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(54) **Electricity storage**

(57) A thermal energy storage system comprises a first thermal energy store arrangement (16) having an upper operating temperature ($T1_{max}$) and a lower operating temperature ($T1_{min}$) and a second thermal energy store arrangement (32) having an upper operating temperature ($T2_{max}$) and a lower operating temperature ($T2_{min}$), wherein the upper operating temperature ($T2_{max}$) of the second thermal energy store arrangement is less than or equal to the lower operating temperature ($T1_{min}$) of the first thermal energy store arrangement (16). The system further comprises a working fluid circuit (8) containing a working fluid for circulation through the first and second thermal energy store arrangements (16, 32). The thermal energy storage system has a charging cycle in which thermal energy is transferred from the circulating working fluid to the first and second thermal energy store arrangements (16, 32) and a discharging cycle in which thermal energy is transferred from the first and second thermal energy store arrangements (16, 32) to the circulating working fluid. The first thermal energy store arrangement (16) preferably uses molten salt as the thermal energy storage medium whilst the second thermal energy store arrangement (32) preferably uses pressurised water as the thermal energy storage medium.

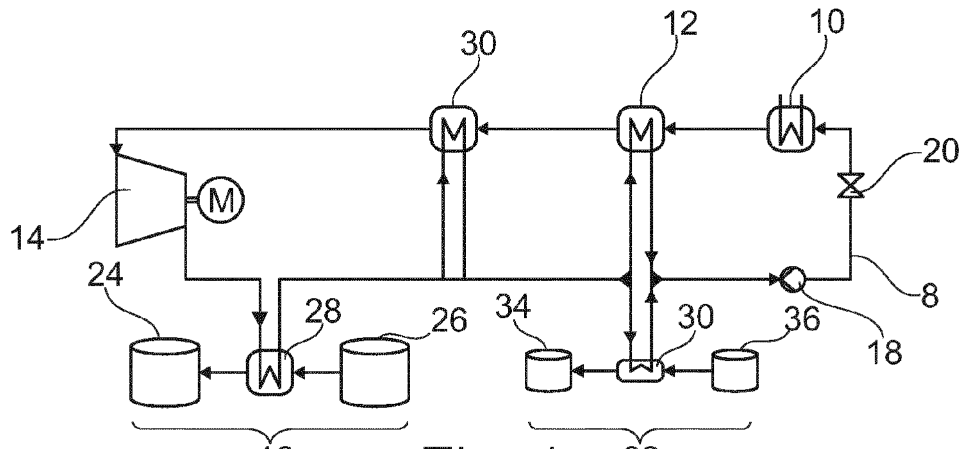

Fig. 1

Description

Technical Field

[0001] The present disclosure relates generally to the field of electricity storage and more particularly to the storage of electricity in the form of thermal energy. The present disclosure is especially concerned with pumped heat electricity storage (PHES). Specific embodiments relate to a thermal energy storage system.

Technical Background

[0002] Further penetration of fluctuating renewable energy production requires economic solutions for bulk electricity storage. Today's leading technology is Pumped Hydro Storage (PHS). A possible alternative is Compressed Air Energy Storage (CAES). Whilst PHS requires the right topography (mountains), CAES relies on the presence of specific geological underground structures, such as salt caverns. Other forms of energy storage include batteries and flywheels.

[0003] Pumped Heat Electricity Storage (PHES) has been proposed as an alternative storage technique. During charging a PHES system pumps heat from a low temperature reservoir to a high temperature reservoir, it therefore operates as a heat pump. During discharging the high temperature heat is used to drive a power cycle whilst the residual heat is rejected into the low temperature reservoir. The obvious advantage of such a system is that the electricity is stored only under the form of heat or thermal energy, i.e. it requires only some kind of thermally isolated containment that is independent of geology or topography.

[0004] An implementation of PHES based on air or other non-condensing gases is described in US 2010/257862 A1. During charging this system is basically an air heat pump. Air or another non-condensing gas is compressed which causes it to heat up. The thermal energy is passed to a high temperature thermal energy store. The now cold compressed gas is expanded back to the original pressure and cools down far below ambient temperature. The cold expanded air is heated back up to approximately ambient temperature by cooling a cold thermal energy store. After charging the electricity is stored under the form of heat at different temperature levels. During discharging low pressure air is cooled in the low temperature thermal energy store and compressed to a higher pressure level. The compressed air then picks up thermal energy from the hot thermal energy store before it is expanded in a turbine operatively connected to an electrical generator for electricity generation. The losses that accumulate during charging and discharging have to be rejected against the environment. This can happen at different points, most obviously at the exhaust of the turbine.

[0005] Some drawbacks of this non-condensing PHES system have to be mentioned. During charging and discharging there are always two turbomachines active: a hot compressor and a cold turbine during charging, a cold compressor and a hot turbine during discharging. Therefore component efficiencies of these turbomachines have a large impact on the overall electrical-to-electrical (i.e. round-trip electrical) efficiency. Furthermore, this PHES system requires turbomachines, heat exchangers and thermal energy stores to operate at temperatures far below zero, which makes material selection a critical factor. The capital expenditure for the turbomachines is proportional to the sum of the power of the components, while the charging power is related to the difference in power of the hot compressor and cold turbine. Similarly, the discharging power is given by the difference in power of the cold compressor and hot turbine while the capital expenditure is proportional to the sum of the power of the components. Therefore, specific costs may be high. Finally, the non-condensing PHES system cannot produce any power once the hot and the cold thermal energy stores are emptied. This is the same as for an adiabatic compressed air energy storage (A-CAES) system where no power can be generated if there is no pressurised air in the cavern. Non-condensing PHES, A-CAES or batteries and flywheels are pure electricity storage devices and cannot deliver power if they have not been charged before. Although this is commonly accepted, it is a major drawback since expensive hardware is installed but cannot help if renewable energy sources such as wind fail to produce electricity for a few days.

[0006] Some of theses drawbacks are at least partially mitigated by the thermoelectric energy storage system described in WO 2010/020480 A2 which employs a heat exchanger to transfer thermal energy between a condensable working fluid and a sensible heat thermal storage medium circulating between cold and hot storage tanks. Thermal energy is transferred from the working fluid to the thermal storage medium during a charging cycle and is transferred from the thermal storage medium to the working fluid during a discharging cycle in which electrical energy is generated by expansion of the heated working fluid in a turbine. The condensable working fluid is heated and compressed to a supercritical state during both the charging and discharging cycles and this maximises the round-trip electrical efficiency of the system.

[0007] Round-trip electrical efficiency is increased further in the thermoelectric energy storage system described in WO 2011/045282 A2 due to the provision of an internal heat exchanger. The internal heat exchanger preheats the working fluid during both the charging and discharging cycles, thereby maximising system efficiency.

[0008] There remains a need for an improved thermal energy storage system which achieves a high round-trip electrical efficiency with minimal capital expenditure.

Summary of the Disclosure

[0009] According to one aspect, there is provided a thermal energy storage system comprising:

a first thermal energy store arrangement having an upper operating temperature ($T1_{max}$) and a lower operating temperature ($T1_{min}$);
a second thermal energy store arrangement having an upper operating temperature ($T2_{max}$) and a lower operating temperature ($T2_{min}$), wherein the upper operating temperature ($T2_{max}$) of the second thermal energy store arrangement is less than or equal to the lower operating temperature ($T1_{min}$) of the first thermal energy store arrangement; and
a working fluid circuit containing a working fluid for circulation through the first and second thermal energy store arrangements;
the thermal energy storage system having a charging cycle in which thermal energy is transferred from the circulating working fluid to the first and second thermal energy store arrangements and a discharging cycle in which thermal energy is transferred from the first and second thermal energy store arrangements to the circulating working fluid.

[0010] The first thermal energy store arrangement may be regarded as a high-temperature thermal energy store arrangement whilst the second thermal energy store arrangement may be regarded as a medium-temperature thermal energy store arrangement. The provision of the second, medium-temperature, thermal energy store arrangement enables the storage temperature of the first, high-temperature, thermal energy store arrangement to be increased. More particularly, the upper and lower operating temperatures ($T1_{max}$, $T1_{min}$) of the first thermal energy store arrangement can be increased because the thermal energy storage is shared between the first and second thermal energy store arrangements. Increasing the upper and lower operating temperatures of the first thermal energy store arrangement enables a suitable and commercially available high-temperature thermal energy storage medium to be used in the first thermal energy store arrangement.

[0011] The first thermal energy store arrangement may utilise a first, high-temperature, thermal energy storage medium to receive and store thermal energy. The first thermal energy storage medium is typically a liquid and may advantageously be a molten salt such as a nitrate salt or a carbonate salt, or a mixture of molten salts. Other forms of storage medium could, however, be used such as thermal oil, water, sand, gravel or any other suitable particulate material. The first thermal energy storage medium is capable of absorbing heat in a highly efficient manner, i.e. with a small temperature approach, at almost constant heat capacity from the working fluid.

[0012] The second thermal energy store arrangement may utilise a second, low-temperature, thermal energy storage medium to receive and store thermal energy. The second thermal energy storage medium may be pressurised water.

[0013] The first thermal energy store arrangement may comprise first and second storage tanks, for example high-temperature and medium-temperature storage tanks, for the first thermal energy storage medium and may comprise a heat exchange device arranged between the storage tanks to transfer thermal energy between the working fluid and the first thermal energy storage medium as it flows through the heat exchange device between the storage tanks. The high temperature storage tank may have an upper operating temperature ($T1_{max}$) between 300 and 600°C and the medium temperature storage tank may have a lower operating temperature ($T1_{min}$) between 200 and 400°C.

[0014] The second thermal energy store arrangement may comprise first and second storage tanks, for example medium-temperature and low-temperature storage tanks, for the second thermal energy storage medium and may comprise a heat exchange device arranged between the storage tanks to transfer thermal energy between the working fluid and the second thermal energy storage medium as it flows through the heat exchange device between the storage tanks. The medium temperature storage tank may have an upper operating temperature ($T2_{max}$) between 100 and 300 °C and the low temperature storage tank may have a lower operating temperature ($T2_{min}$) between 20 and 100 °C.

[0015] The system may further comprise first and second heat exchangers and a compressor. The first and second heat exchangers may be operable during the charging cycle to transfer thermal energy to the circulating working fluid to heat the working fluid prior to compression in the compressor. The compressor may be operable during the charging cycle to compress the heated working fluid prior to circulation through the first and second thermal energy store arrangements. The compressor may be arranged to compress the working fluid to a supercritical state.

[0016] The working fluid may be a condensable working fluid. The condensable working fluid may be an organic fluid such as an alkane. The condensable organic working fluid may advantageously be a C_3 alkane, a C_4 alkane or a C_5 alkane.

[0017] The first heat exchanger may be operable during the charging cycle to transfer thermal energy from the ambient environment to the circulating working fluid to heat the circulating working fluid. The first heat exchanger may be operable during the discharging cycle to transfer thermal energy from the circulating working fluid to the ambient environment to cool the circulating working fluid. The first heat exchanger is typically arranged to transfer thermal energy between ambient temperature sea water or river water and the circulating working fluid.

[0018] The first heat exchanger may be an evaporator/condenser that is operable during the charging cycle as an evaporator to evaporate the circulating condensable working fluid and operable during the discharging cycle as a condenser to condense the circulating condensable working fluid.

[0019] The system may include a first flow divider/combiner valve between the first and second thermal energy store arrangements. The first flow divider/combiner valve may be operable during the charging cycle to divide the circulating working fluid into a first fluid flow to the second heat exchanger to transfer thermal energy from the first fluid flow to the circulating working fluid output from the first heat exchanger; and a second fluid flow to the second thermal energy store arrangement to transfer thermal energy from the second fluid flow to the second thermal energy store arrangement. The first fluid flow may comprise between about 60 and 90% of the total flow of circulating working fluid whilst the second fluid flow may comprise between about 10 and 40% of the total flow of circulating working fluid.

[0020] The system may include a second flow divider/combiner valve which may be operable during the charging cycle to combine the first and second fluid flows respectively from the second heat exchanger and the second thermal energy store arrangement.

[0021] The system may further comprise an expander which may be operable during the charging cycle to expand the circulating working fluid output from the second flow divider/combiner valve. The expander may be a liquid expander which may be arranged to expand the circulating working fluid to a pressure above which evaporation of the working fluid occurs.

[0022] The system may further comprise a throttle which may be operable during the charging cycle to reduce the pressure of the circulating working fluid output from the expander. The throttle may be arranged to reduce the pressure of the expanded circulating working fluid at least to level at which the working fluid separates into a vapour and a liquid phase. The throttle thus closes the reversible cycle and after throttling into the wet region, the circulating working fluid is returned to the first heat exchanger.

[0023] The system may further comprise a third heat exchanger which may be operable during the charging cycle to transfer thermal energy to the circulating working fluid output from the second heat exchanger prior to compression of the heated circulating working fluid in the compressor. The third heat exchanger may be operable during the charging cycle to transfer thermal energy from the circulating working fluid output from the first thermal energy store arrangement to the circulating working fluid output from the second heat exchanger prior to compression of the heated circulating working fluid in the compressor.

[0024] The system may further comprise a turbine which may be operable during the discharging cycle to expand heated circulating working fluid. The second flow divider/combiner valve may be operable during the discharging cycle to divide the circulating working fluid into a first fluid flow to the second heat exchanger to enable the transfer of residual thermal energy from the circulating working fluid output from the turbine to the first fluid flow to thereby heat the first fluid flow; and a second fluid flow to the second thermal energy store arrangement to enable the transfer of thermal energy from the second thermal energy store arrangement to the second fluid flow to thereby heat the second fluid flow. The first fluid flow may comprise between about 60 and 90% of the total flow of circulating working fluid whilst the second fluid flow may comprise between about 10 and 40% of the total flow of circulating working fluid. The first flow divider/combiner valve may be operable during the discharging cycle to combine the heated first and second fluid flows output respectively from the second heat exchanger and the second thermal energy store arrangement.

[0025] The third heat exchanger may be operable during the discharging cycle to transfer residual thermal energy from the circulating working fluid output from the turbine to the combined flow of circulating working fluid output from the first flow divider/combiner valve. The circulating working fluid is, thus, preheated during the discharging cycle before it is supplied to the first thermal energy store arrangement for further heating.

[0026] The compressor may be an adiabatic compressor. The system may include an electric motor which is arranged to drive the compressor during the charging cycle. The turbine may be an adiabatic turbine. The turbine may be operatively associated with an electrical generator to generate electrical power during the discharging cycle.

Brief Description of the Drawings

[0027]

Figure 1 is schematic illustration of a charging cycle of a first embodiment of a thermal energy storage system;

Figure 2 is schematic illustration of a discharging cycle of the thermal energy storage system illustrated in Figure 1; and

Figure 3 is a schematic illustration of a discharging cycle of a second embodiment of a thermal energy storage system similar to that illustrated in Figure 2.

Detailed Description of Embodiments

[0028] Embodiments will now be described by way of example only and with reference to the accompanying drawings.

[0029] Referring initially to both Figures 1 and 2, there is shown a first embodiment of a thermal energy storage system in the form of a pumped heat electricity storage (PHES) system. Electricity is stored in the form of thermal energy during the charging cycle illustrated in Figure 1 in which the system operates as a heat pump. Electricity is generated by recovering thermal energy during the discharging cycle illustrated in Figure 2 in which the system operates as a Rankine cycle.

[0030] The thermal energy storage system comprises a reversible closed cycle working fluid circuit 8 in which the circulating working fluid is a condensable organic working fluid. The condensable organic working fluid may typically be a C_3 alkane such as propane, a C_4 alkane such as butane or a C_5 alkane such as pentane.

[0031] The system comprises a first heat exchanger in the form of an evaporator/condenser 10 which is arranged to transfer thermal energy between ambient temperature sea water or river water and the circulating working fluid. The system comprises a second heat exchanger in the form of a recuperator 12, a third heat exchanger in the form of a recuperator 30, a first (high-temperature) thermal energy store arrangement 16, a second (medium-temperature) thermal energy store arrangement 32, a liquid expander/pump 18 and a throttle 20. The system further comprises an adiabatic compressor 14 which is driven by a motor M during the charging cycle illustrated in Figure 1 and an adiabatic turbine 22 operatively connected to a generator G so that electricity can be generated during the discharging cycle illustrated in Figure 2.

[0032] The high temperature thermal energy store arrangement 16 utilises molten salt as a sensible heat storage medium and includes an insulated hot salt storage tank 24, an insulated relatively warm salt storage tank 26 and a heat exchanger 28 to transfer heat between the working fluid and the molten salt as the molten salt flows between the hot and warm salt storage tanks 24, 26. During the charging cycle, the heat exchanger 28 transfers heat from the circulating working fluid to the molten salt as the molten salt flows from the relatively warm salt storage tank 26 to the hot salt storage tank 24. During the discharging cycle, the heat exchanger 28 recovers heat from the hot molten salt as it flows from the hot salt storage tank 24 to the relatively warm salt storage tank 26 and transfers the recovered heat to the working fluid as it is circulated through the heat exchanger 28.

[0033] The medium temperature thermal energy store arrangement 32 utilises pressurised water as a thermal energy storage medium and includes a relatively warm insulated pressurised water storage tank 34, a relatively cool pressurised water storage tank 36 and a heat exchanger 38 to transfer heat between the working fluid and the pressurised water as the pressurised water flows between the warm and cool water storage tanks 34, 36. During the charging cycle, the heat exchanger 38 transfers heat from the circulating working fluid to the pressurised water as the pressurised water flows from the relatively cool water storage tank 36 to the relatively warm water storage tank 34. During the discharging cycle, the heat exchanger 38 recovers heat from the warm pressurised water as it flows from the relatively warm water storage tank 34 to the relatively cool water storage tank 36 and transfers the recovered heat to the working fluid as it is circulated through the heat exchanger 38.

[0034] The high-temperature thermal energy store arrangement 16 operates at a higher temperature than the medium-temperature thermal energy store arrangement 32. In one implementation of the system illustrated in Figures 1 and 2, molten salt is stored in the hot salt storage tank 24 at an upper operating temperature ($T1_{max}$) in the region of 440°C and in the relatively warm salt storage tank at a lower operating temperature ($T1_{min}$) in the region of 290°C, whereas pressurised water is stored in the relatively warm water storage tank 34 at an upper operating temperature ($T2_{max}$) in the region of 240°C and in the relatively cool water storage tank 36 at a lower operating temperature ($T2_{min}$) in the region of 50°C.

[0035] The system includes first and second flow divider/combiner valves 40, 42 which are selectively operable to divide and combine the flow of circulating working fluid into first and second fluid flows. More particularly, during the charging cycle the first flow divider/combiner valve 40 divides the flow of circulating working fluid into a first fluid flow that is supplied to the recuperator 12 and a second fluid flow that is supplied to the medium-temperature thermal energy store arrangement 32 whilst the second flow divider/combiner valve 42 combines the first and second fluid flows. Conversely, during the discharging cycle the second flow divider/combiner valve 42 divides the flow of circulating working fluid into a first fluid flow that is supplied to the recuperator 12 and a second fluid flow that is supplied to the medium-temperature thermal energy store arrangement 32 whilst the first flow divider/combiner valve 40 combines the first and second fluid flows.

[0036] The operation of the thermal energy storage system in the charging and discharging cycles will now be explained in more detail.

<u>Charging Cycle - Electricity Storage</u>

[0037] As indicated above, the system operates as a heat pump during the charging cycle illustrated in Figure 1. In more detail, the condensable organic working fluid is evaporated in the evaporator 10 by absorbing heat from ambient sea water or river water. The resulting vapour is then preheated in the recuperators 12, 30 by absorbing residual heat recovered from the high pressure side of the closed cycle. The preheated vapour is compressed in the adiabatic compressor 14, driven by the motor M, to supercritical conditions, i.e. so that it has a temperature and pressure in the supercritical range.

[0038] The supercritical working fluid is passed to the heat exchanger 28 of the high-temperature thermal energy store arrangement 16 where heat is transferred to the molten salt as it flows from the relatively warm salt storage tank 26 to the hot salt storage tank 24. Because the working fluid is compressed in the compressor 14 to supercritical conditions, the supercritical working fluid entering the heat exchanger 28 has an almost constant heat capacity that is well matched to the molten salt. This maximises the efficiency of the transfer of thermal energy to the molten salt.

[0039] After exiting the heat exchanger 28, the circulating working fluid is supplied to the recuperator 30 which recovers residual heat from the circulating working fluid and transfers the recovered residual heat as described above to the working fluid in the low pressure side of the cycle to preheat the low pressure working fluid prior to delivery to the compressor 14.

[0040] The high-pressure working fluid output from the recuperator 30 is then divided by the first flow divider/combiner valve 40 into first and second fluid flows. The first fluid flow, comprising approximately 60 to 90% of the total flow of working fluid, is supplied to the recuperator 12 which recovers residual heat from the first fluid flow and transfers the residual heat as described above to the working fluid in the low pressure side of the cycle to preheat the working fluid before it is further preheated by the recuperator 30 as described above. The second fluid flow comprising the remaining proportion of the total flow of working fluid is supplied to the heat exchanger 38 of the medium temperature thermal energy store arrangement 32 where heat is transferred to the pressurised water as it flows from the relatively cool water storage tank 36 to the relatively warm water storage tank 34. The split of the working fluid flow is advantageous since the heat capacity of the working fluid in the high pressure side of the closed cycle is larger than that of the low pressure vapour in the low pressure side of the closed cycle.

[0041] The first and second fluid flows output respectively from the recuperator 12 and the medium-temperature thermal energy store arrangement 32 are combined by the second flow divider/combiner valve 42 to once again form a single flow of working fluid.

[0042] In order to close the cycle, it is necessary to reduce the pressure of the working fluid to a sufficient level that it evaporates. The necessary pressure reduction is achieved by firstly expanding the pressurised working fluid in the liquid expander 18. The working fluid is then throttled by the throttle 20 into the wet region before the working fluid is returned to the evaporator 10.

<u>Discharging Cycle - Electricity Generation</u>

[0043] During the discharging cycle illustrated in Figure 2 in which electricity is generated, the system operates as an organic Rankine cycle. In more detail, the condensable organic working fluid is condensed in the condenser 10 by rejecting heat to ambient temperature sea water or river water. The condensed working fluid is then pumped to a supercritical pressure by the liquid pump 18.

[0044] The condensed working fluid is then divided by the second flow divider/combiner valve 42 into first and second fluid flows. The first fluid flow comprising approximately 60 to 90% of the total flow of working fluid is supplied to the recuperator 12 where it is preheated using the residual exhaust heat in the working fluid output from the turbine 22 on the low pressure side of the cycle. The second fluid flow comprising the remaining proportion of the total flow of condensed working fluid is supplied to the heat exchanger 38 of the medium-temperature thermal energy store arrangement 32 where heat is transferred from the pressurised water to the working fluid as the pressurised water flows from the relatively warm water storage tank 34 to the relatively cool water storage tank 36.

[0045] The first and second fluid flows output respectively from the recuperator 12 and the medium-temperature thermal energy store arrangement 32 are combined by the first flow divider/combiner valve 40 to once again form a single flow of working fluid.

[0046] The recombined flow of circulating working fluid is further preheated in the recuperator 30 using the residual exhaust heat in the working fluid output from the turbine 22 on the low pressure side of the cycle. The preheated working fluid is then supplied to the heat exchanger 28 of the high-temperature thermal energy store arrangement 16 where heat is transferred from the molten salt to the circulating working fluid as the molten salt flows from the hot salt storage tank 24 to the relatively warm salt storage tank 26.

[0047] The heated working fluid is expanded in the turbine 22. This drives the electrical generator G to generate electricity. After expansion in the turbine 22, residual waste heat from the working fluid is transferred via the recuperators 30, 12 to the high pressure side of the cycle to preheat the high pressure working fluid in the manner described above. The working fluid is finally returned to the condenser 10 to close the cycle.

[0048] There may be circumstances in which insufficient heat is stored in the high-temperature and medium-temperature thermal energy store arrangements 16, 32 to enable the system illustrated in Figures 1 and 2 to generate electricity during the discharging cycle. In such circumstances, external heat input(s) Q as shown in the system illustrated in Figure 3 can be used to raise the temperature of the molten salt and pressurised water in the high-temperature and medium-temperature thermal energy store arrangements 16, 32 for recovery during the discharging cycle. The external heat input(s) Q could, for example, be provided by means of a combustion process, or a renewable energy source such as a geothermal or solar heat source, and each of the high-temperature and medium-temperature thermal energy store arrangements 16, 32 includes a heat exchange arrangement 44, 46 via which thermal energy from the external heat input (s) can be transferred respectively to the molten salt and pressurised water.

[0049] The use of a first, high-temperature, thermal energy store arrangement 16 and a comparatively smaller scale second, medium-temperature, thermal energy store arrangement 32 is advantageous because the circulating working fluid has different heat capacities in the low pressure and supercritical high pressure conditions. In particular, the high pressure working fluid possesses too much thermal energy during the charging cycle and requires too much thermal energy during the discharging cycle. The second, medium-temperature, thermal energy store arrangement 32 overcomes this problem by temporarily storing excess thermal energy during charging and supplying it during discharging. This arrangement, along with the recuperator 30, allows the operating temperature of the high-temperature thermal energy store arrangement 16 to be shifted to a level that is favourable with regard to the storage medium. For example, in embodiments in which the condensable organic working fluid is butane, the operating temperature range can be shifted from approximately 100°C in the relatively warm salt storage tank 26 and 270°C in the hot salt storage tank 24 to values of approximately 290°C and 450°C respectively. These higher temperatures are within the operating range of economic molten salts used in solar thermal power plants, whereas operating temperatures in the range of 100°C to 270°C can only be achieved using molten salts with the expensive addition of lithium nitrates.

[0050] As indicated above, the thermal energy storage system according to embodiments of this disclosure operates as an organic Rankine cycle during the discharging cycle and such cycles are well-known in the industry. The system also operates as a heat pump during the charging cycle. Accordingly, the density of the working fluid vapour after the evaporator 10 is an important factor in order to have a reasonably sized compressor 14. It is believed that methane and ethane have a too high pressure and density for a reasonable range of evaporation temperatures. N-butane and iso-butane have a pressure in the order of 1 to 2 bar with densities of 3 to 4 kg/m^3 at evaporation temperature (2 to 3 bar in front of the compressor 14). N-pentane and iso-pentane have a lower vapour pressure and density that is, however, comparable at high temperature to the properties of the butanes at low temperature. Propane has a higher vapour pressure and density that is, however, comparable at low temperature to the properties of the butanes at high temperatures.

[0051] In summary it appears that a C_3 alkane (propane), C_4 alkanes (n-butane or isobutane) or C_5 alkanes (n-pentane or isopentane) could be used as the organic working fluid for the system according to embodiments of the present disclosure. The selection could be adapted to the evaporation temperature. Assuming river or sea water cooling/heating as described above, a lighter blend (C_3 or C_4) could be used in winter and a heavier blend (C_4 or C_5) could be used in summer.

[0052] Although exemplary embodiments have been described in the preceding paragraphs, it should be understood that various modifications may be made to those embodiments without departing from the scope of the appended claims. Thus, the breadth and scope of the claims should not be limited to the above-described exemplary embodiments. Each feature disclosed in the specification, including the claims and drawings, may be replaced by alternative features serving the same, equivalent or similar purposes, unless expressly stated otherwise.

[0053] For example, the recuperator 30 (third heat exchanger) is not essential and could be omitted from the embodiments described with reference to Figures 1 to 3. The hot and warm salt storage tanks 24, 26 could be replaced by a single thermocline storage tank. Likewise, the warm and cool water storage tanks 34, 36 could alternatively or additionally be replaced by a single thermocline storage tank.

[0054] Unless the context clearly requires otherwise, throughout the description and the claims, the words "comprise", "comprising", and the like, are to be construed in an inclusive as opposed to an exclusive or exhaustive sense; that is to say, in the sense of "including, but not limited to".

Claims

1. A thermal energy storage system comprising:

a first thermal energy store arrangement (16) having an upper operating temperature ($T1_{max}$) and a lower operating temperature ($T1_{min}$);
a second thermal energy store arrangement (32) having an upper operating temperature ($T2_{max}$) and a lower operating temperature ($T2_{min}$), wherein the upper operating temperature ($T2_{max}$) of the second thermal energy store arrangement is less than or equal to the lower operating temperature ($T1_{min}$) of the first thermal energy store arrangement (16); and
a working fluid circuit (8) containing a working fluid for circulation through the first and second thermal energy store arrangements (16, 32);
the thermal energy storage system having a charging cycle in which thermal energy is transferred from the circulating working fluid to the first and second thermal energy store arrangements (16, 32) and a discharging cycle in which thermal energy is transferred from the first and second thermal energy store arrangements (16, 32) to the circulating working fluid.

2. A system according to claim 1, wherein the first thermal energy store arrangement (16) utilises a first thermal energy storage medium and the second thermal energy store arrangement (32) utilises a second thermal energy storage medium which is different from the first thermal energy storage medium.

3. A system according to claim 2, wherein the first thermal energy storage medium is a molten salt or a mixture of molten salts and the second thermal energy storage medium is pressurised water.

4. A system according to claim 2 or claim 3, wherein the first thermal energy store arrangement (16) comprises high-temperature and medium-temperature storage tanks (24, 26) for the first thermal energy storage medium and a heat exchange device (28) arranged between the storage tanks (24, 26) to transfer thermal energy between the circulating working fluid and the first thermal energy storage medium as it flows through the heat exchange device (28) between the storage tanks (24, 26), the high temperature storage tank (24) having an upper operating temperature ($T1_{max}$) between 300 and 600°C and the medium temperature storage tank (26) having a lower operating temperature ($T1_{min}$) between 200 and 400°C.

5. A system according to any of claims 2 to 4, wherein the second thermal energy store arrangement (32) comprises medium-temperature and low-temperature storage tanks (34, 36) for the second thermal energy storage medium and a heat exchange device (38) arranged between the storage tanks (34, 36) to transfer thermal energy between the circulating working fluid and the second thermal energy storage medium as it flows through the heat exchange device (38) between the storage tanks (34, 36), the medium temperature storage tank (34) having an upper operating temperature ($T2_{max}$) between 100 and 300°C and the low temperature storage tank (36) having a lower operating temperature ($T2_{min}$) between 20 and 100°C.

6. A system according to any preceding claim, further comprising first and second heat exchangers (10, 12) and a compressor (14), wherein:

the first and second heat exchangers (10, 12) are operable during the charging cycle to transfer thermal energy to the circulating working fluid to heat the working fluid prior to compression in the compressor (14); and
the compressor (14) is operable during the charging cycle to compress the heated working fluid prior to circulation through the first and second thermal energy store arrangements (16, 32).

7. A system according to claim 6, wherein the system includes a first flow divider/combiner valve (40) between the first and second thermal energy store arrangements (16, 32) which is operable during the charging cycle to divide the circulating working fluid into:

a first fluid flow to the second heat exchanger (12) to transfer thermal energy from the first fluid flow to the circulating working fluid output from the first heat exchanger (10); and
a second fluid flow to the second thermal energy store arrangement (32) to transfer thermal energy from the second fluid flow to the second thermal energy store arrangement (32).

8. A system according to claim 7, wherein the system includes a second flow divider/combiner valve (42) which is operable during the charging cycle to combine the first and second fluid flows respectively from the second heat exchanger (12) and the second thermal energy store arrangement (32).

9. A system according to any of claims 6 to 8, further comprising a third heat exchanger (30) operable during the charging cycle to transfer thermal energy to the circulating working fluid output from the second heat exchanger (12) prior to compression of the heated circulating working fluid in the compressor (14).

10. A system according to claim 9, wherein the third heat exchanger (30) is operable during the charging cycle to transfer thermal energy from the circulating working fluid output from the first thermal energy store arrangement (16) to the circulating working fluid output from the second heat exchanger (12) prior to compression of the heated circulating working fluid in the compressor (14).

11. A system according to any of claims 8 to 10, further comprising a turbine (22) operable during the discharging cycle to expand heated circulating working fluid, wherein the second flow divider/combiner valve (42) is operable during the discharging cycle to divide the circulating working fluid into:

a first fluid flow to the second heat exchanger (12) to enable the transfer of residual thermal energy from the circulating working fluid output from the turbine (22) to the first fluid flow; and
a second fluid flow to the second thermal energy store arrangement (32) to enable the transfer of thermal energy from the second thermal energy store arrangement (32) to the second fluid flow.

12. A system according to claim 11, wherein the first flow divider/combiner valve (40) is operable during the discharging cycle to combine the first and second fluid flows output respectively from the second heat exchanger (12) and the second thermal energy store arrangement (32).

13. A system according to claims 9, 11 and 12 or claims 9, 10, 11 and 12, wherein the third heat exchanger (30) is operable during the discharging cycle to transfer residual thermal energy from the circulating working fluid output from the turbine (22) to the combined flow of circulating working fluid output from the first flow divider/combiner valve (40).

14. A system according to any preceding claim, wherein the working fluid is a condensable working fluid, preferably a condensable organic working fluid.

15. A system according to claim 14, wherein the condensable organic working fluid is an alkane, preferably a C_3 alkane or a C_4 alkane or a C_5 alkane.

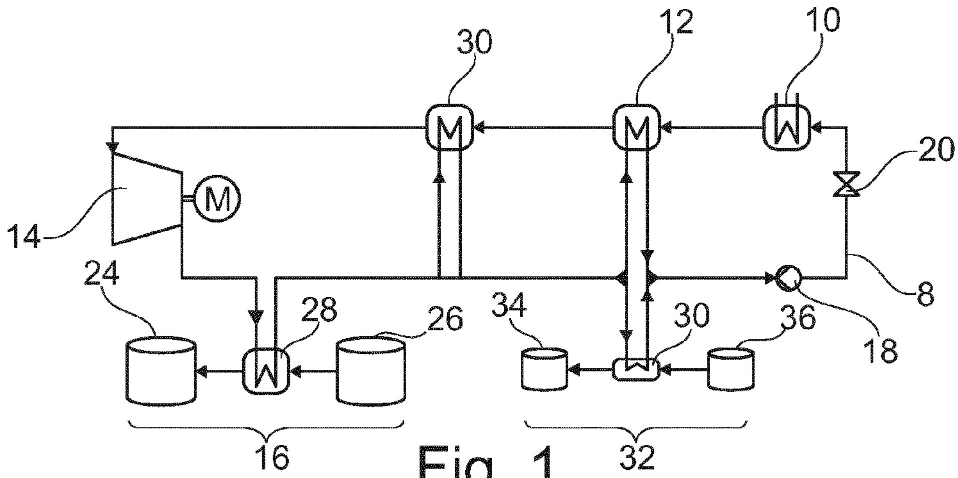

Fig. 1

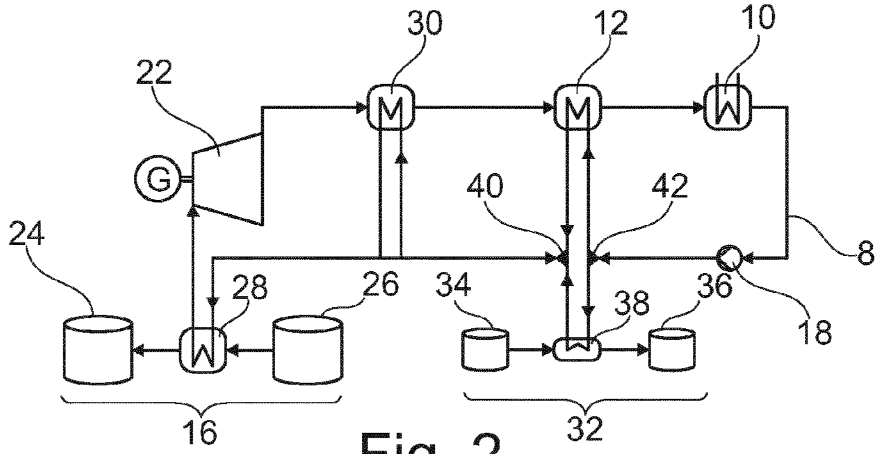

Fig. 2

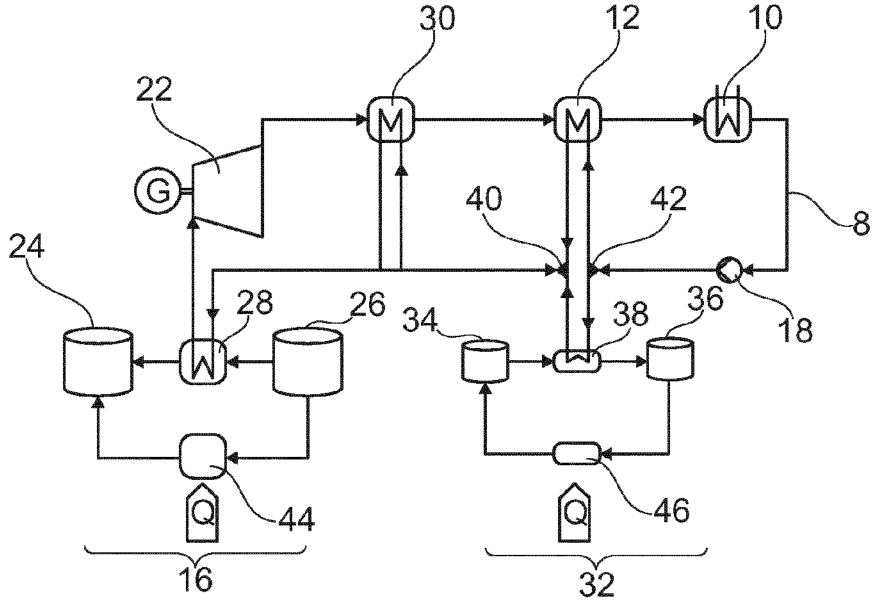

Fig. 3

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EUROPEAN SEARCH REPORT

Application Number
EP 12 19 5864

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The present search report has been drawn up for all claims

1

Place of search	Date of completion of the search	Examiner
Munich	17 April 2013	Röberg, Andreas

CATEGORY OF CITED DOCUMENTS

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