A system and method are provided for thermoelectric energy storage. A thermoelectric energy storage system having at least one hot storage unit is provided. In an exemplary embodiment, each hot storage unit includes a hot tank and a cold tank connected via a heat exchanger and containing a thermal storage medium. The thermoelectric energy storage system also includes a working fluid circuit for circulating working fluid through each heat exchanger for heat transfer with the thermal storage medium. Improved roundtrip efficiency is achieved by minimizing the temperature difference between the working fluid and the thermal storage medium in each heat exchanger during heat transfer. Exemplary embodiments realize this improved roundtrip efficiency through modification of thermal storage media parameters.
FIG. 5

- (a)
- (b)
- (c)
- (d)
- (e)
- (f)
FIG. 7

Temperature [°C] vs. Enthalpy [kJ/kg]

Points: D1, D2, D3, Y1, Y2, F1, F2, F3, C2, G

Legend:
- Solid line: D1
- Dotted line: D2
- Dashed line: D3
- Dashed-dotted line: Y1, Y2, F1, F2, F3
- Solid line with asterisk: C2, G
THERMOELECTRIC ENERGY STORAGE SYSTEM AND METHOD FOR STORING THERMOELECTRIC ENERGY

RELATED APPLICATIONS

[0001] This application claims priority as a continuation application under 35 U.S.C. §120 to PCT/EP2009/058475, which was filed as an International Application on Jul. 6, 2009, designating the U.S., and which claims priority to European Application 0816052.0.6 filed in Europe on Jul. 16, 2008. The entire contents of these applications are hereby incorporated by reference in their entireties.

FIELD

[0002] The present disclosure relates generally to the storage of electric energy. More particularly, the present disclosure relates to a system and method for storing electric energy in the form of thermal energy in a thermal energy storage system.

BACKGROUND INFORMATION

[0003] Base load generators such as nuclear power plants and generators with stochastic, intermittent energy sources, such as wind turbines and solar panels, generate excess electrical power during times of low power demand. Large-scale electrical energy storage systems are a means of diverting this excess energy to times of peak demand and balancing overall electricity generation and consumption.

[0004] In EP1577548, the applicant described the idea of a thermo-electric energy storage (TEES) system. A TEES converts excess electricity to heat, stores the heat, and converts the heat back to electricity, when necessary. Such an energy storage system is robust, compact, site independent and is suited to the storage of electrical energy in large amounts. Thermal energy can be stored in the form of sensible heat via a change in temperature or in the form of latent heat via a change of phase or a combination of both. The storage medium for the sensible heat can be a solid, liquid, or a gas. The storage medium for the latent heat occurs via a change of phase and can involve any of these phases or a combination of the phases in series or in parallel.

[0005] All electric energy storage technologies inherently have a limited round-trip efficiency. Thus, for every unit of electrical energy used to charge the storage, only a certain percentage is recovered as electrical energy upon discharge. The rest of the electrical energy is lost. If, for example, the heat being stored in a TEES system is provided through resistor heaters, it has approximately 40% round-trip efficiency. The efficiency of thermo-electric energy storage is limited for various reasons rooted in the second law of thermodynamics. Firstly, the conversion of heat to mechanical work is limited to the Carnot efficiency. Secondly, the coefficient of performance of any heat pump declines with increased temperature difference between the input level and the output level. Thirdly, any heat flow from a working fluid to a thermal storage and vice versa requires a temperature difference in order to happen. This fact inevitably degrades the temperature level and thus the capability of the heat to do work.

[0006] It is noted that many industrial processes involve provision of thermal energy and storage of the thermal energy. Examples are refrigeration devices, heat pumps, air conditioning and the process industry. In solar thermal power plants, heat is provided, possibly stored, and converted to electrical energy. However, all these applications are distinct from TEES systems because they are not concerned with heat for the exclusive purpose of storing electricity.

[0007] It is known in the art that heat can be provided to the thermal storage unit through a heat pump. For example, a Stirling machine (for reference, see U.S. Pat. No. 3,080,706, column 2, lines 22-30). Also, WO 2007/134466 discloses a TEES system having an integrated heat pump.

[0008] A heat pump requires work to move thermal energy from a cold source to a warmer heat sink. Since the amount of energy deposited at the hot side is greater than the work required by an amount equal to the energy taken from the cold side, a heat pump will "multiply" the heat as compared to resistive heat generation. The ratio of heat output to work input is called a coefficient of performance, and the ratio is a value larger than one. In this way, the use of a heat pump will increase the round-trip efficiency of a thermo-electric energy storage system. The round-trip efficiency is the amount of electricity provided from the storage divided by the amount of electricity provided to the storage.

[0009] U.S. Pat. No. 4,089,744 discloses a method of thermal energy storage by means of reversible heat pumping. Excess electrical output is stored in the form of sensible heat by using it to raise the temperature level of a heat storage fluid. In this scheme, the source of low level heat is stored hot water, which also serves as the working fluid in the heat pump and the turbine cycles. A thermodynamic analysis, such as the type of analysis shown in FIG. 6 of the "744 patent, shows that the efficiency of schemes equivalent to that of U.S. Pat. No. 4,089,744 is limited to about 50%.

[0010] In view of this background, exemplary embodiments of the present disclosure provide an efficient thermo-electric energy storage having a round-trip efficiency of, for example, greater than 55%.

SUMMARY

[0011] An exemplary embodiment provides a thermo-electric energy storage system for providing thermal energy to a thermodynamic machine for generating electricity. The exemplary system includes a heat exchanger, and a hot storage unit which is connected to the heat exchanger and which contains a thermal storage medium. The exemplary system also includes a working fluid circuit configured to circulate a working fluid through the heat exchanger for heat transfer with the thermal storage medium contained in the hot storage unit. The working fluid circuit is also configured to minimize a temperature difference between the working fluid and the thermal storage medium in the hot storage unit during heat transfer.

[0012] An exemplary embodiment also provides a method for storing thermo-electric energy in a thermo-electric energy storage system. The exemplary method includes charging a hot storage unit by providing heat via a heat exchanger to a thermal storage medium by compressing a working fluid. The exemplary method also includes discharging the hot storage unit by expanding the working fluid heated via the heat exchanger from the thermal storage medium through a thermodynamic machine. In addition, the exemplary method includes modifying thermal storage medium parameters to ensure that a temperature difference between the working fluid and the thermal storage medium is minimized during charging and discharging.

BRIEF DESCRIPTION OF THE DRAWINGS

[0013] Additional refinements, advantages and features of the present disclosure are described in more detail below with reference to exemplary embodiments illustrated in the drawings, in which:
FIG. 1 shows an excerpt of a Substation Automation (SA) system, and a simplified schematic diagram of an exemplary thermo-electric energy storage system according to an embodiment of the present disclosure;

FIG. 2 is an enthalpy-pressure diagram of the heat pump cycle and the turbine cycle in an exemplary TEES system according to an embodiment of the present disclosure;

FIG. 3 is a schematic illustration of a cross-section through a heat pump cycle portion of an exemplary TEES system according to an embodiment of the present disclosure;

FIG. 4 is a schematic illustration of a cross-section through a turbine cycle portion of an exemplary TEES system according to an embodiment of the present disclosure;

FIGS. 5(a)-5(f) depict simplified enthalpy-temperature diagrams of the working fluids and thermal storage fluids in exemplary heat exchangers during charging and discharging, according to an embodiment of the present disclosure;

FIG. 6 shows an enthalpy-temperature diagram of the heat transfer from the cycles in an exemplary TEES system according to an embodiment of the present disclosure; and

FIG. 7 shows an enthalpy-temperature diagram of the heat transfer from the cycles in an optimized scenario in an exemplary TEES system according to an embodiment of the present disclosure.

For consistency, the same reference numerals are used to denote similar elements or similarly functioning elements illustrated throughout the drawings.

DETAILED DESCRIPTION

Exemplary embodiments of the present disclosure provide a thermo-electric energy storage system for converting electrical energy into thermal energy to be stored and converted back to electrical energy with an improved round-trip efficiency. An exemplary embodiment provides a thermo-electric energy storage system for providing thermal energy to a thermodynamic machine for generating electricity. Another exemplary embodiment provides a method for storing thermo-electric energy in a thermo-electric energy storage system.

According to an exemplary embodiment of the present disclosure, a thermo-electric energy storage system is provided which includes a hot storage unit which is in connection with a heat exchanger and which contains a thermal storage medium. The exemplary system also includes a working fluid circuit for circulating a working fluid through the heat exchanger for heat transfer with the thermal storage medium. The temperature difference between the working fluid and the thermal storage medium in the hot storage unit is minimized during heat transfer.

When the thermo-electric energy storage system is in a charging (or “heat pump”) cycle, the thermodynamic machine includes a turbine, and when the thermo-electric energy storage system is in a discharging (or “turbine”) cycle, the thermodynamic machine includes a compressor.

According to an exemplary embodiment, the hot storage unit can include at least two hot storage units, where each hot storage unit is in connection with a respective heat exchanger and contains a thermal storage medium.

In accordance with an exemplary embodiment, the heat exchanger or heat exchangers are common to both the charging and discharging cycles. However, it is also possible that there are separate heat exchangers for the charging and discharging cycles. Two or more heat exchangers utilized in series can be connected hydraulically, for example.

The thermal storage medium may be a liquid, and a flow rate of the thermal storage medium may be modified such that the temperature difference between the working fluid and the thermal storage medium in each hot storage unit is minimized during heat transfer.

According to an exemplary embodiment, the thermal storage medium may be a solid or a liquid. The particular exemplary embodiment illustrated in FIGS. 3 and 4 of the accompanying description shows an example in which the thermal storage medium is a liquid.

In accordance with an exemplary embodiment, a single working fluid circuit containing a single type of working fluid can be utilized for both the charging and discharging cycles. However, it is also possible for there to be separate working fluid circuits for the charging and discharging cycles. Further, each separate working fluid circuit can contain a different type of working fluid.

According to an exemplary embodiment, the temperature of the thermal storage medium at entry and exit points of each connected heat exchanger can be modified such that the temperature difference between the working fluid and the thermal storage medium in each hot storage unit is minimized during heat transfer.

In accordance with an exemplary embodiment, at least one of the hot storage units may contain a different type of thermal storage medium such that the temperature difference between the working fluid and the thermal storage medium in each hot storage unit is minimized during heat transfer.

In accordance with an exemplary embodiment, the hot storage unit or units include a thermal storage medium for sensible heat storage and a phase change storage medium for latent heat storage, which are arranged such that the temperature difference between the working fluid and the thermal storage medium in each heat exchanger unit is minimized during heat transfer.

In accordance with an exemplary embodiment, the temperature difference between the working fluid and the thermal storage medium in each hot storage unit can be less than 50°C during heat transfer, for example.

In accordance with an exemplary embodiment of the present disclosure, a method is provided for storing thermo-electric energy in a thermo-electric energy storage system. The exemplary method includes charging a hot storage unit by providing heat via a heat exchanger to a thermal storage medium by compressing a working fluid. The exemplary method also includes discharging the hot storage unit by expanding the working fluid heated via the heat exchanger from the thermal storage medium through a thermodynamic machine. In addition, the exemplary method includes modifying the thermal storage media parameters to ensure the temperature difference between the working fluid and the thermal storage medium is minimized during charging and discharging.

In accordance with an exemplary embodiment, the step of modifying the thermal storage media parameters can include modifying the flow rate of the thermal storage medium.

In accordance with an exemplary embodiment, the step of modifying the thermal storage media parameters can include modifying the initial temperature and final temperature of the thermal storage medium.
In accordance with an exemplary embodiment, the step of modifying the thermal storage media parameters can include modifying the type of thermal storage medium.

FIG. 1 depicts a schematic diagram of an exemplary TEES system 10 in accordance with an embodiment of the present disclosure. The TEES system 10 includes a heat storage 12 and a cold storage 14 which are coupled to each other by means of a heat pump cycle system 16 and a turbine cycle system 18. The hot storage 12 contains a thermal storage medium. According to an exemplary embodiment, the cold storage 14 can be a heat sink, for example. Both the heat pump cycle and the turbine cycle contain a working fluid.

The heat pump cycle system 16 includes, in the flow direction of the working fluid, an evaporator 20, a compressor train 22, a heat exchanger 24, and an expansion valve 26. The turbine cycle system 18 includes, in the flow direction of the working fluid, a feed pump 28, a heat exchanger 30, a turbine 32, and a condenser 34. The heat exchangers 24, 30 in both the heat pump cycle system 16 and the turbine cycle system 18, respectively, are arranged to exchange heat with the hot storage 12. The evaporator 20 and the condenser 34 in the heat pump cycle system 16 and the turbine cycle system 18, respectively, are arranged to exchange heat with the cold storage 14.

The cold storage 14 is a heat reservoir at any temperature lower than the hot storage temperature. However, the cold storage temperature may be higher or lower than the ambient temperature. For example, the cold storage may be another heat sink such as cooling water or air from the ambient. In an alternative embodiment, the turbine and compressor trains 22, 32 may be thermodynamic machines based on positive displacement such as reciprocating or rotary expanders or compressors.

In operation, the working fluid flows around the TEES system 10 in the following manner. The working fluid in the compressor 22 is initially in vapor form, and surplus electrical energy is utilized to compress and heat the working fluid. The working fluid is fed through the heat exchanger 24 where the working fluid discards heat into the storage medium of the hot storage 12. The compressed working fluid exits the heat exchanger 24 and enters the expansion valve 26. Here, the working fluid is expanded to the lower pressure of the evaporator 20. The working fluid flows from the expansion valve into the evaporator 20 where the working fluid is heated to vaporization. This is realized using available heat from the cold storage 14.

In the condenser 34, working fluid is condensed by exchanging heat with the cold storage 14. The condensed working fluid exits the condenser 34 via an outlet and is pumps through the heat exchanger 30 at the hot storage 12 via the feed pump 28. Here, the working fluid is heated, evaporated, and overheated from the stored heat from the storage medium in the hot storage 12. The working fluid exits the heat exchanger 30 and enters the turbine 32 where the working fluid is expanded to thereby cause the turbine to generate electrical energy.

The expansion valve 26, the evaporator 20, and the compressor 22 are in operation during a period of charging, or the "heat pump cycle". Similarly, the turbine 32, the condenser 34 and the feed pump 28 are in operation during a period of discharging, or the "turbine cycle". The hot storage 12 is in operation at all times, i.e., during charging, storage, and discharging. These two cycles can be clearly shown in an enthalpy-pressure diagram, such as FIG. 2.

The solid-line cycle shown in FIG. 2 represents the heat pump cycle that is charging the hot storage, and the heat pump cycle follows a counter-clockwise direction as indicated by the arrows. The working fluid is assumed to be water for this exemplary embodiment. The heat pump cycle starts in the evaporator at point A where steam is evaporated to form vapor using heat from the cold storage (transition A→B1 in FIG. 2). In the next stage of the heat pump cycle, the vapor is compressed utilizing electrical energy in two stages from point B1 to C1 and D2 to C2. Where compression occurs in two stages, this is a consequence of the compressor train 22 including two individual units. In between these two compression stages, the working fluid is cooled from point C1 to B2. The hot, compressed, overheated vapor exits the compression train 22 at point C2 where it is cooled down to the saturation temperature at D1, condensed at D2, and further cooled down to point D3. This cooling down and condensation is realized by transferring heat from the working fluid into the hot storage 12 thereby storing the heat energy. The cooled working fluid is returned to its initial low pressure state at point A via the expansion valve 26.

The dotted-line cycle shown in FIG. 2 represents the Rankine turbine cycle that is discharging the hot storage, and the cycle follows a clockwise direction as indicated by the arrows. The Rankine turbine cycle starts at point E, where the pump 28 is utilized to pump the working fluid in its liquid state from point E to F1. Next, from point F1 to point G, the working fluid receives the heat from the thermal storage medium. In detail, the heat is transferred from the thermal storage medium to the working fluid causing the working fluid to heat up at F2, to boil at F3, and attain a certain degree of superheat at G. The superheated working fluid vapor at point G is expanded down to point H in a mechanical device such as a turbine to generate electricity. Following the expansion, the working fluid enters the condenser 34 where it is condensed to its initial state at point E by exchanging heat with the cold storage 14.

The roundtrip efficiency of the complete energy storage process, that is the heat pump cycle and the Rankine turbine cycle, is calculated in the following manner; the work provided by the turbine expansion divided by the work used in the heat pump compressor:

\[ \frac{h_{G}-h_{H}}{h_{C2}-h_{H}+h_{C1}-h_{B1}} \]

where the letter h denotes the enthalpy of the corresponding point. For the exemplary conditions depicted in FIG. 2, the roundtrip efficiency is 50.8%. It is not possible from the enthalpy-pressure diagram alone to judge if this is a particularly efficient TEES system, or how it could be improved in efficiency.

With reference to the exemplary TEES system illustrated in FIG. 1, the heat exchanger 24 in the heat pump cycle components 16 and the heat exchanger 30 in the turbine cycle components 18 may include several individual heat exchangers arranged in series, as illustrated in FIGS. 3 and 4, respectively.

FIG. 3 depicts a simplified schematic diagram of the heat pump cycle components 16 in a thermoelectric energy...
storage system 10 according to an exemplary embodiment of the present disclosure. Here, three individual hot storage units x, y, z are arranged in series. Each hot storage unit x, y, z comprises a heat exchanger 36, 38, 40 in connection with a storage tank pair 42, 44, 46, respectively. Each storage tank pair comprises a cold tank and a hot tank wherein the flow of the thermal storage medium is from the cold tank to the hot tank via the associated heat exchanger. The three hot storage units in FIG. 3 are denoted x, y and z from left to right in the diagram. In the present embodiment, the heat exchangers are counterflow heat exchangers, and the working fluid of the cycle is water.

In operation, the heat pump cycle components 16 of FIG. 3 perform essentially in a similar manner as the heat pump cycle components 16 of the TEES system 10 described with respect to FIGS. 1 and 2. In addition, the working fluid flows through the additional two separate heat exchangers. In the exemplary embodiment shown in FIG. 3, in the direction of flow of the working fluid, the initial and final temperatures of the working fluid as it passes through heat exchanger 40 are 510°C and 270°C, through heat exchanger 38 are 270°C and 270°C, and through heat exchanger 36 are 270°C and 100°C. Thus, an overall temperature drop of 410°C is achieved.

The characteristics of the working fluid (shown as a solid line) and thermal storage medium (shown as a dashed line) of each of the three heat exchangers 36, 38, 40 and the associated storage tank pairs 42, 44, 46 during discharging are shown in FIG. 5 in the enthalpy-temperature graphs a), b) and c), respectively. The temperature of the thermal storage medium in each stage is increasing, while the temperature of the working fluid decreases only in stages a) and c).

FIG. 4 depicts a simplified schematic diagram of the turbine cycle components 18 in a thermoelectric energy storage system 10 according to an exemplary embodiment of the present disclosure. Here, the arrangement of three individual hot storage units x, y, z, arranged in series, are the same units shown in FIG. 3. Again, each storage tank pair 42, 44, 46 includes a hot tank and a cold tank. However, in the exemplary embodiment of FIG. 4, the flow of the thermal storage medium is from the hot tank to the cold tank via the heat exchanger.

In operation, the turbine cycle components 18 of FIG. 4 perform essentially in a similar manner as the turbine cycle components of the TEES system described with respect to FIGS. 1 and 2. In addition, the working fluid flows through the additional two separate heat exchangers. In the exemplary embodiment shown in FIG. 4, in the direction of flow of the working fluid, the initial and final temperatures of the working fluid as it passes through heat exchanger 36 are 80°C and 240°C, through heat exchanger 38 are 240°C and 240°C, and through heat exchanger 40 are 240°C and 490°C. Thus, an overall temperature increase of 410°C is achieved.

When the heat pump cycle components 16 are in operation, then the working fluid conduit for the heat pump cycle is coupled to the hot storage units x, y, z. When the turbine cycle components 18 are in operation, then the working fluid conduit for the turbine cycle is coupled to the hot storage units x, y, z, instead. In this way, the turbine cycle obtains thermal energy from the hot storage units that was deposited by the heat pump cycle.

The characteristics of the working fluid (shown as a solid line) and thermal storage medium (shown as a dashed line) of each of the three heat exchangers 36, 38, 40 and associated storage tank pairs 42, 44, 46 during discharging are shown in FIG. 5 in the enthalpy-temperature graphs d), e) and f), respectively. The temperature of the thermal storage medium in each stage is decreasing, while the temperature of the working fluid increases only in stages d) and f).

FIG. 6 shows the isobars, e.g., lines of constant pressure, from FIG. 5 a)-f) on a single temperature-enthalpy graph for a particular exemplary embodiment of the present disclosure. Further, the capital letters used are consistent with FIG. 2. Thus, FIG. 6 illustrates the heat transfer process at the three separate hot storage units x, y, z during the charging and discharging of the TEES system 10.

The solid line isobars C2 to D3 represent the heat pump cycle, the dotted line isobars F1 to G represent the Rankine turbine cycle, and the dashed line isobars X1 to X2, Y1 to Y2, Z1 to Z2 represent the thermal storage media in the three hot storage units x, y, z, respectively.

Heat can only flow from a higher to a lower temperature. Consequently, the characteristic isobars for the working fluid during cooling in the heat pump cycle have to be above the characteristic isobars for the thermal storage media, which in turn have to be above the characteristic isobars for the working fluid during heating in the turbine cycle. The slope of these characteristic isobars is defined by the product of the mass flow (kg/s) and heat capacity (J/kg/K) of each thermal storage medium relative to the mass flow of the working fluid. This product is different for each of the three heat transfer subsections: heating/cooling of liquid water in hot storage unit x; boiling/condensation in hot storage unit y; and providing/extracting heat to the supersaturation region in hot storage unit z.

The temperature profiles are stationary in time due to the sensible heat storage in the thermal storage media. Thus, while the volume of thermal storage media in each heat exchanger remains constant, the volume of hot and cold thermal storage media stored in the hot and cold tanks changes. Also, the temperature distribution in the heat exchangers remains constant.

Importantly, exemplary embodiments of the present disclosure determine that the smaller the average temperature difference between the working fluid and the heat storage media during heat transfer, the greater the efficiency of the TEES system. In an enthalpy-temperature graph, this feature is observed as a relatively closer positioning of the characteristic isobars of the charging and discharging cycles, as shown in FIG. 7.

Exemplary embodiments of the present disclosure determine that the thermal storage media may be the same or a different fluid in each hot storage unit x, y and z. Further, exemplary embodiments of the present disclosure determine that the thermal storage media may be at a different temperature in each hot storage unit x, y and z. Also, the flow rate of the thermal storage media within each hot storage unit may differ. Specifically, in order to achieve an optimized roundtrip efficiency of the TEES system, various combinations of the thermal storage media, the initial and final temperature of the thermal storage media and the thermal storage media flow-rates may be utilized.

In the improved efficiency scenario illustrated in FIG. 7, the flow-rate of the thermal storage medium through heat exchanger 36 of hot storage unit y is increased by a factor of three in comparison with the scenario in FIG. 6. (It should be noted that the flow rate in heat exchanger 36, in FIG. 6, was set to an arbitrary rate that was relatively larger than the flow
rate in heat exchangers 36 and 40, but the flow rate was not optimized as in FIG. 7.) A decrease in average temperature differences between the thermal storage medium and the working fluid during heat transfer in heat exchanger 38 of hot storage unit y can be noted. Consequently, a resultant TEES system design has a higher saturation temperature in heat exchanger 38 in the turbine cycle than before (denoted as T' and T" in FIG. 7 in comparison with T' and T" in FIG. 6). This equates to a temperature of 230°C in FIG. 7, in comparison with 200°C in FIG. 6. Consequently, the roundtrip efficiency of the TEES system in the embodiment of FIG. 7 is 61.1% in comparison to an efficiency of 50.8% in FIG. 2.

[0064] In other words, exemplary embodiments of the present disclosure require the temperature difference between the working fluid of the heat pump cycle and the heat storage media, as well as the temperature difference between the working fluid of the turbine cycle and the heat storage media to be relatively small (for example, smaller than 50°C on average). This is achieved through modification of certain TEES parameters as specified above.

[0065] In accordance with another exemplary embodiment of the present disclosure, the three thermal storage media are fluids. For example, these may be three different liquid sensible heat storage media such as water, oil, or molten salts. Also, in accordance with an exemplary embodiment of the present disclosure, the heat exchangers are counterflow heat exchangers, having a minimal approach temperature 10 K (e.g., the minimal temperature difference between the two fluids exchanging heat is 10 K). The expansion device can be a thermostatic expansion valve.

[0066] In accordance with another exemplary embodiment, the heat at the boiling/condensation heat exchanger 38 is transferred to the latent heat of a phase transition of a storage medium enabling an even closer match of the temperature profiles in the boiling/condensation region. An exemplary embodiment uses steam as the working fluid for both the heat pump cycle and the turbine cycle.

[0067] In an alternative exemplary embodiment, there is no cold storage reservoir, but the evaporator and condenser instead use heat from the ambient as an (infinately large) reservoir for the cold side of the heat pump cycle and the turbine cycle. The cold storage of FIG. 1, which is a second heat storage reservoir, has latent heat storage at temperatures around 100°C at the cold side of the heat pump cycle and the turbine cycle. Because of the temperature dependence of the saturation pressure of working fluids such as water, such an additional heat storage reservoir may result in greater economy with respect to the compressor and the turbine. It is envisaged that this economy would more than compensate for the additional cost for this reservoir at moderately long storage times.

[0068] The skilled person will appreciate that the exemplary TEES system, as illustrated in FIGS. 1, 3 and 4, may be realized in several different ways. For example, the hot storage can be constituted by:

[0069] A solid structure with embedded heat exchangers equipped with appropriate means of handling the expansion-contraction of the storage medium with changing temperatures.

[0070] A two-tank molten salt storage system with heat exchangers between the tanks and flow of molten salt from the cold to the hot tank during charging and from the hot to the cold tank during discharging periods.

[0071] A multiple-hot-tank multiple-cold-tank molten salt and liquid heat storage media cascaded at different temperatures between the evaporator operating temperature and the temperature of the heat pump working fluid at the exit of the compression processes.

[0072] A phase change material with a suitable phase change temperature below the condensation temperature of the heat pump working fluid at the high operating pressure and above the boiling point of the turbine cycle working fluid at the high operating pressure.

[0073] Any combination of the above mentioned thermal storage options in series and in parallel.

[0074] Two, three (as shown in FIGS. 3 and 4), four or more hot storage units in the hot storage.

[0075] The skilled person will appreciate that the condenser and the evaporator in the exemplary TEES system may be replaced with a multi-purpose heat exchange device that can assume both roles, since the evaporation for the heat pump cycle and the condensation for the turbine cycle will be carried out in different periods. Similarly, the turbine and the compressor roles can be carried out by the same machinery, referred to herein as a thermodynamic machine, capable of achieving both tasks.

[0076] In the exemplary embodiments described above, The working fluid for is water, due, in part, to the higher efficiencies of a water-based heat pump cycle and turbine cycle, and the amiable properties of water as a working fluid, e.g., no global warming potential, no ozone depletion potential, no health hazards, etc. However, the present disclosure is not limited thereto. For the operation of the present disclosure at ambient temperatures below the freezing point of water, a commercial refrigerant can be chosen as the heat pump working fluid, or a second bottoming heat pump cycle can be cascaded with the water-based cycle to provide the heat of evaporation, for example.

[0077] It will be appreciated by those skilled in the art that the present invention can be embodied in other specific forms without departing from the spirit or essential characteristics thereof. The presently disclosed embodiments are therefore considered in all respects to be illustrative and not restricted. The scope of the invention is indicated by the appended claims rather than the foregoing description and all changes that come within the meaning and range and equivalence thereof are intended to be embraced therein.

What is claimed is:

1. A thermoelectric energy storage system for providing thermal energy to a thermodynamic machine for generating electricity, the system comprising:
   a) a heat exchanger;
   b) a hot storage unit which is connected to the heat exchanger and which contains a thermal storage medium; and
   c) a working fluid circuit configured to circulate a working fluid through the heat exchanger for heat transfer with the thermal storage medium contained in the hot storage unit, and
   where the working fluid circuit is configured to minimize a temperature difference between the working fluid and the thermal storage medium in the hot storage unit during heat transfer.

2. The system according to claim 1, wherein the hot storage unit comprises at least two hot storage units, wherein each of the at least two hot storage units is connected to a respective heat exchanger and contains a thermal storage medium.
3. The system according to claim 1, wherein the thermal storage medium is a liquid, and the working fluid circuit is configured to modify a flow rate of the thermal storage medium such that the temperature difference between the working fluid and the thermal storage medium in the hot storage unit is minimized during heat transfer.

4. The system according to claim 1, wherein the working fluid circuit is configured to modify the temperature of the thermal storage medium at entry and exit points of the heat exchanger such that the temperature difference between the working fluid and the thermal storage medium in the hot storage unit is minimized during heat transfer.

5. The system according to claim 2, wherein at least one of the hot storage units contains a different type of thermal storage medium such that the temperature difference between the working fluid and the thermal storage medium in each hot storage unit is minimized during heat transfer.

6. The system according to claim 1, wherein the temperature difference between the working fluid and the thermal storage medium in the hot storage unit is less than 50°C during heat transfer.

7. A method for storing thermoelectric energy in a thermoelectric energy storage system, comprising:
   - charging a hot storage unit by providing heat via a heat exchanger to a thermal storage medium by compressing a working fluid;
   - discharging the hot storage unit by expanding the working fluid heated via the heat exchanger from the thermal storage medium through a thermodynamic machine; and
   - modifying thermal storage media parameters to ensure that a temperature difference between the working fluid and the thermal storage medium is minimized during charging and discharging.

8. The method according to claim 7, wherein the step of modifying the thermal storage media parameters comprises modifying the flow rate of the thermal storage medium.

9. The method according to claim 7, wherein the step of modifying the thermal storage media parameters comprises modifying an initial temperature and final temperature of the thermal storage medium.

10. The method according to claim 7, wherein the step of modifying the thermal storage media parameters comprises modifying the type of thermal storage medium.

11. The system according to claim 2, wherein the thermal storage medium is a liquid, and the working fluid circuit is configured to modify a flow rate of the thermal storage medium such that the temperature difference between the working fluid and the thermal storage medium in each hot storage unit is minimized during heat transfer.

12. The system according to claim 11, wherein at least one of the hot storage units contains a different type of thermal storage medium such that the temperature difference between the working fluid and the thermal storage medium in each hot storage unit is minimized during heat transfer.

13. The system according to claim 2, wherein the working fluid circuit is configured to modify the temperature of the thermal storage medium at entry and exit points of each connected heat exchanger such that the temperature difference between the working fluid and the thermal storage medium in each hot storage unit is minimized during heat transfer.

14. The system according to claim 13, wherein at least one of the hot storage units contains a different type of thermal storage medium such that the temperature difference between the working fluid and the thermal storage medium in each hot storage unit is minimized during heat transfer.

15. The system according to claim 3, wherein the working fluid circuit is configured to modify the temperature of the thermal storage medium at entry and exit points of each connected heat exchanger such that the temperature difference between the working fluid and the thermal storage medium in each hot storage unit is minimized during heat transfer.

16. The system according to claim 15, wherein the temperature difference between the working fluid and the thermal storage medium in each hot storage unit is less than 50°C during heat transfer.

17. The system according to claim 2, wherein the temperature difference between the working fluid and the thermal storage medium in each hot storage unit is less than 50°C during heat transfer.

18. The method according to claim 8, wherein the step of modifying the thermal storage media parameters comprises modifying an initial temperature and final temperature of the thermal storage medium.

19. The method according to claim 8, wherein the step of modifying the thermal storage media parameters comprises modifying the type of thermal storage medium.

20. The method according to claim 9, wherein the step of modifying the thermal storage media parameters comprises modifying the type of thermal storage medium.

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