump 248 and of a thermoelectric module (TEM) 250 to carry out that function. Moreover, similar design cold sink and hot sink are used for the advantages described above.

The wet vapour phase 232 condenses in the secondary condenser module 234a to condensate feed water 212 before exiting and back into the thermal cycle. As with previous embodiments, heat transfers from the secondary condenser module 234a to the feed water 212 to provide pre-heating prior to boiler (not shown) entry.

This modular arrangement allows for prior art thermal cycle apparatus to be modified by providing the branch arrangement. Moreover, the condenser 234 and the secondary condenser module 234a may be run in parallel, and the proportion of wet vapour phase 232 entering each may be varied from 100% to the condenser 234 through to 100% to the secondary condenser module 234a, through any proportion in-between. This may allow for not only the fitting of the secondary condenser module 234a to a prior art design, but also for the thermal cycle to switch to 100% reliance on the condenser 234 should maintenance or replacement of the secondary condenser module 234a or its components be required. It will be appreciated by the skilled addressee that the secondary condenser module 234a is acting as a condenser in the thermal cycle.

The present invention may offer several advantages over prior art designs.

First, by reducing the energy rejected to the environment the overall power plant efficiency is increased.

Moreover, cooling towers depend on the ambient temperature for the removal of heat from the feed water pumped around the condenser loop. If the ambient temperature or relative humidity is high, the cooling tower efficiency is reduced. The leading in prior art solution is simply to larger, and therefore more inconvenient and costly, cooling towers.

By being able to force a portion of the condenser to a temperature below atmospheric, and towards the dew point of the working fluid, the requirement for a larger cooling tower in more humid / hotter environments may be mitigated.

Further, the use of smaller cooling towers for a desired operational temperature range. “Smaller” in this context could mean physically smaller structures, or the same structure could be used but with a lower flow rate: the pumping requirements in the circuit between the condenser and the cooling tower are reduced because a percentage of the thermal energy removed from the steam in the feed water loop is re-used to heat the feed water, rather than being expelled to ambient via the cooling tower. Typical flow rate in this loop is 14,000kg/sec and therefore even a small percentage saving is a significant number. The pump is powered by electrical means and therefore the reduction in energy input to the process (from coal) is the ΔPower / overall plant efficiency η. For a carbon capture plant this will typically be in the range 3X → 4X.

There is also the possibility of reversing the polarity of the thermoelectric modules such that they can be used to transfer heat in the opposite direction and therefore chill rather than heat the feed water. This could be used to provide a more rapid cool-down of parts of a power plant and therefore shorter service intervals. Heat from the plant may be picked up by the feed water and the feed water then cooled by the cooling tower. In this situation, the electric generator 30 would not be operational and electrical energy would need drawn from an alternative source such as a generator or from an electrical supply grid. The heat pump may also rely upon other forms of energy instead of electrical energy.

Modifications and improvements can be made to the embodiments herein before described without departing from the scope of the invention.

For example, the person skilled in the art will recognise that there are many different heat pumping technologies which have associated with them different COPs which in turn depend on the process by which the heat is pumped and also the temperature difference through which heat is pumped. The figures used here are illustrative of the process and do not limit the scope of the patent to the use of thermoelectric modules as the heat-pumping means.

Moreover, although described with water as a working fluid, the skilled addressee will appreciate that alternative working fluids may be used, for example, carbon dioxide, ammonia, etc.

Claims

1. A condenser for use in condensing a working fluid in a thermal cycle apparatus prior to its re-entry into a working fluid line and onto a boiler, the condenser including a heat exchanger with a cold surface portion and a hot surface portion, the cold surface portion being disposed within a condensing chamber of the condenser, and the hot surface portion being deployed such that, in use, it transfers heat to condensed working fluid in a working fluid line prior to said condensed working fluid entering a boiler.

2. A condenser according to claim 1 wherein the heat exchanger includes a heat pump.

3. A condenser according to claims 1 or 2 wherein the heat exchanger includes a thermo-electric module.

4. A condenser according to any preceding claim wherein condensate channels are provided on or adjacent to said cold surface portion.

5. A condenser according to any preceding claim wherein the part of the working fluid line to which the hot surface portion is adjacent is located upstream of a boiler.

6. A condenser according to any preceding claim wherein the part of the working fluid line to which the hot surface portion is adjacent is located upstream of a pressurisation pump used to pressurise working fluid prior to its entry into a boiler.

7. A condenser according to any preceding claim wherein the cold surface portion includes a cold sink.

8. A condenser according to claim 7 wherein the cold sink comprises one or more cold sink fins.

9. A condenser according to claim 8 wherein the cold sink fins taper from a first thickness adjacent the cold surface to a second thickness at a distal end, the first thickness being greater than the second thickness.

10. A condenser according to any preceding claim wherein the hot surface portion includes a heat sink.

11. A condenser according to claim 10 wherein the heat sink comprises one or more heat sink fins.

12. A condenser according to claim 11 wherein the heat sink fins taper from a first thickness adjacent the hot side to a second thickness at a distal end, the first thickness being greater than the second thickness.

13. A condenser according to any of claims 8 to 10 wherein the cold sink fin(s) comprise a material that results in a temperature gradient forming from a tip of the cold sink fin(s) to a root of the cold sink fin(s) when in use.

14. A condenser according to claim 13 wherein said temperature gradient is from a higher temperature at the fin tip to a lower temperature at the fin root.

15. A condenser according to any of claims 8, 9 or 10, 13 or 14 (when dependent on claim 8 or 9) wherein a portion of said cold sink fin(s) is hydrophobic.

16. A condenser according to any of claims 11 or 12 wherein the heat sink fin(s) have a portion which is hydrophobic.

17. A condenser according to any of claims 15 or 16 wherein the hydrophobic portion of said fins is limited to less than the total surface of said fin(s).

18. A condenser according to any of claims 17 wherein said hydrophobic portion is disposed at the fin root and the non-hydrophobic portion toward the fin tip.

19. A condenser according to any of claims 9 or 12 wherein the tapering of said fins is generally hyperbolic.

20. A condenser according to any of claims 9 or 12 wherein the tapering of said fins is generally exponential.

21. A condenser according to any of claims 8, 9 or 10, 13 or 14 (when dependent on claim 8 or 9) wherein the cold sink fins are orientated downwardly in the condenser.

22. A thermal power station employing the condenser of any preceding claim.

23. A method of operating a thermal cycle involving a working fluid comprising the steps of pressurising the working fluid in a substantially

liquid phase, applying heat to the working fluid to transfer it to a substantially gaseous phase, extracting mechanical energy from the gaseous phase, condensing the working fluid to a liquid phase and therefore extracting heat, using said extracted heat to preheat the working fluid prior to applying further heat to the working fluid.

24. A method of operating a thermal cycle according to claim 23 wherein said extracted heat is applied prior to the working fluid being re-pressurised in a liquid phase.

25. A thermal power station employing the method of any of claims 23 or 24.

26. A method of modifying a thermal power station including the steps of fitting a condenser according to any of claims 1 to 21.

27. A temperature sink comprising a portion of hydrophobic material.

28. A temperature sink according to claim 27 wherein the temperature sink comprises one or more fins, the fins tapering from a first thickness adjacent a one side of the temperature sink to a second thickness at a distal end, the first thickness being greater than the second thickness.

29. A temperature sink according to claim 28 wherein the hydrophobic portion of said fins is limited to less than the total surface of said fin(s).

30. A temperature sink according to claims 28 or 29 wherein said hydrophobic portion is disposed at the fin root and the non-hydrophobic portion toward the fin tip.

31. A temperature sink according to claims 28 to 30 wherein the tapering of said fins is generally hyperbolic.

32. A temperature sink according to claims 28 to 30 wherein the tapering of said fins is generally exponential.

33. A heat exchanger including at least one temperature sink according to claims 27 to 32.

34. A heat pump including at least one temperature sink according to claims 27 to 32.

35. A thermal power station employing at least one temperature sink according to claims 27 to 32.

36. A thermal power station employing at least one heat exchanger according to claim 33.

37. A thermal power station employing at least one heat pump according to claim 34.

38. A condenser suitable for use in a thermal cycle including at least one thermo-electric module.

39. Apparatus for operating a thermal cycle including at least one condenser according to claim 38.

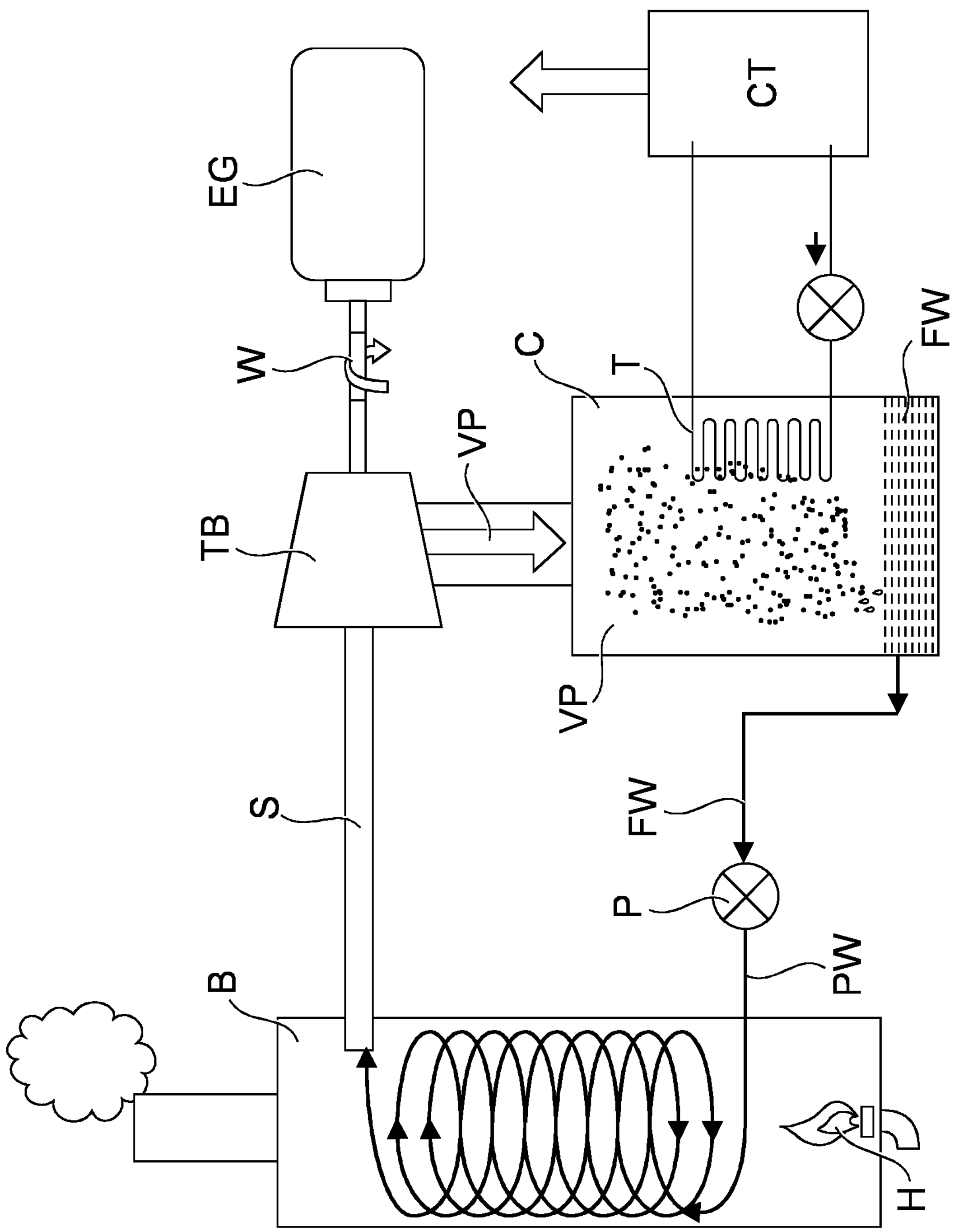

Fig. 1

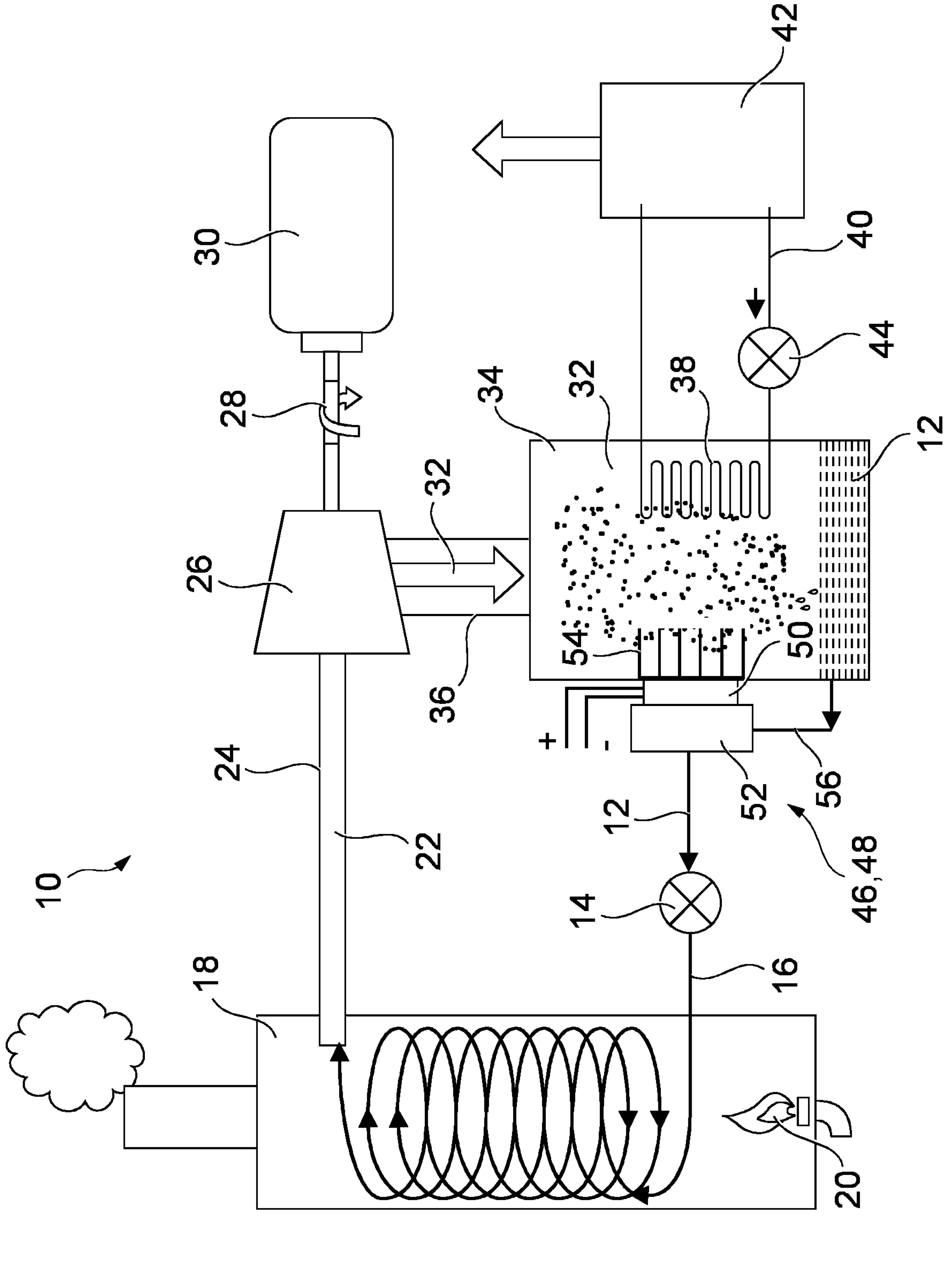
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Fig. 2

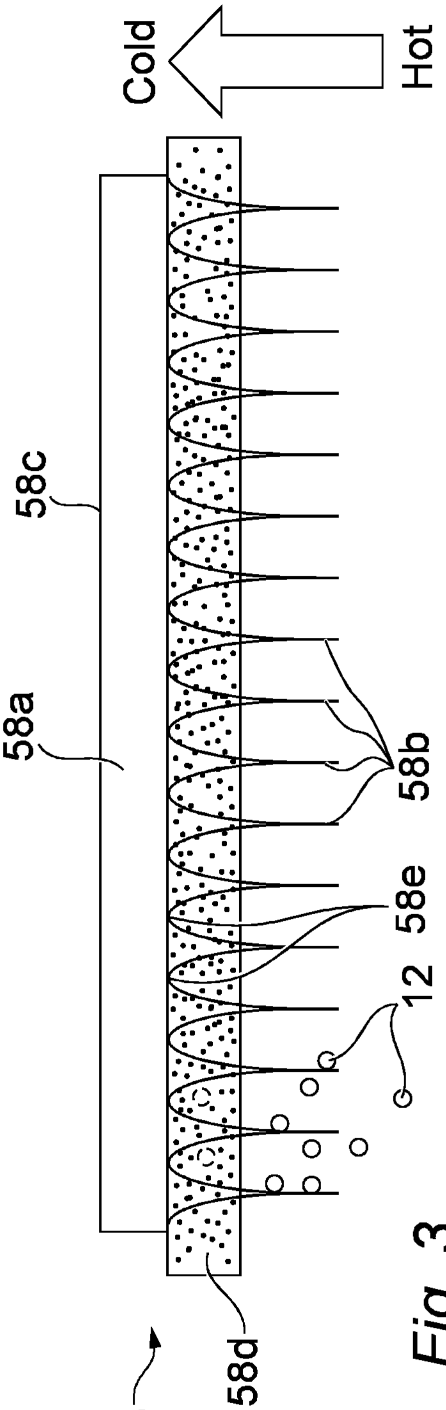


Fig. 3

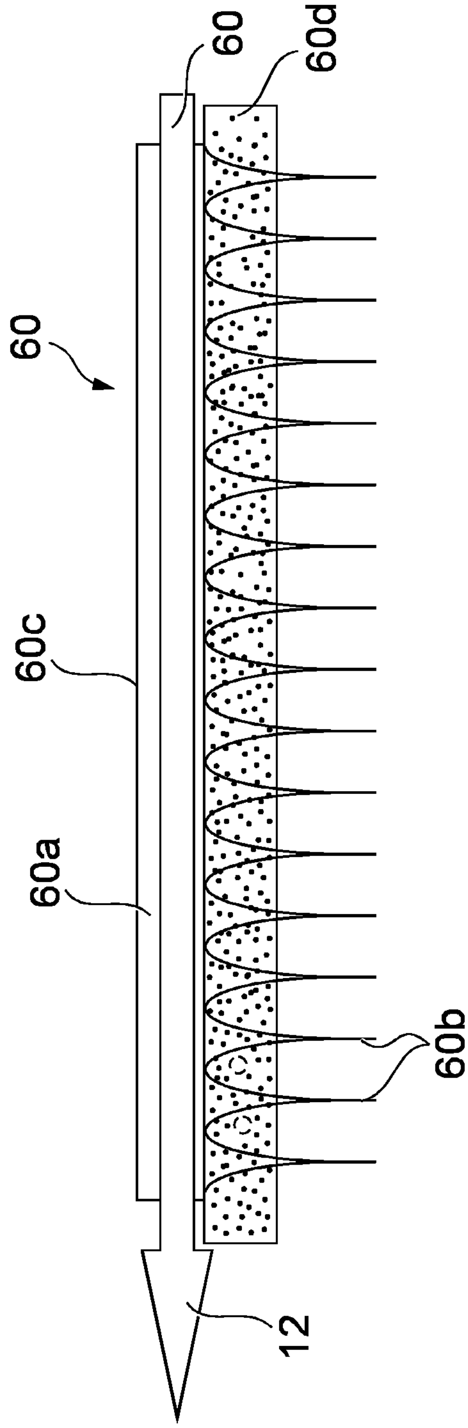


Fig. 4

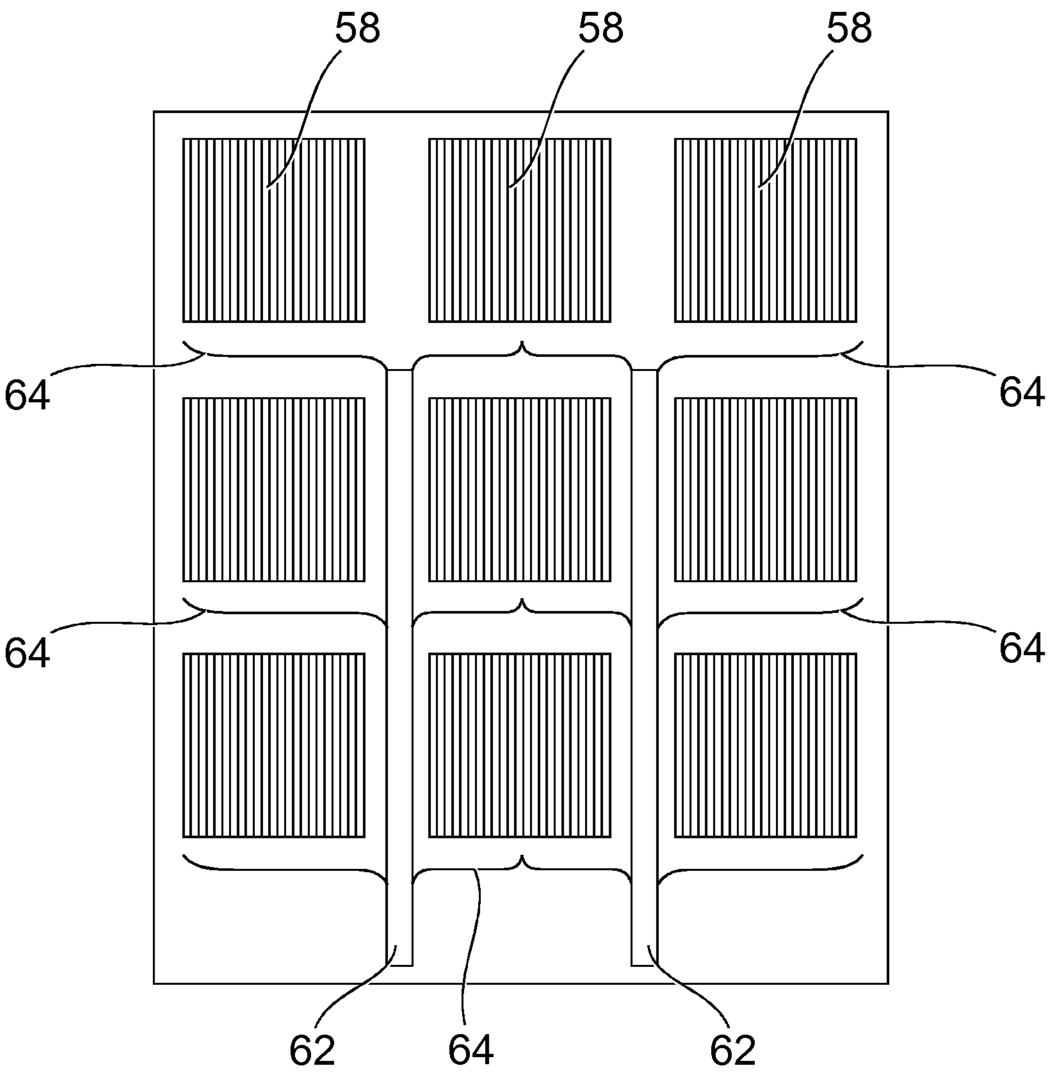
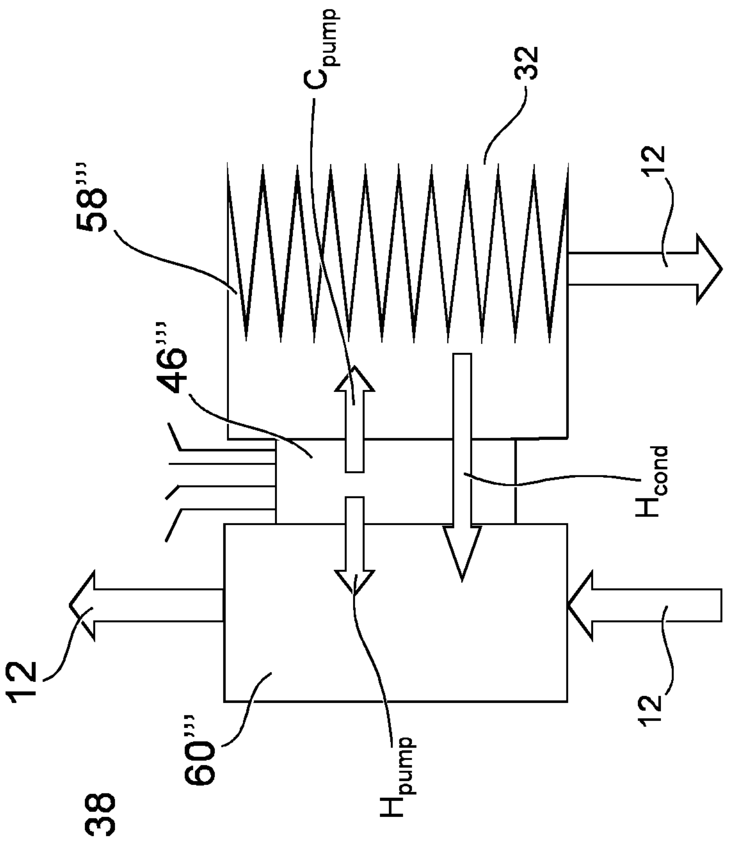
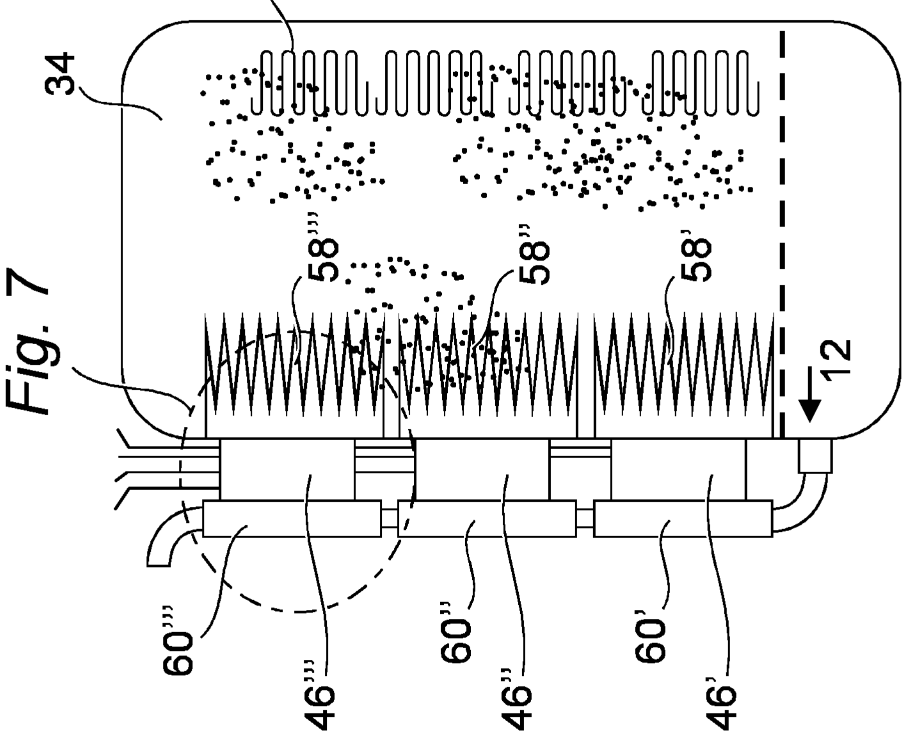


Fig. 5



C_{pump}
32
58'''
46'''
12
H_{cond}
12
60'''
H_{pump}
12

Fig. 7



34
38
Fig. 7
58'''
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Fig. 6

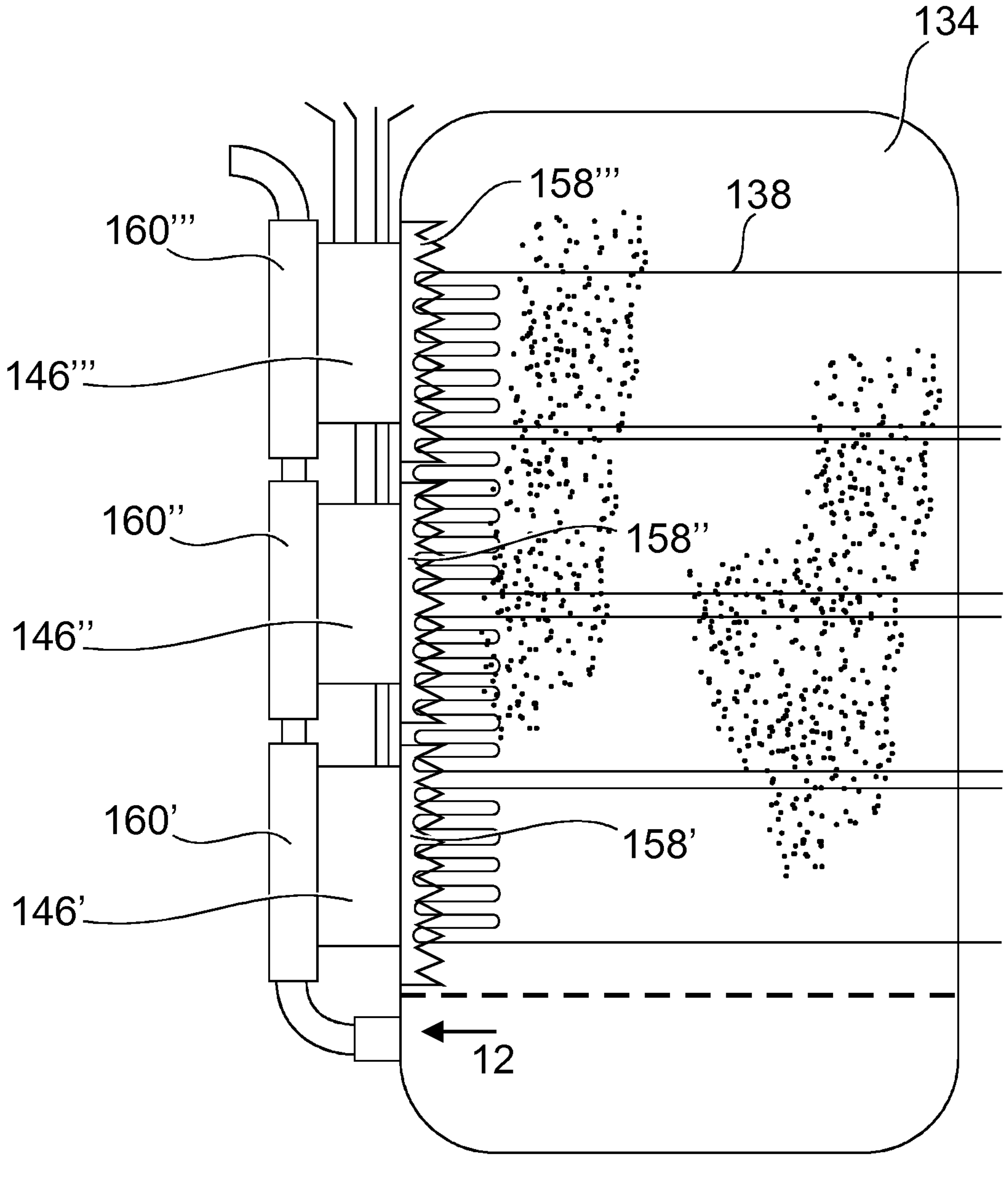

Fig. 8

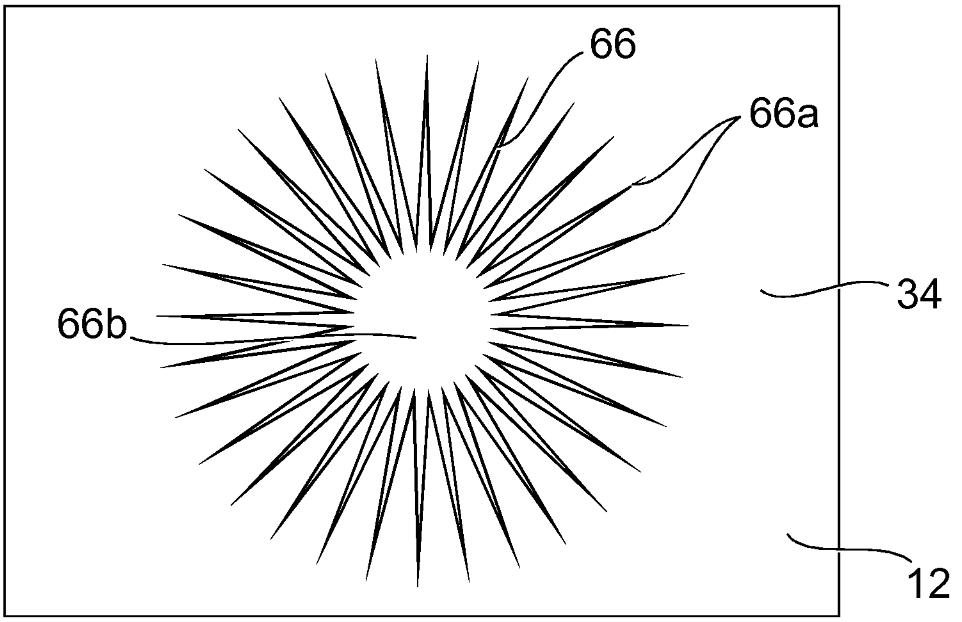

Fig. 9

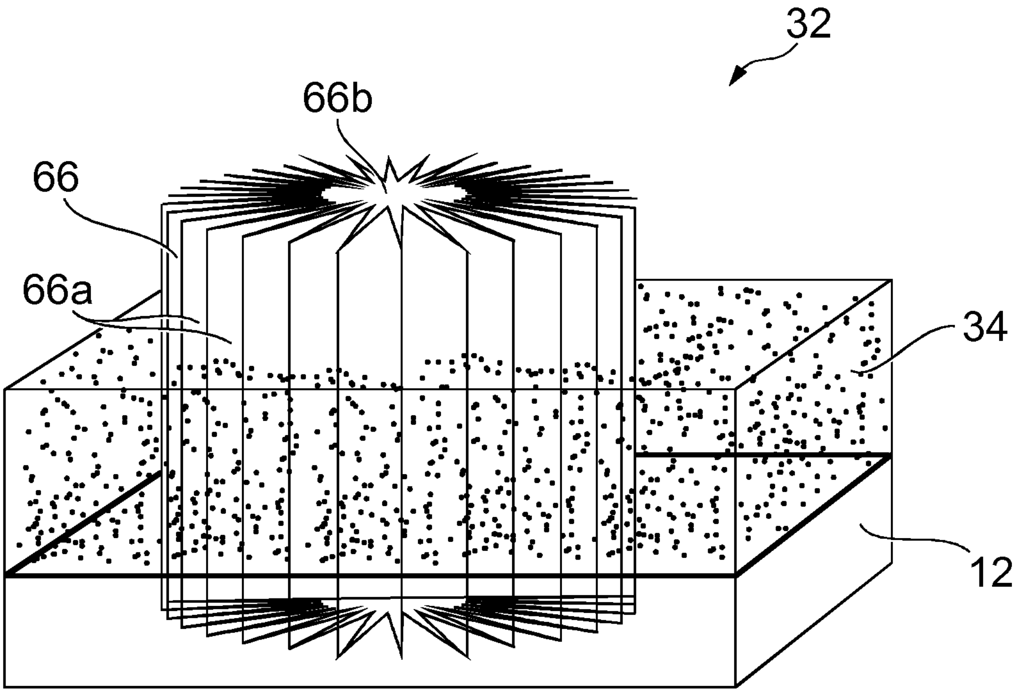

Fig. 10

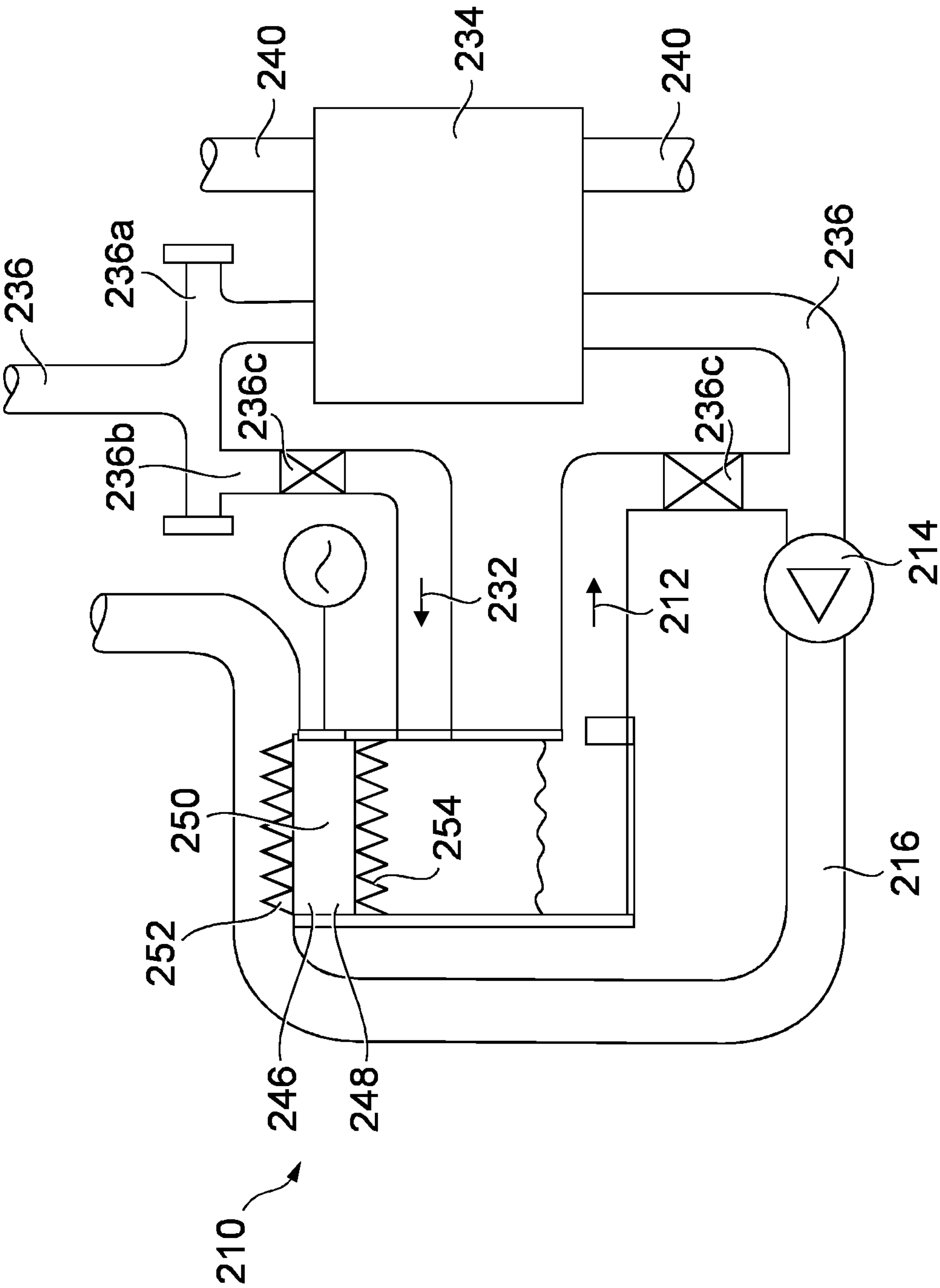
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Fig. 11