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(54) Thermo electric energy storage system

(57) Steam bled from a turbine 14 of a power plant 12 is used to heat, through a split stream heat exchanger 20, an energy storage system 10 with a low temperature storage 26, an intermediate thermal storage 24 and a high temperature storage 22. A further split stream heat exchanger 30 is used to efficiently heat the working fluid of a heat engine cycle 29 for generating electricity. During times when the load on the power plant 12 is low, surplus heat from the turbine 14 may be stored in the energy storage system 10. The stored heat may be used to generate electricity for compensating load peaks on the power plant 12. For example, the heat engine cycle 29 may drive an additional generator 40.

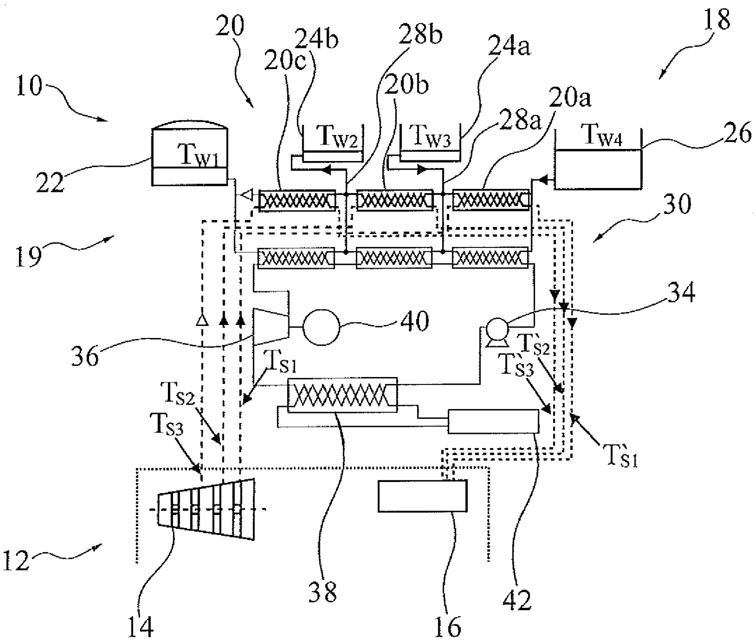

Fig. 2

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Description

FIELD OF THE INVENTION

[0001] The invention relates to the field of storing energy. In particular, the invention relates to an energy storage system for a power plant, a use of an energy storage system, a power plant with an energy storage system and a method for storing excess thermal energy of a power plant.

BACKGROUND OF THE INVENTION

[0002] Nearly all types of base load power plants (powered by coal, fuel or nuclear energy) use a turbine for converting thermal energy of hot steam into mechanical energy for driving an electrical generator.

[0003] To satisfy the varying demand in electrical energy, the output of the generator may have to be varied. Many concepts exist which allow base load power plants to vary their output power while keeping the primary energy input of the power plant constant or not significantly changed. In particular, for steam power plants, steam may be bled from one or several steam feeds from the turbine in order to reduce the power output of the power plant when needed. Bleeding steam may be understood as extracting surplus steam from the turbine to lower the power output of the turbine.

[0004] The heat content of the bled steam may be stored either directly or may be stored indirectly into a heat storage material, for example during a period of reduced power output. During periods of required higher power output of the power plant, the stored heat may be typically reused to preheat the water feed to a boiler, thus increasing the turbine power output in the case the boiler power remains constant.

[0005] It is also possible to use the stored heat in run time-delayed bottoming heat engine cycles. It is known to use thermodynamic cycles or processes for efficiently converting heat into work and vice versa. For example, in a heat pump cycle a working fluid may be compressed by a compressor and the generated heat may be transferred via a heat exchanger from the working fluid to a thermal storage medium. Vice versa, in a heat engine cycle heat may be transferred to a working fluid and the heated working fluid may be expanded in a turbine, thus generating mechanical work.

[0006] For example, US 4,428,190 shows a power plant including a steam boiler that delivers a rated amount of high-pressure steam at rated temperature and pressure to a steam turbine. A main generator, driven by the steam turbine, furnishes electricity to a variable load. When the load decreases below a rated value, the boiler operation is maintained, but steam exhausted from the turbine is diverted to a heat store large enough to accumulate heat during the time that the power plant operates at less than rated load. A waste heat converter, having its own generator, is responsive to the heat stored in the heat store, and can be operated selectively to furnish electricity to the load to supplement the output of the power plant, when the load increases above a rated value.

[0007] However, when using real thermodynamic cycles, the transfer of heat over large temperature differences enabling or enabled by mechanical work is thermodynamically only partially reversible. Thus, temperature differences in a heat exchanger between the working fluid and the storage medium have to be minimized, to maximize the thermal efficiency (or thermal factor) of the thermodynamic processes. When as much work as possible has to be recovered from the thermal storage or as much heat as possible has to be stored in the thermal storage, the maximal temperature difference between the working fluid and the storage medium may have to be minimized.

DESCRIPTION OF THE INVENTION

[0008] It is an object of the invention to increase the thermal efficiency of a power plant.

[0009] This objective is achieved by the subject-matter of the independent claims. Further exemplary embodiments are evident from the dependent claims and the following description.

[0010] An aspect of the invention relates to an energy storage system for a power plant.

[0011] According to an embodiment of the invention, the energy storage system comprises a thermal storage circuit containing a thermal storage medium, the thermal storage circuit comprising a high temperature storage, one or more intermediate temperature storage and a low temperature storage connected together, for example by means of at least one stream splitter (the number of stream splitters is equal to the number of intermediate temperature storages); a steam heat exchanger arrangement for heating a thermal storage medium flowing from the low temperature storage to the high temperature storage (where either all or part of the thermal storage medium flows from the low temperature storage to the high temperature storage), wherein the thermal storage medium is heated with steam bled from a turbine of the power plant; a working fluid heat exchanger arrangement for heating a working fluid, the working fluid flowing through a low temperature working fluid heat exchanger and a high temperature working fluid heat exchanger, for example, interconnected by a stream splitter.

[0012] The thermal storage medium flows from the high temperature storage through the high temperature working fluid heat exchanger and either all or part of the thermal storage medium flows through the low temperature working fluid heat exchanger to the low temperature storage.

[0013] The intermediate temperature storage is connected to the thermal storage circuit between the high temperature working fluid heat exchanger and the low temperature working fluid heat exchanger, for example by the stream splitter; wherein the energy storage system is adapted to generate different flow rates of thermal storage medium in the high temperature working fluid heat exchanger and the low temperature working fluid heat exchanger by injecting thermal storage medium from the intermediate temperature storage into a flow of thermal storage medium between the high temperature working fluid heat exchanger and the low temperature working fluid heat exchanger or by extracting thermal storage medium from a flow of thermal storage medium between the high temperature working fluid heat exchanger and the low temperature working fluid heat exchanger and redirect it to the intermediate temperature storage. The injecting and removing of thermal storage medium may be done by means of the stream splitter that is a valve or piping and valve arrangement enabling the generation of different flow rates of storage medium to and from the intermediate temperature storage.

[0014] It has to be understood that the terms "high", "intermediate" and "low" may only indicate that the thermal storage medium in the high temperature storage is hotter than the thermal storage medium in the intermediate temperature storage, which is hotter than the thermal storage medium in the low temperature storage.

[0015] The skilled person will be aware that the heat exchangers, in particular the steam heat exchanger and the working fluid heat exchanger, may be the same piece of equipment or may comprise the same pieces of equipment such as the thermal storage circuit. Since the operations of the steam heat exchanger and the working fluid heat exchanger may be decoupled, this may save equipment. Furthermore, the heat exchangers, in particular the steam heat exchanger and the working fluid heat exchanger, may be arrangements of single heat exchangers i.e. heat exchangers without internal stream splitters, for example a series or a network of coupled single heat exchangers.

[0016] The injecting and removing of thermal storage medium may be performed by means of the stream splitter. The heat exchanger arrangement may have an integral stream splitter adapted to divide or to join a flow of thermal storage medium from or to the intermediate temperature storage.

[0017] It is a gist of the invention to use steam bled from a turbine to heat a thermal storage with a low temperature storage, an intermediate thermal storage and a high temperature storage and to use a split stream heat exchanger network to efficiently heat the working fluid of a heat engine cycle for generating electricity. During times, when the load on the power plant is low, surplus heat from the turbine may be stored in the thermal storage. The stored heat may be used to generate electricity for compensating load peaks on the power plant. For example, the heat engine cycle may drive an additional generator. Further, due to the energy storage system, a power plant with lower maximal main generator output may satisfy a power demand with load peaks.

[0018] Instead of one single intermediate temperature storage, two or more intermediate temperature storages of different temperatures may be used. In this way, a fitting of temperature profiles between the steam, the thermal storage medium and the working fluid may be enhanced.

[0019] According to an embodiment of the invention, the thermal storage medium flows from the low temperature storage through a low temperature steam heat exchanger and a high temperature steam heat exchanger to the high temperature storage; wherein in the low temperature steam heat exchanger steam of a low temperature bled from the turbine is used for heating and in the high temperature heat exchanger steam of a higher temperature bled from the turbine is used for heating; wherein the intermediate temperature storage is connected via a stream splitter to the thermal storage circuit between the high temperature steam heat exchanger and the low temperature steam heat exchanger; wherein the energy storage system is adapted for generating different flow rates in the low temperature steam heat exchanger and the high temperature heat exchanger by injecting or extracting via the stream splitter thermal storage medium from or into the intermediate temperature storage into or from the thermal storage circuit or by extracting or injecting via the stream splitter thermal storage medium from or into the thermal storage circuit into or from the intermediate temperature storage.

[0020] It is a further gist of the invention to use a split stream heat exchanger network to store heat of bled steam from a turbine of a power plant in a thermal storage. To minimize the temperature differences in the heat exchanger, streams of bled stream of different temperatures from the turbine may be input into the heat exchanger network at different positions.

[0021] With a heat engine thermodynamic cycle fitting as close as possible the temperature profile of the heat source represented by the bled steam, as well as a thermal storage system having itself a temperature profile as close as possible to the working fluid temperature profile in the gas heater (heat exchanger), the efficiency of the system may be increased even more.

[0022] In order to reach high indirect storage efficiency, a high adequacy may be needed between the temperature profile of the bled steam (at one or more stages), the temperature profile of the heat storage, and the temperature profile of the working fluid of the bottoming cycle.

[0023] For power plants operating at least one steam heat engine cycle, with the present energy storage system the power rating of the plant during defined periods of time may be energetically efficiently reduced or increased, while the primary energy input of the power plant may be kept constant or not significantly changed. This relates to the specific arrangement of the cycle in order to achieve a high ratio between the power increase and the power decrease of the power plant. Compared to a power plant running at its nominal power rating, the integrated power reduction over the period of reduced power output (the charging period) may be considered as indirectly stored electric energy, while the integrated power increase over the period of increased power output (the discharging period) may be considered as stored energy delivered back by the storage. The ratio between the integrated power increase and the integrated power decrease may be seen as an effective electricity storage efficiency.

[0024] A close adequacy in terms of a temperature profile of the hot/cold source with the working fluid temperature profile of a heat pump/heat engine cycle may be needed in order to reach a high ratio of stored energy. This may be applied to power plants operating steam turbines in order to optimize the indirect electricity storage efficiency. Due to its efficiency (ratio of more than 70%), the present system acts as efficient indirect electricity storage system where the value of the storage efficiency is high, but also allows base-load power plants to be more flexible and to quickly react on load variations, to increase their maneuverability, or to take benefit of low-cost/high-cost electricity price variations.

[0025] In the present invention, the working fluid is preferably carbon dioxide (CO_2), or may comprise ammonia (NH_3) and/or an organic fluid (such as methane, propane or butane) and/or a refrigerant fluid (such as R 134a (1,1,1,2-Tetrafluoroethane), R245 fa (1,1,1,3,3-Pentafluoropropane)).

[0026] In the present invention, the thermal storage medium is preferably water, water with additives and/or a sensible heat storage medium.

[0027] In the present invention, the working fluid is heated fully in a supercritical phase in the working fluid heat exchanger arrangement (or depending on the condenser temperature, the working fluid may be heated from a liquid state at a pressure above the critical pressure up to a supercritical state).

[0028] In the present invention, the heat engine cycle may be a transcritical cycle. A transcritical cycle may be a cycle in which the high pressure part is entirely in a supercritical phase or goes from a liquid state at a pressure above the critical pressure to a supercritical state, and in which the low pressure part of the cycle is in the subcritical phase and goes from the gas phase to the liquid phase.

[0029] In order to enhance the efficiency of the system, the mechanical work produced by a secondary transcritical CO_2 heat engine (time-delayed bottoming cycle) may be maximized during the period of increased power demand of the power plant with respect to the mechanical work reduction of the steam heat engine during the period of decreased power rating of the power plant, i.e. the effective electrical storage efficiency. Therefore, for a heat engine thermodynamic cycle operating at given conditions, the thermal efficiency (work produced by the heat engine divided by the heat provided to the heat engine) may be maximized if the temperature profile of the hot source providing the heat is as close as possible to the temperature profile of the gas heater or boiler of the heat engine thermodynamic cycle.

[0030] A further aspect of the invention relates to a use of an energy storage system as described in the above and in the following in a large scale power plant. A large scale power plant may be a power plant with an output of more than 1 MW.

[0031] A further aspect of the invention relates to a large scale power plant with an energy storage system as described in the above and in the following.

[0032] A further aspect of the invention relates to a method for storing excess thermal energy of a power plant.

[0033] According to an embodiment of the invention, the method comprises a storage cycle for storing energy in thermal energy storage and a working cycle for generating electricity from the stored energy. The energy storage cycle or energy storage process may be performed during low power demand. The working cycle, heat engine cycle or energy generation process may be performed during high power demand or peak loads.

[0034] The method may comprises the steps of: in the storage cycle, bleeding steam from a turbine of the power plant; in the storage cycle, heating a thermal storage medium with the steam, in the working cycle, heating a working fluid with a flow of thermal storage medium coming with a first flowrate from a high temperature storage; in the working cycle, generating a flow of thermal storage medium of a second flowrate flowing to a low temperature storage by adding thermal storage medium from an intermediate temperature storage to the flow or by removing thermal storage medium from the flow into a intermediate temperature storage; in the working cycle, heating the working fluid with the flow of thermal storage medium of the second flow rate.

[0035] Wherein the step of heating a thermal storage medium with the steam comprises; in the working cycle with a flow of thermal storage medium coming with a first flowrate from a low temperature storage; in the storage cycle, generating a flow of thermal storage medium of a second flowrate flowing to a high temperature storage by adding thermal storage medium from an intermediate temperature storage to the flow or by removing thermal storage medium from the flow into an intermediate temperature storage; in the storage cycle, heating the thermal storage medium of the second flow rate with the steam.

[0036] It has to be understood that features of the method as described in the above and in the following may be features of the system as described in the above and in the following.

[0037] If technically possible but not explicitly mentioned, also combinations of embodiments of the invention described in the above and in the following may be embodiments of the method and the system.

[0038] These and other aspects of the invention will be apparent from and elucidated with reference to the embodiments described hereinafter.

BRIEF DESCRIPTION OF THE DRAWINGS

[0039] The subject matter of the invention will be explained in more detail in the following text with reference to exemplary embodiments which are illustrated in the attached drawings.

Fig.1a schematically shows aspects of the storage cycle of an energy storage system according to an embodiment of the invention.

Fig.1b schematically shows aspects of the working cycle of an energy storage system according to an embodiment of the invention.

Fig. 2 schematically shows an energy storage system having a storage cycle according to an embodiment of the invention.

Fig. 3 schematically shows an energy storage system having a working cycle according to an embodiment of the invention.

Fig. 4 shows a diagram depicting temperatures and state transitions of the thermal storage medium and the working fluid according to an embodiment of the invention.

[0040] The reference symbols used in the drawings, and their meanings, are listed in summary form in the list of reference symbols. In principle, identical parts are provided with the same reference symbols in the figures.

DETAILED DESCRIPTION OF EXEMPLARY EMBODIMENTS

[0041] Fig.1a schematically shows an energy storage system 10 for a power plant 12. The power plant 12 comprises a turbine 14 powered by steam from a boiler (not shown) that, for example, may be heated by conventional energy sources like coal, fuel, nuclear material or by renewable energy sources like sun energy. The power plant 12 comprises further a condenser 16, in which the steam cooled by the turbine is cooled to (or close to) ambient temperature.

[0042] Fig. 1a shows the process of steam bleeding and of storing the steam heat content into the energy storage system 10, for example for reducing the output of the power plant 12. During the storage cycle 18 of the energy storage system 10, steam of different temperatures is bled from different stages of the turbine 14, flows through a steam heat exchanger arrangement 20 and to the condenser 16.

[0043] For storing the heat, the energy storage system 10 comprises a thermal storage circuit 19 with a high temperature storage 22, an intermediate temperature storage 24 and a low temperature storage 26 connected together over a stream splitter 28 in the heat exchanger arrangement 20. The thermal storage circuit 19 and in particular the storages 22, 24, 26 (that may be tanks) contain water (that may contains additives) as thermal storage medium. It is possible that the storages contain another sensible heat storage medium. The heat energy from the steam (mostly latent heat) is stored in the form of sensible heat in the water. In the steam heat exchanger arrangement 20, the thermal storage medium flowing from the low temperature storage 26 to the high temperature storage 22 is heated with steam bled from the turbine 14. To generate different flowrates in the heat exchanger arrangement 20, with the stream splitter 28 the flow inside the heat exchanger arrangement 20 is joined with a flow of thermal storage medium from the intermediate temperature storage 24 or split into a flow to the intermediate temperature storage 24. With this stream splitting the amounts of differently heated thermal storage medium are generated that are used by the working cycle 29 shown in Fig. 1b. The stream splitting has the effect that the temperature differences in the heat exchanger 20 between the thermal storage medium and the steam may be minimized.

[0044] Fig. 1b shows the recovery of the heat stored in the storage system 10 to power a bottoming transcritical CO_2 (carbone dioxide) heat engine or working cycle 29, for example to increase to overall power output of the power plant.

[0045] The system 10 has a working fluid heat exchanger arrangement 30 for heating CO_2 as a working fluid. The high temperature storage 22, the intermediate temperature storage 24 and the low temperature storage 26 are connected together over a stream splitter 32 in the heat exchanger arrangement 30. For heating the working fluid, thermal storage medium flows from the high temperature storage 22 through the heat exchanger 30 to the low temperature storage 26. To generate different flowrates inside the heat exchanger arrangement 30, with the stream splitter 32 the flow inside the heat exchanger arrangement 30 is joined with a flow of thermal storage medium from the intermediate temperature storage 24 or split into a flow to the intermediate temperature storage 24. This stream splitting has the effect that the temperature differences in the heat exchanger 30 between the thermal storage medium and the working fluid may be minimized.

[0046] The heat exchanger arrangement 30 is part of a working cycle 29 which further comprises a pump 34 for compressing the working fluid before the heating in the heat exchanger arrangement 30, a turbine 36 for generating mechanical energy by expanding the working fluid after the heating and a condenser 38 for cooling the working fluid between the turbine 36 and the pump 34. The turbine 36 drives an electrical generator 40 for generating additional electrical energy for the power plant 12. The condenser 38 may be a further heat exchanger in which water from a river may be used as cooling medium 42 or may be an (forced) air cooler.

[0047] To achieve the desired functionality and efficiencies of more than 70% of indirect electricity storage, the system 10 is based on the combination of steam bleeding at several stages of the steam turbine 14, the low and high temperature storages 22, 26, an intermediate sensible heat storage 24 using the concept of stream splitting 28, 32, and a time delayed bottoming transcritical CO_2 working cycle 29 for on-demand electricity generation from the storages 22, 24, 26.

[0048] The concept of stream splitting is implemented by the heat storage system 10 in which the temperature profile of the thermal storage medium is controlled by stream splitters 28, 30 in the heat exchangers 20, 30 connected to an intermediate sensible heat storage 24.

[0049] The temperature profile of the thermal storage medium in the storage circuit 19 is found between the temperature profile of the bled steam (heat source) and the temperature profile of the working fluid in the gas heater (heat exchanger arrangement 30) of the transcritical CO_2 heat engine cycle 29. A good fit between the temperature profiles of the thermal storage medium in the storage circuit 19 and of the CO_2 in the gas heater 30 allows in the end the heat source temperature profile to be effectively as close as possible to the gas heater temperature profile. This is achieved by stream splitting of the thermal storage medium.

[0050] Due to the use of stream splitting the loss of potential generated work (and not the loss of waste heat) during heating of the thermal storage medium and during heating of the working fluid may be minimized.

[0051] To achieve temperature profiles adapted to the transcritical CO_2 working cycle 29, existing 120° C steam of the turbine 14 or lower stages (for example steam of 60 °C) may be used for heating the thermal storage medium in the storage circuit 19, since the transcritical CO_2 working cycle 29 has an overall targeted working temperature below 150 °C. For example, in the storage cycle the bled steam may be cooled in the heat exchanger 20 to 45° C and water of 40 °C in the low temperature storage 26 may be heated to 120 °C. In the working cycle, the cooled CO_2 of 20 °C may be compressed to a temperature of 35 °C and may be heated in the supercritical phase in the heat exchanger 30 to 120 °C.

[0052] Fig. 2 and 3 schematically show more details of the energy storage system 10. The system comprises a storage cycle 18 for storing energy in the thermal energy storage 22, 24a, 24b, 26 and a working cycle 29 for generating electricity from the stored energy. The storage cycle 18 will be explained with reference to Fig. 2, the working cycle 29 with reference to Fig. 3.

[0053] Fig. 2 shows the charging of the storage system 10. During the period of reduction of the electric output of the power plant 12 (storage charging), steam is extracted at several pressure/temperature stages from the steam turbine 14 in order to exchange heat to the intermediate sensible heat to the thermal storage circuit 19 based on the concept of stream splitting of the thermal storage fluid (for example water). During this period, the instantaneous power generation output of the steam turbine 14 is reduced by the mechanical work that could have been produced by the steam that is extracted instead of going through all the downstream turbine stages.

[0054] The output power of the power plant 12 is reduced when steam is bled to heat the sensible heat storage system 10. The concept of split stream is implemented with three heat exchangers 20a, 20b, 20c and four tanks 22, 24a, 24b, 26 for the thermal storage liquid.

[0055] In particular, the steam heat exchanger arrangement 20 comprises a low temperature heat exchanger 20a, an intermediate heat exchanger 20b and a high temperature heat exchanger 20c connected together in series. Some steam is bled at the three last stages of the steam turbine 14. The extracted steam is directed to the series of heat exchangers (water heaters) 20a, 20b, 20c in which water is heated to the storage temperatures (in this example the heat storage 10 uses water as the thermal storage fluid or thermal storage medium).

[0056] During charging of the system 10, thermal storage medium flows from the low temperature storage 26 through the heat exchangers 20a, 20b, 20c to the high temperature storage. Each of the intermediate temperature storages 24a, 24b is connected with a stream splitter 28a, 28b to the thermal storage circuit 19 between the high temperature heat exchanger 20c and the low temperature heat exchanger 20a. In particular, the first intermediate temperature storage 24a is interconnected between the heat exchangers 20a and 20b with a first stream splitter 28a and the second intermediate temperature storage 24b is interconnected between the heat exchangers 20b and 20c with a second stream splitter 28b.

[0057] In the low temperature heat exchanger 20a steam of a low temperature T_{S1} bled from the turbine 14 at a first stage is heating the thermal storage medium from T_{W4} to T_{W3}. In the intermediate heat exchanger 20b steam of a temperature T_{S2} bled from the turbine 14 at a second stage is heating the thermal storage medium from the temperature T_{W3} to T_{W2}. In the high temperature heat exchanger steam 20c steam of a temperature T_{S3} bled from the turbine at a third stage is heating the thermal storage medium from T_{W2} to T_{W1}. Examples of the different temperatures will be given with reference to Fig. 4.

[0058] The energy storage system 10 generates different flowrates in the different heat exchangers 20a, 20b, 20c by injecting or removing thermal storage medium from or injected to the first intermediate temperature storage 24a into or from the thermal storage circuit 19 over the stream splitter 28a and by injecting or removing thermal storage medium from the thermal storage circuit 19 into the second intermediate temperature storage 24b over the stream splitter 28b.

[0059] During the energy storage cycle, the system 10 is bleeding steam from the turbine 14 of the power plant 12 and heats a thermal storage medium flowing through the thermal storage circuit 19 with the steam. In particular, the system 10 generates a flow of thermal storage medium of low temperature T_{W4} (for example with a pump) and the flow inside the heat exchanger 20a coming from the low temperature storage 26 is heated with steam of the temperature T_{S1}. A flow of another higher (or lower) flowrate is generated in the heat exchanger 20b by adding (or removing) thermal storage medium from (or injected into) the first intermediate storage 24a (for example with a further pump). The flow of the higher (or lower) flowrate is heated in the intermediate heat exchanger 20b to T_{W2} with the steam of temperature T_{S2}. After the heat exchanger 20b, a flow of a lower (or higher) flowrate is generated in the heat exchanger 20c by removing (or adding) thermal storage medium from (or to) the flow leaving the heat exchanger 20b into (or from) the second intermediate temperature storage 24b. The flow through the heat exchanger 20c is heated to T_{W1} with steam of temperature T_{S3}.

[0060] Fig. 3 shows the system 10 during discharging of the storage. During the period of increased power demand, no steam is extracted from the steam heat engine cycle at the turbine 14 and the sensible heat stored in heated water is used to power a secondary transcritical CO_2 heat engine cycle (i.e. discharge or working cycle) 29 running in parallel to the steam heat engine turbine 14, thus increasing the instantaneous power generation output of the power plant 12.

[0061] The power output of power plant 12 is increased when the flows of the sensible heat storage are reversed to heat the heat exchanger (gas heater) 30a, 30b, 30c of the bottoming transcritical CO_2 heat engine cycle 29. The concept of split stream is implemented with the heat exchanger arrangement 30 being divided in three parts 30a, 30b, 30c with four connections on the side of the heat storage liquid.

[0062] In particular, the working fluid heat exchanger arrangement 30 comprises a low temperature heat exchanger 30a, an intermediate temperature heat exchanger 30b and a high temperature heat exchanger 30c. The heat exchangers 30a, 30b, 30c are connected together such that the compressed working fluid coming from the pump 34 flows first through the heat exchanger 30a, second through the heat exchanger 30b and after that through the heat exchanger 30c.

[0063] On the side of the thermal storages, the heat exchangers 30a, 30b, 30c are interconnected by stream splitters 32a, 32b. The thermal storage medium coming from the high temperature storage 22 flows through the heat exchanger 30c, the stream splitter 32b, the heat exchanger 30b, the stream splitter 32a and the heat exchanger 30a to the low temperature storage 26. The first intermediate temperature storage 24a is connected to the thermal storage circuit 19 via the stream splitter 32a between the heat exchanger 30a and the heat exchanger 30b. The second intermediate temperature storage 24b is connected to the thermal storage circuit 19 via the stream splitter 32b between the heat exchanger 30b and the heat exchanger 30c.

[0064] The energy storage system 10 is adapted to generate different flowrates of thermal storage medium in the heat exchangers 30a, 30b, 30c by injecting or removing thermal storage medium from or to the intermediate temperature storages 24a, 24b into or from the thermal storage circuit 19.

[0065] In the working cycle 29, the compressed working fluid coming from the pump 34 is first heated inside the heat exchanger 30a with a flow of thermal storage medium coming from the stream splitter 32a. This flow has been generated by removing (or adding) thermal storage medium from (or to) the flow coming from the heat exchanger 30b into (or from) the intermediate storage 24a. After that the working fluid is heated in the heat exchanger 30b with a flow thermal storage medium of a different flowrate. This flow has been generated by adding (or removing) thermal storage medium from (or injected into) the intermediate storage 24b to the flow from the heat exchanger 30c. In the heat exchanger 30c, the working fluid is thus heated with a flow of thermal storage medium of a further different flowrate coming from the high temperature storage 22 and generated, for example, by a pump.

[0066] Fig. 4 shows a diagram depicting temperatures and state transitions of the bled steam, the thermal storage medium and the working fluid inside the system 10. The diagram is a T-h-diagram (temperature-enthalpy-diagram) in which isobars 50 of CO_2 and the saturation line 64 are indicated. The vertical axis shows the temperature T. The enthalpy h is plotted on the horizontal axis. Further, the diagrams shows the temperature profile from T_{C1} to T_{C4} (following the isobar 52 corresponding to the high pressure side of the CO_2 heat engine thermodynamic cycle) of the CO_2 in the heat exchanger 30, the bottoming trans critical CO_2 heat engine thermodynamic cycle 54, the temperature profile 56 of the water in the heat exchangers 20, 30 and the temperature profile 58 of the bled steam. The temperature profile 60 is the profile of the condenser cooling water in the condenser 38.

[0067] In the following, the charging cycle shown in Fig. 2 will be explained with respect to the temperature profile of Fig. 4. The steam bled at the lowest stage of the turbine 14 at a temperature T_{S1} heats the water of the storage 26 from T_{W4} (about 40 °C) to T_{W3} (about 60 °C) through the heat exchanger 20a. The steam is cooled in the heat exchanger (water heater) 20a from T_{S1} (about 65 °C) to T_{S1} (about 40 °C), and provides its energy to the storage water mostly isothermally (at about 60 °C) in a latent form during the steam condensation (see steam temperature profile 58a). T_{W3} is selected so that the temperature profile 56a of the heated water (the straight line from T_{W4} to T_{W3} in the Fig. 4) matches as close as possible (keeping however a reasonable temperature difference to allow the heat transfer) the part 54a of the temperature profile of the CO_2 heat engine cycle that will be run at the time this stored water will be used for the discharging cycle (between T_{C1} and T_{C2} in the Fig. 4).

[0068] The skilled person will be aware that the temperature profile of the CO_2 heat engine cycle should be chosen such that it will match as close as possible to the temperature profile of the steam (for example between T_{C1} and T_{C4}) yielding the highest work output ie. generally for the given steam bleed conditions operated at the highest technical possible CO_2 pressure in the heat exchanger; having in between the two temperature profiles a temperature profile of a sensible heat storage.

[0069] Prior to entering the second heat exchanger 20b, the heated water leaving the first heat exchanger 20a goes to the stream splitter 28a where it is either mixed with water coming from the tank 24a at the same temperature T_{W3}, or on the contrary, part of it is stored in the tank 24a.

[0070] The resulting water stream or flow (after mixing/splitting) is then heated from T_{W3} to T_{W2} (about 70 °C) through the second heat exchanger 20b by the steam bled at a second stage of the turbine 14 at a temperature T_{S2} (about 95 °C). The steam is cooled in the water heater from T_{S2} to T'_{S2} (about 60 °C), wherein the steam condensation occurs at about 80 °C (see steam temperature profile 58b). The choice of the flow rate for either mixing or splitting the water leaving the first heat exchanger 20a allows a close matching of the part 54b of the temperature profile of the CO_2 heat engine between T_{C2} and T_{C3} with the temperature profile 56b of the heated water (straight line from T_{W3} to T_{W2}). This is a direct use of the stream splitting concept used to match the temperature profile of a heat source with the one of a heat engine, having in between the two a sensible heat storage.

[0071] In the same way, the water leaving the second heat exchanger 20b goes to the stream splitter 28b where it is either mixed with water coming from a tank 24b at the same temperature T_{W2}, or split so that part of it is stored in a tank 24b. The resulting water stream or flow is then heated from T_{W2} to T_{W1} through a third heat exchanger 20c by the steam bled at a third stage of the turbine at a temperature T_{S3} (about 140 °C). The steam is cooled in the heat exchanger 20c from T_{S3} to T'_{S3} (about 80 °C), wherein the steam condensation occurs at about 120 °C (see steam temperature profile 58c). The water heated at T_{W1} is stored in the high temperature tank 22 that could be pressurized to keep water in its liquid form if its temperature exceeds the boiling temperature of water at ~100°C.The choice of the flow rate for either mixing or splitting the water leaving the second heat exchanger 20b is done in order to obtain the best matching of the part 54c of the temperature profile of the CO_2 heat engine between T_{C3} and T_{C4} with the temperature profile 56c of the heated water (straight line from T_{W2} to T_{W1} in the Fig. 4). After each of the heat exchangers 20a, 20b, 20c, the condensed steam is returned into the power plant circuit.

[0072] In the following the working cycle shown in Fig. 3 will be explained with respect to the temperature profile of Fig. 4.

[0073] During this mode, all of the water streams are reverted. The water stored in the high temperature storage 22 at the temperature T_{W1} heats the CO_2 entering the turbine from the temperature T_{C3} (about 75 °C) to T_{C4} (about 120 °C), in the heat exchanger 30c (temperature profiles 54c and 56c). The water leaving the heat exchanger 30c at T_{W2} is then either mixed with water from the intermediate storage 24b at T_{W2} (if the water stream leaving the heat exchanger 20b was split during charging) or part of it is split and redirected to the intermediate storage 24b (if the water stream leaving the heat exchanger 20b was mixed during charging), in order to keep the same temperature profile 56c of the water than during the charging period. This implies that the mass flow ratios of the reverted water streams must be kept. I. e. the flowrates of the thermal storage medium through the heat exchangers 30a, 30b, 30c should be proportional to the flowrates of the thermal storage medium through the heat exchangers 20a, 20b, 20c. In other words, the ratios of the flowrates of thermal storage medium in the heat exchangers 30a, 30b, 03c should be equal to the ratios of the flowrates of thermal storage medium in the heat exchangers 20a, 20b, 20c.

[0074] In the same way, the resulting water entering the heat exchanger 30b at a temperature T_{W2} heats the CO_2 from the temperature T_{C2} (about 55°C) to T_{C3} (see temperature profiles 54b and 56b). The water leaves the heat exchanger 30b at a temperature T_{W3}, and is either mixed (if the water stream leaving the heat exchanger 20a was split during charging) or split (if the water stream leaving the heat exchanger 20a was mixed during charging) with water from or going to the intermediate storage 24a. Finally, the resulting water entering the heat exchanger 30a at a temperature T_{W3} heats the CO_2 leaving the pump 34 from the temperature T_{C1} (about 35 °C) to T_{C2} (see temperature profiles 54a and 56a), and the water leaving the heat exchanger 30a at T_{W4} is stored in the low temperature storage 26.

[0075] The working fluid leaving the heat exchanger 30c at T_{C4} is expanded in the turbine 36 (temperature profile 54d) and cooled in the condenser 38 (temperature profile of CO_2 54e). After that the working fluid is compressed by the pump 34 and enters the heat exchanger 30a again at T_{C1} (temperature profile 54f).

[0076] The duration of the periods of power reduction (charging) and power increase (discharging) can be different, and are only restrained by the size of the heat storage system 10. In such a case, with respect to the storage (charging) cycle, all reverted mass flows in the working (discharging) cycle could be selected as increased or reduced by the same factor in order to optimize the discharging power and/or discharging time, while keeping the same temperature profiles in the heat exchangers than during charging.

[0077] Fig. 4 shows that the CO_2 is heated in the supercritical phase in the heat exchanger arrangement 30. The pressure 52 of the CO_2 is set such that the state of the working fluid never crosses the liquid-gas domain and its temperature profile 54a, 54b, 54c in the present case is above the critical point (marked in Figure 4 on saturation line 64) of the CO_2.

[0078] The overall CO_2 heat engine cycle is called transcritical since the high pressure part is entirely in a supercritical phase or goes from a liquid state at a pressure above the critical pressure to a supercritical state, and in which the low pressure part of the cycle is in the subcritical phase and goes from the gas phase to the liquid phase.

[0079] The storages 22, 24, 24a, 24b, 26 may be tanks. In particular, the storage 22 may be a pressure vessel such that the water heated to 120° C is not boiling. The exemplary sizing of the storages for a 5MW electrical output of the CO_2 heat engine 29 may be the following (with daily cycles of 8 hours); the storage 22 may have a volume of 3000m^3,the storage 24b a volume of 2000m3, the storage 24a a volume of 500m3 and the storage 26 a volume of 5000m3. This makes a total volume of 10500m3,which is equivalent to a sphere of radius of 13.5 m.

[0080] The better the fit between the temperature profiles of the steam 58a, 58b, 58c with the temperature profiles of the thermal storage medium in the heat exchanger arrangement 20 and the temperature profiles of the working fluid 54a, 54b, 54c with the temperature profiles of the thermal storage medium 56a, 56b, 56c inside the heat exchanger arrangement 30, the less work is lost during the heat transfer with respect to work that would have been done by the bled steam if it went directly through the steam turbine instead. The lost work is dependent upon the area between the temperature profiles in a temperature-enthalpy or temperature-entropy diagram (the larger the area, the greater the work lost). In such a way, by minimizing the area between the temperature profiles, the efficiency of the heat engine cycle may be increased.

[0081] However, the temperature of the steam bled at different stages usually is predetermined by the design of the turbine 14. Thus, the temperatures T_{W1}, T_{W2} and T_{W3} (and thus T_{C2}, T_{C3} and T_{C4}) may be adapted to the temperatures of the bled steam.

[0082] The better the fit between the temperature profile of the working fluid in the heat exchanger 30 in the transcritical CO_2 heat engine cycle with the temperature profile 58a, 58b, 58c given by the inlet temperatures of the lower stages steam, the more electric energy is produced by the transcritical CO_2 bottoming heat engine cycle (for a given steam temperature profile) with respect to work that would have been done by the bled steam if it would go directly through the steam turbine instead.

[0083] While the invention has been illustrated and described in detail in the drawings and foregoing description, such illustration and description are to be considered illustrative or exemplary and not restrictive; the invention is not limited to the disclosed embodiments. Other variations to the disclosed embodiments can be understood and effected by those skilled in the art and practising the claimed invention, from a study of the drawings, the disclosure, and the appended claims. In the claims, the word "comprising" does not exclude other elements or steps, and the indefinite article "a" or "an" does not exclude a plurality. A single processor or controller or other unit may fulfil the functions of several items recited in the claims. The mere fact that certain measures are recited in mutually different dependent claims does not indicate that a combination of these measures cannot be used to advantage. Any reference symbol in the claims should not be construed as limiting the scope.

LIST OF REFERENCE SYMBOLS

[0084]

10	energy storage system
12	power plant
14	turbine
16	condenser
18	storage cycle
19	thermal storage circuit
20	heat exchanger arrangement (gas heater)
22	hot temperature storage
24	intermediate temperature storage
26	low temperature storage
28	stream splitter
29	working cycle
30	heat exchanger arrangement
32	stream splitter
34	pump
36	turbine
38	condenser
40	generator
42	cooling medium
24a	first intermediate temperature storage
24b	second intermediate temperature storage
20a	low temperature heat exchanger
20b	intermediate heat exchanger

20c high temperature heat exchanger

28a stream splitter

28b stream splitter

30a low temperature heat exchanger

30b intermediate temperature heat exchanger

30c high temperature heat exchanger

32a stream splitter

32b stream splitter

50 temperature levels of CO_2

52 temperature profile of CO_2 in heat exchanger

54 working cycle

56 temperature profile of water in heat exchanger

58 temperature profile of bled steam

60 temperature profile of condenser cooling water

64 saturation line

Claims

1. An energy storage system (10) for a power plant (12), comprising:

a thermal storage circuit (19) containing a thermal storage medium, the thermal storage circuit comprising a high temperature storage (22), an intermediate temperature storage (24, 24a, 24b) and a low temperature storage (26) connected together;
a steam heat exchanger arrangement (20) for heating the thermal storage medium with steam bled from a turbine (14) of the power plant (12);
a working fluid heat exchanger arrangement (30) for heating a working fluid, the working fluid flowing through a low temperature working fluid heat exchanger (30a) and a high temperature working fluid heat exchanger (30c);
wherein the thermal storage medium flows from the high temperature storage (22) through the high temperature working fluid heat exchanger (30c) and the low temperature working fluid heat exchanger (30a) to the low temperature storage (26);
wherein the intermediate temperature storage (24, 24a, 24b) is connected to the thermal storage circuit (18) between the high temperature working fluid heat exchanger (30c) and the low temperature working fluid heat exchanger (30a) with a stream splitter (32a, 32b);
wherein the energy storage system (10) is adapted to generate different flowrates of thermal storage medium in the high temperature working fluid heat exchanger (30c) and the low temperature working fluid heat exchanger (30a) by injecting thermal storage medium from the intermediate temperature storage (24, 24a, 24b) into a flow of thermal storage medium between the high temperature working fluid heat exchanger (30c) and the low temperature working fluid heat exchanger (30a) or by extracting thermal storage medium from a flow of thermal storage medium between the high temperature working fluid heat exchanger (30c) and the low temperature working fluid heat exchanger (30a).

2. The energy storage system (10) of claim 1,
wherein a high temperature (T_{W1}) of the thermal storage medium inside the high temperature storage (22), an intermediate temperature (T_{W3}) of the thermal storage medium inside the intermediate temperature storage (24a) and a low temperature (T_{W4}) of the thermal storage medium inside the low temperature storage (26), a flowrate of thermal storage medium through the high temperature working fluid heat exchanger (30c), a flowrate of thermal storage medium through the low temperate working fluid heat exchanger (30a) and a flowrate of working fluid through the working fluid heat exchanger arrangement (30) are set, such that a temperature profile (56a, 56b, 56c) of the thermal storage medium in the working fluid heat exchanger arrangement (30) is matched with a temperature profile (54a, 54b, 54c) of the working fluid.

3. The energy storage system (10) of one of the preceding claims,
wherein the working fluid is heated in the supercritical phase in the working fluid heat exchanger arrangement (30).

4. The energy storage system (10) of one of the preceding claims, further comprising:

a pump (34) for compressing the working fluid before the heating in the working fluid heat exchanger arrangement (30), a turbine (36) for generating mechanical energy by expanding the working fluid after the heating and a condenser (38) for cooling the working fluid between the turbine (36) and the pump (34).

5. The energy storage system (10) of one of the preceding claims, wherein the working fluid is carbon dioxide (CO_2).

6. The energy storage system (10) of one of the preceding claims, wherein the thermal storage medium is water.

7. The energy storage system (10) of one of the preceding claims, wherein the working fluid heat exchanger arrangement (30) comprises at least one intermediate working fluid heat exchanger (30b) arranged between the low temperature working fluid heat exchanger (30a) and the high temperature working fluid heat exchanger (30c); wherein the thermal storage circuit (19) comprises at least one further intermediate temperature storage (24a); wherein the working fluid in the intermediate working fluid heat exchanger (30b) is heated by a flow of thermal storage medium with an intermediate flowrate generated by injecting or extracting thermal storage medium from the intermediate storage (24b) into or from the thermal storage circuit (19) before the intermediate working fluid heat exchanger (30b) and by extracting or injecting thermal storage medium from or into the thermal storage circuit (19) after the intermediate working fluid heat exchanger (30b) into or from the further intermediate temperature storage (24a).

8. The energy storage system (10) of one of the preceding claims, wherein the thermal storage medium flows from the low temperature storage (26) through a low temperature steam heat exchanger (20a) and a high temperature steam heat exchanger (20c) to the high temperature storage (22); wherein in the low temperature steam heat exchanger (20a) steam of a low temperature (T_{S1}) bled from the turbine (14) is used for heating, and in the high temperature steam heat exchanger (20c) steam of a high temperature (T_{S3}) bled from the turbine (14) is used for heating; wherein the intermediate temperature storage (24, 24a, 24b) is connected to the thermal storage circuit (19) between the high temperature steam heat exchanger (20c) and the low temperature steam heat exchanger (20a); wherein the energy storage system (10) is adapted for generating different flow rates in the low temperature steam heat exchanger (20a) and the high temperature heat exchanger (20c) by injecting thermal storage medium from the intermediate temperature storage (24b) into the thermal storage circuit (19) or by extracting thermal storage medium from the thermal storage circuit (19) to the intermediate temperature storage (24b).

9. The energy storage system (10) of claim 8, wherein the ratio of a flowrate of thermal storage medium through the high temperature steam heat exchanger (20c) and a flowrate of thermal storage medium through the low temperate steam heat exchanger (20a) is equal to a ratio of a flowrate of thermal storage medium through the high temperature working fluid heat exchanger (30c) and a flowrate of thermal storage medium through the low temperate working fluid heat exchanger (30a).

10. The energy storage system (10) of one of the claims 8 or 9, wherein the steam heat exchanger arrangement (20) comprises at least one intermediate steam heat exchanger (20b) arranged between the low temperature steam heat exchanger (20a) and the high temperature steam heat exchanger (20c); wherein the thermal storage medium inside the intermediate steam heat exchanger (20b) is heated by steam of an intermediate temperature (T_{S2}) bled from the turbine (14); wherein a flow of thermal storage medium with an intermediate flow rate is generated inside the intermediate steam heat exchanger (20b) by injecting thermal storage medium from the further intermediate storage (24a) into the thermal storage circuit (19) before the intermediate steam heat exchanger (20b) and by extracting thermal storage medium from the thermal storage circuit (19) after the intermediated steam heat exchanger (20b) into the intermediate temperature storage (24b).

11. A use of an energy storage system (10) according to one of the claims 1 to 10 in a power plant (12).

12. A power plant (12) with an energy storage system (10) according to one of the claim 1 to 10.

13. A method for storing excess thermal energy of a power plant (12), the method comprising a storage cycle (18) for storing energy in a thermal energy storage and a working cycle (29) for generating electricity from the stored energy; the method comprising the steps of:

in the storage cycle, bleeding steam from a turbine (14) of the power plant (12);
in the storage cycle, heating a thermal storage medium with the steam;
in the working cycle, heating a working fluid with a flow of thermal storage medium coming with a first flowrate from a high temperature storage (22);
in the working cycle, generating a flow of thermal storage medium of a second flowrate flowing to a low temperature storage (26) by adding ther-

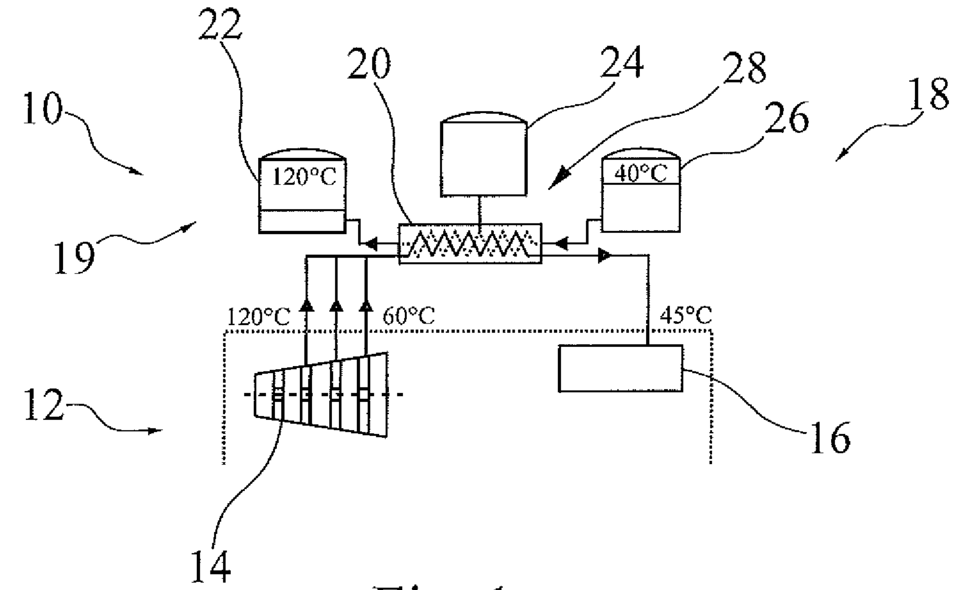
mal storage medium from an intermediate temperature storage (24a) to the flow or by removing thermal storage medium from the flow into a intermediate temperature storage (24a);
in the working cycle, heating the working fluid with the flow of thermal storage medium of the second flow rate.

14. The method of claim 13, further comprising the steps:

in the storage cycle, heating a flow of thermal storage medium with a first flowrate coming from the low temperature storage (26) with steam of a low temperature (T_{S1}) bled from the turbine (14);
in the storage cycle, generating a flow of thermal storage medium of a second flowrate flowing to the high temperature storage (22) by adding thermal storage medium from an intermediate temperature storage (24b) or by removing thermal storage medium into a intermediate temperature storage (24b);
in the storage cycle, heating the flow of thermal storage medium of the second flow rate with steam of a high temperature (T_{S3}) bled from the turbine (14).

15. The method of claim 14, further comprising the steps:

in the storage cycle, generating a flow of thermal storage medium of an intermediate flowrate by adding thermal storage medium to the flow of the first flowrate from a further intermediate temperature storage (24a);
in the storage cycle, heating the flow of thermal storage medium of the intermediate flow rate with steam of an intermediate temperature (T_{S2}) bled from the turbine (14);
in the storage cycle, generating the flow with the second flowrate by removing or adding thermal storage medium to the intermediate temperature storage (24a);
in the working cycle, generating a flow of thermal storage medium of an intermediate flowrate by adding or removing thermal storage medium from the intermediate temperature storage (24b) to the flow with the first flowrate coming from the storage (22);
in the working cycle, heating the working fluid with the flow of the intermediate flow rate;
in the working cycle, generating the flow of thermal storage medium of the second flowrate by removing thermal storage medium from the flow with intermediate flowrate into the further intermediate temperature storage (24b).


22
20
24
28
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18
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40°C
19
120°C
60°C
45°C
12
16
14

Fig. 1a

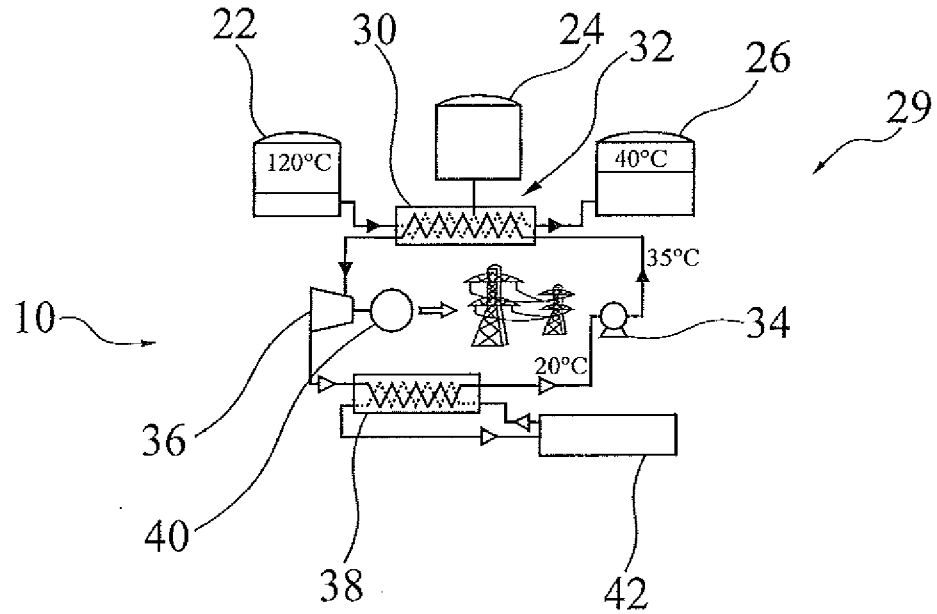

22
30
24
32
26
29
120°C
40°C
35°C
10
34
20°C
36
40
38
42

Fig. 1b

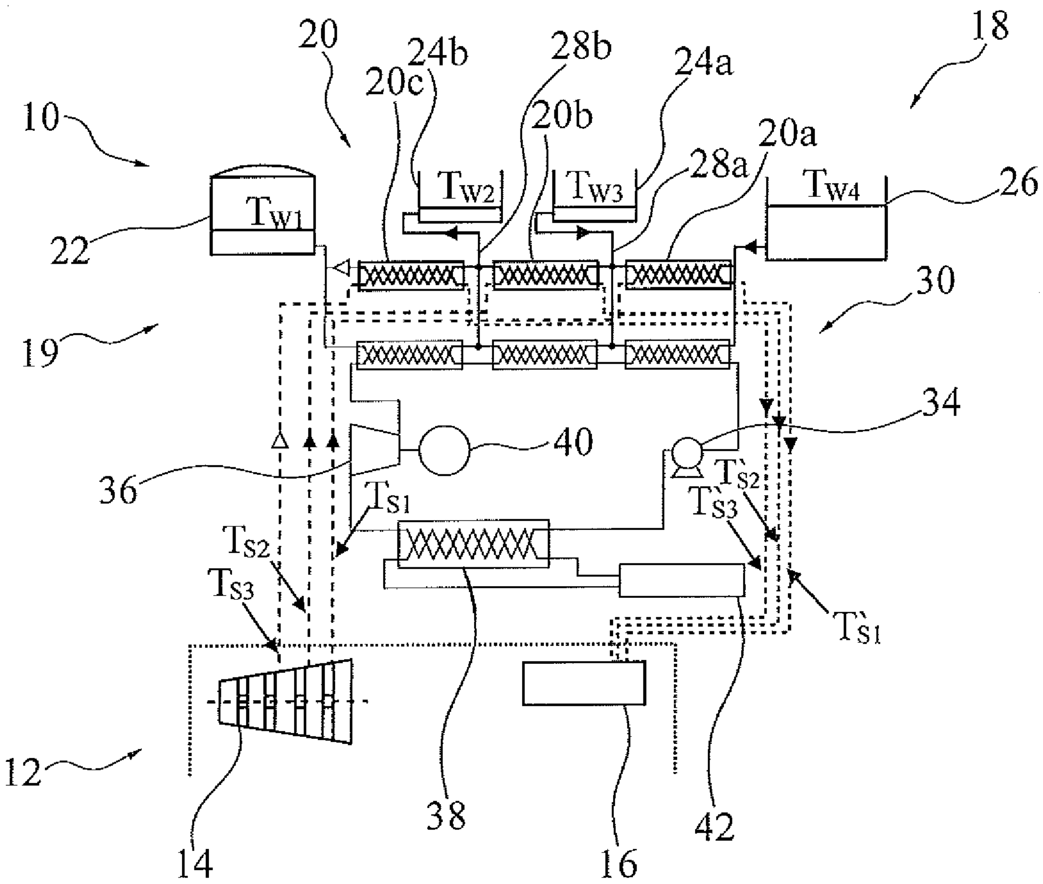

10
12
14
16
18
19
20
20a
20b
20c
22
24a
24b
26
28a
28b
30
34
36
38
40
42
T_{W1}
T_{W2}
T_{W3}
T_{W4}
T_{S1}
T_{S2}
T_{S3}
$T`_{S1}$
$T`_{S2}$
$T`_{S3}$

Fig. 2

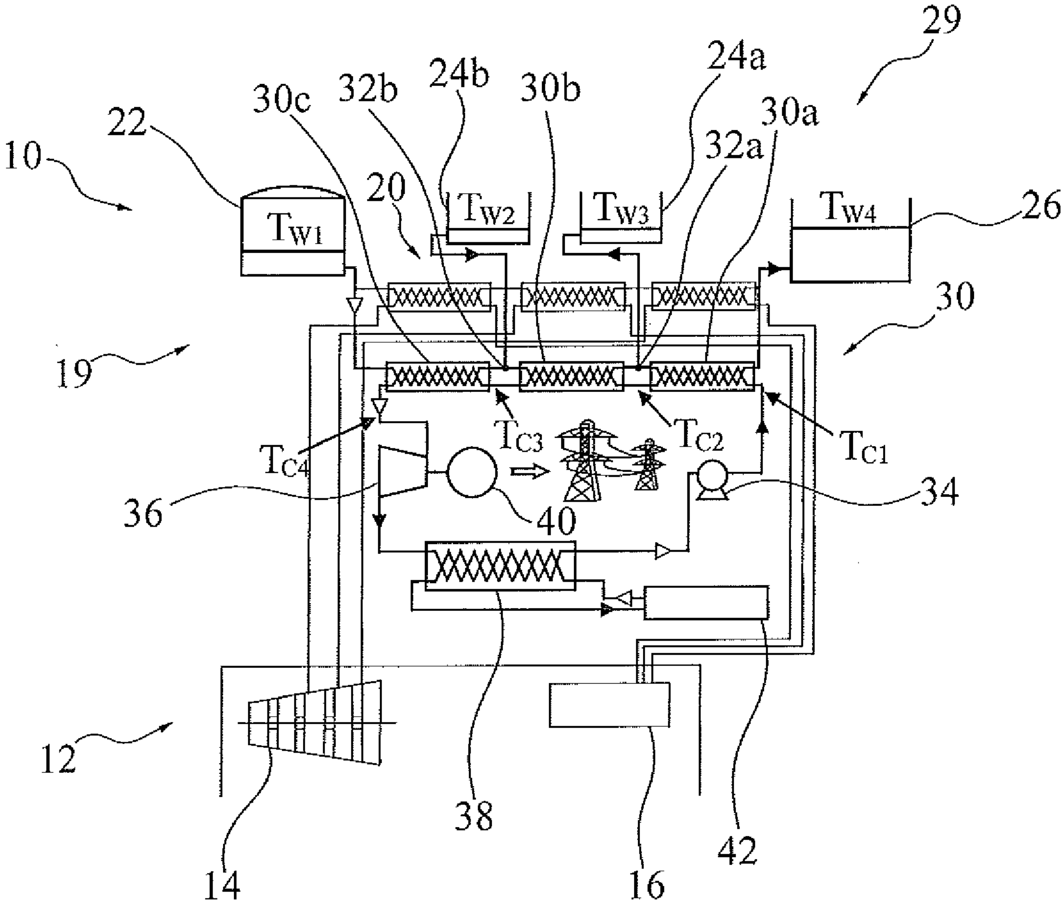
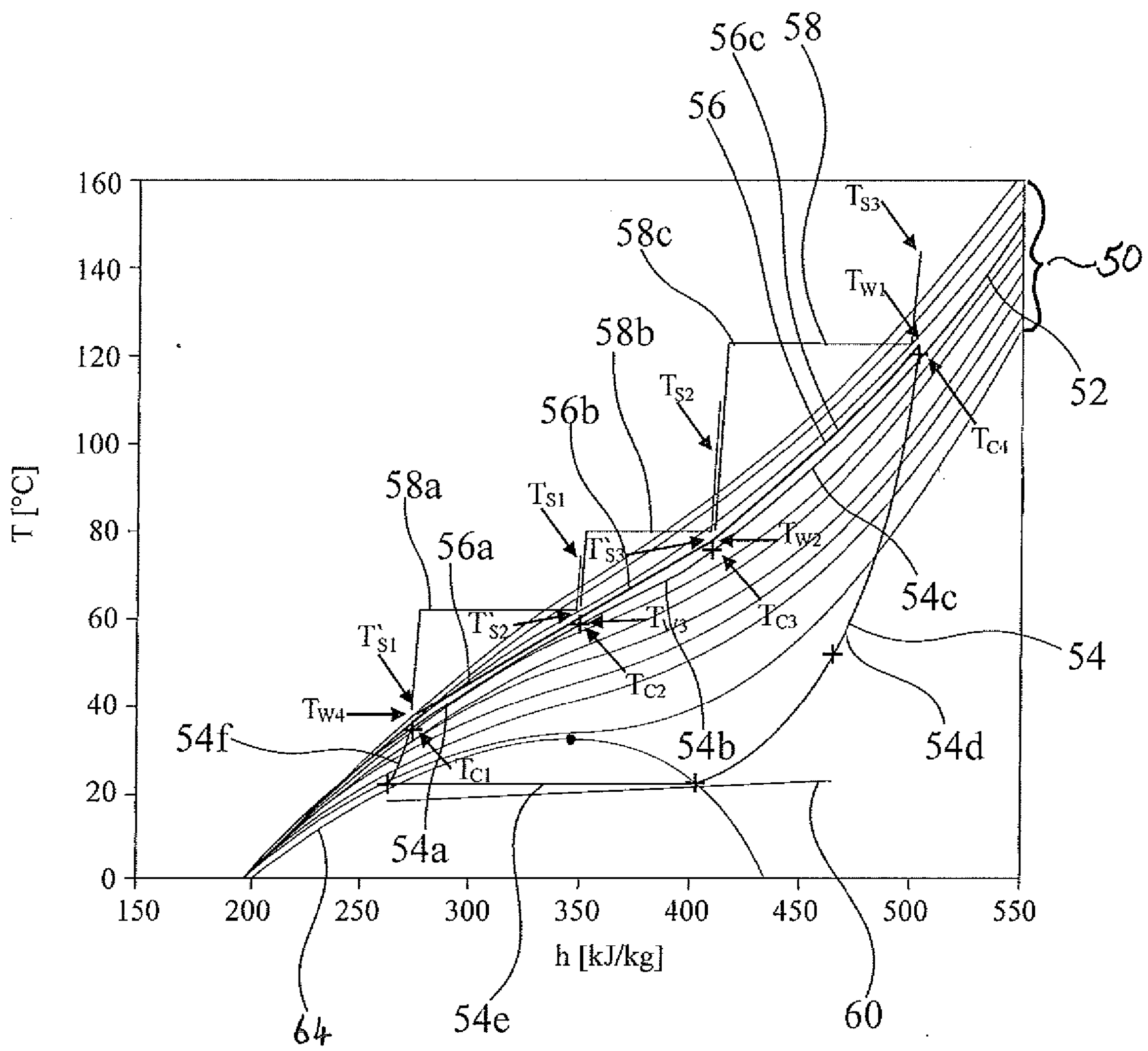

29
24a
24b
30b
30c
32b
30a
22
32a
10
20
T_{W2}
T_{W3}
T_{W4}
26
T_{W1}
30
19
T_{C3}
T_{C2}
T_{C1}
T_{C4}
36
34
40
12
38
42
14
16

Fig. 3



56c
58
56
160
140
120
100
80
60
40
20
0
T [°C]
150
200
250
300
350
400
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500
550
h [kJ/kg]
50
52
54
54a
54b
54c
54d
54e
54f
56a
56b
58a
58b
58c
60
64

Fig. 4

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EUROPEAN SEARCH REPORT

Application Number
EP 10 18 7581

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (IPC)
A	GB 1 524 236 A (EXXON RESEARCH ENGINEERING CO) 6 September 1978 (1978-09-06) * figure 1 * -----	1-15	INV. F01K3/12 F01K25/10
			TECHNICAL FIELDS SEARCHED (IPC)
			F01K

1

The present search report has been drawn up for all claims

Place of search	Date of completion of the search	Examiner
Munich	19 October 2011	Coquau, Stéphane

CATEGORY OF CITED DOCUMENTS

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19-10-2011

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