

A METHOD AND SYSTEM FOR GENERATING POWER FROM A HEAT SOURCE

FIELD OF THE INVENTION

The present invention relates to a method and system for generating power from a
5 heat source, and in particular to a method and system for generating power from a
geothermal heat source.

The invention has been developed primarily for use in generating power from a
geothermal heat source and will be described hereinafter with reference to this application.
However, it will be appreciated that the invention is not limited to this particular field of
10 use.

BACKGROUND OF THE INVENTION

Any discussion of the prior art throughout the specification should in no way be
considered as an admission that such prior art is widely known or forms part of common
general knowledge in the field.

15 Increased concern regarding the consumption of fossil fuels and reduction of
greenhouse gases has led to research and development focusing on maximising the
efficiency in power generation and on renewable energy resources. One renewable energy
resource is geothermal energy, which is derived from the thermal energy stored deep
within the Earth. Whilst increasing the efficiency of power generation is a common
20 concern for all energy resources, it is of particular interest for geothermal power plants.

The production of power from geothermal energy basically involves extracting
geothermal fluid from a reservoir and converting the thermal energy stored in the
geothermal fluid into mechanical work and then into electricity. Conventional geothermal
power cycles can generally be classified into non-condensing direct steam cycles,
25 condensing direct steam cycles (single flash or double flash), binary cycles and combined
cycles. All but condensing direct steam cycles use a working fluid to exchange heat with
the geothermal heat source and drive the turbine to generate power. Condensing direct

steam cycles are limited to dry-steam geothermal reservoirs, which are much rarer than other geothermal reservoirs, such as hot-water and hot-dry-rock reservoirs.

However, these conventional power cycles were originally designed for large scale power production from fossil fuels, where higher temperature sources are available for heat exchange. Consequently, in these conventional power cycles, the evaporation and condensation of the working fluid both occur at constant temperatures. In the context of geothermal sources, this results in large temperature mismatches between the working fluid and the geothermal heat source during the heat addition or the rejection processes in the thermodynamic cycle. For example, in a binary cycle, the temperature difference between the working and geothermal fluids in the primary heat exchanger could be as high as 80°C to 100°C. In terms of thermodynamics, larger temperature differences in the heat exchange process increase the entropy in the power cycle, thereby reducing the efficiency, particularly the second law efficiencies related to exergy (availability), of the heat exchange process and resulting in poor energy recovery for power generation.

To address this problem, the Kalina cycle employs a multi-component zeotropic mixture of ammonia and water as its working fluid, and additional absorption and distillation equipment to reconstitute the mixture at the low temperature end of the cycle. The multi-component working fluid has a variable phase change temperature during evaporation so that the evaporation of the working fluid occurs over a range of temperatures. Hence, the mixture temperature can more closely match the temperature of the geothermal fluid to increase the amount of thermal energy that is recovered and minimising the entropy in the cycle, thus improving the efficiency of the heat exchange process for low temperature applications, such as geothermal heat sources, as opposed to fossil fuel based power generation.

A disadvantage of the Kalina cycle is that the absorption and distillation equipment added to the cycle creates further complexity to the system, and significantly increases the cost of plant installation compared with other types of power plants. Furthermore, the Kalina cycle has a high sensitivity towards the pressure and composition of the ammonia-water mixture, which limits the operation of the cycle over the whole range of possible geothermal reservoir temperatures and effectively sets a lower limit to the minimum temperature at which a deep geothermal energy source may be commercialised.

SUMMARY OF THE INVENTION

According to one aspect of the invention, there is provided a method of generating power from a heat source, said method including:

compressing a working fluid to increase its temperature;

5 exchanging heat between said working fluid and said heat source to superheat said working fluid;

expanding said superheated working fluid to drive a turbine, thereby reducing its temperature;

condensing said working fluid to further reduce its temperature; and

10 returning said working fluid to said compressing step,

the method further including the step of regenerating the heat of said working fluid, wherein working fluid passing between said compressing step and said heat exchanging step exchanges heat with working fluid passing between said expanding step and said condensing step,

15 wherein said steps are performed in a thermodynamic cycle within a supercritical region above the saturation dome of said working fluid, and wherein said heat regenerating step is performed under isenthalpic conditions to induce continuous heat exchange.

According to another aspect of the invention, there is provided a system for generating power from a heat source, said system including:

20 a compressor for compressing a working fluid to increase its temperature;

a first heat exchanger fluidly connectable to said compressor and said heat source for exchanging heat between said working fluid and said heat source to superheat said working fluid;

25 a turbine fluidly connectable to said first heat exchanger for expanding said superheated working fluid, thereby reducing its temperature;

a second heat exchanger for condensing said working fluid to further reduce its temperature, said second heat exchanger being fluidly connectable to said turbine and said compressor, and

30 a heat regenerator, said regenerator being fluidly interconnectable between said compressor and said first heat exchanger to pre-heat said working fluid prior to entering said first heat exchanger, and fluidly interconnectable between said turbine and said second

heat exchanger to cool said working fluid after exiting said turbine, wherein working fluid passing between said compressor and said first heat exchanger exchanges heat with working fluid passing between said turbine and said condenser,

5 wherein said system operates in a thermodynamic cycle within a supercritical region above the saturation dome of said working fluid, and wherein said regenerator operates under isenthalpic conditions to induce continuous heat exchange.

Preferably, the temperature in the heat regenerating step is controlled to maintain said isenthalpic conditions. Preferably, the heat regenerating step includes controlling the temperature of at least one of the working fluid passing between the compressing step and the heat exchanging step and the working fluid passing between the expanding step and the condensing step. Preferably, the temperature in said heat regenerating step is such that $\frac{\Delta h}{\Delta T} \neq 0$, where Δh is the difference in enthalpy between the working fluid passing between the compressing step and the heat exchanging step and the working fluid passing between the expanding step and the condensing step and ΔT is the difference in temperature between the working fluids.

Preferably, the method further includes the step of monitoring the temperature in said heat regenerating step. Preferably, the heat regenerating step includes controlling the pressure of at least one of the working fluid passing between the compressing step and the heat exchanging step and the working fluid passing between the expanding step and the condensing step, in response to said temperature monitoring step, thereby controlling the temperature of said at least one working fluid.

Preferably, the pressure in said heat regenerating step is controlled to maintain said isenthalpic conditions. Preferably, the heat regenerating step includes controlling the pressure of at least one of the working fluid passing between the compressing step and the heat exchanging step and the working fluid passing between the expanding step and the condensing step. Preferably, the heat regenerating step includes controlling the pressure of the working fluid passing between the compressing step and the heat exchanging step. Preferably, the heat regenerating step includes controlling the upstream pressure of said at least one working fluid to induce a change in the downstream temperature.

Preferably, the heat regenerator includes means for controlling the temperature within said heat regenerator to maintain said isenthalpic conditions. Preferably, the temperature controlling means controls the temperature of at least one of the working fluid passing between the compressor and the first heat exchanger and the working fluid passing
5 between the turbine and the condenser. Preferably, the temperature in said heat regenerator is such that $\frac{\Delta h}{\Delta T} \neq 0$, where Δh is the difference in enthalpy between the working fluid passing between the compressor and the first heat exchanger and the working fluid passing between the turbine and the condenser and ΔT is the difference in temperature between the working fluids.

10 Preferably, the system includes means for monitoring the temperature within said regenerator. Preferably, the temperature monitoring means includes one or more thermocouples located within the regenerator. Preferably, the regenerator includes means for controlling the pressure of at least one of the working fluid passing between the compressor and the first heat exchanger and the working fluid passing between the turbine
15 and the condenser in response to said temperature monitoring means, thereby controlling the temperature of said at least one working fluid.

Preferably, the system includes means for controlling the pressure within said regenerator to maintain said isenthalpic conditions. Preferably, the pressure controlling means controls the pressure of at least one of the working fluid passing between the
20 compressor and the first heat exchanger and the working fluid passing between the turbine and the condenser. Preferably, the pressure controlling means controls the pressure of the working fluid passing between the compressor and the first heat exchanger. Preferably, the pressure controlling means controls the upstream pressure of said at least one working fluid to induce a change in the downstream temperature. Preferably, the pressure
25 controlling means includes at least one or more valves for controlling the pressure of said at least one working fluid. Preferably, the valves are throttle valves.

Preferably, the operating pressure used in the method and of the system is more than the critical point of the working fluid. Preferably, the operating pressure is less than 30 MPa. It is preferred that the operating pressure is less than 15 MPa. Preferably, the

operating pressure is between 8 and 12 MPa. Preferably, the operating temperature used in the method and the system is between 100°C and 200°C.

Preferably, the working fluid has a critical pressure of between 3.3 MPa and 7.5 MPa. Preferably, the working fluid has a critical temperature of between 30°C to 200°C.

5 It is preferred that the working fluid is composed of a single component. Alternatively, a multi-component working fluid can be used if desired.

Preferably, the working fluid includes carbon dioxide, n-Pentane (C_5H_{12}), HFC-245ca ($CF_2H-CF_2-CFH_2$), HFC-245fa ($CF_3-CH_2-CF_2H$), HFC-134a (CH_2F-CF_3), refrigerant 125 and pentafluoroethane (F_4CH_2F).

10 Preferably, the heat source includes a geothermal heat source or a waste heat source. Preferably, the geothermal heat source includes a hot-dry-rock reservoir or a hot-water reservoir. Alternatively, the waste heat source includes cooling water or waste steam from a conventional power station, including a coal, peat, oil, gas or other fossil fuel fired power station.

15 BRIEF DESCRIPTION OF THE DRAWINGS

Preferred embodiments of the invention will now be described, by way of example only, with reference to the accompanying drawings, in which:

Figure 1 is a temperature-entropy phase diagram of a working fluid;

Figure 2 is a temperature-entropy phase diagram illustrating a Rankine cycle;

20 Figure 3 is a temperature-entropy phase diagram illustrating a method according to a first embodiment of the invention;

Figure 4 is a schematic drawing of a system according to the first embodiment;

Figure 5 is a graph illustrating the change in enthalpy against the temperature for a pure working fluid;

25 Figure 6 is a schematic drawing of a regenerator for a system according to a second embodiment of the invention;

Figure 7 is a temperature-entropy phase diagram illustrating a method according to the second embodiment;

Figure 8 is a graph illustrating comparing the thermal conversion efficiency of the second embodiment with a conventional power cycle;

Figure 9 is a graph illustrating comparing the exergetic efficiency of the second embodiment with a conventional power cycle; and

5 Figure 10 is graph illustrating comparing the specific work obtained by the second embodiment with a conventional power cycle.

PREFERRED EMBODIMENTS OF THE INVENTION

Figure 1 is a phase diagram of a pure working fluid in the temperature (T) - entropy (S) domain. In the diagram, dotted lines P are isobars representing constant pressure. The
10 saturation dome A defines the boundary at which the working fluid is in a saturated state. Most conventional power cycles, such as the Rankine cycle shown in Figure 2, operate in or around the saturation phase of the working fluid as defined by the saturation dome A so that any phase change associated with the cycle takes place under constant pressure and temperature. This means that energy from the heat source is lost during phase changes of
15 the working fluid within the saturation dome A. By contrast, the embodiments of the invention avoid these heat losses by operating above the saturation dome A.

Referring to Figures 3 and 4, one embodiment of the invention is illustrated. As best shown in Figure 3, the method of generating power from a heat source includes the step 10 of compressing a working fluid to increase its temperature (states S4-S1) and the
20 step 11 of exchanging heat between the working fluid and the heat source to superheat the working fluid (states S1'-S2). At step 12, the superheated working fluid is expanded to drive a turbine, thereby reducing its temperature (states S2-S3). Then at step 13, the working fluid is condensed to further reduce its temperature (states S3'-S4) before being returned to the compressing step 10. The method further includes a heat regenerating step
25 14 wherein working fluid passing between the compressing step 10 and the heat exchanging step 11 (states S1-S1') exchanges heat with the working fluid passing between the expanding step 12 and the condensing step 13 (states S3-S3'). Thus, in effect the working fluid is pre-heated after the compressing step 10 and prior to exchanging heat with the heat source at step 11, and the working fluid is cooled after the expansion step 12 prior
30 to the condensing step 13. These steps 10 to 14 are performed in a thermodynamic cycle

(S1-S1'-S2-S3-S3'-S4) within the supercritical region SC above the saturation dome A of the working fluid, and the heat regenerating step 14 is performed under isenthalpic conditions to induce continuous heat exchange.

The method illustrated in Figure 3 can be implemented in a system 15 for generating power from a heat source 16, as best shown in Figure 4. The system includes a compressor 17 in the form of a feed pump for compressing a working fluid to increase its temperature, and a first heat exchanger 18 in the form of a high temperature heat exchanger that is fluidly connectable to the compressor 17 and the heat source 16 for exchanging heat between the working fluid and the heat source 16 to superheat the working fluid. A turbine 19 is fluidly connectable to the high temperature heat exchanger 18 for expanding the superheated working fluid, thereby reducing its temperature. A second heat exchanger 20 in the form of a low temperature heat exchanger is fluidly connectable to the turbine 19 and the compressor 17 for condensing the working fluid to further reduce its temperature using a cooler fluid obtained from a suitable cool fluid reservoir 21. A heat regenerator 22 is fluidly interconnectable between the compressor 17 and the first heat exchanger 18 to pre-heat the working fluid prior to entering the high temperature heat exchanger, and fluidly interconnectable between the turbine 19 and the low temperature heat exchanger 20 to cool the working fluid after exiting the turbine 19. The heat regenerator 22 uses the "hot" working fluid passing between the turbine 19 and the condenser 20 (stream 24) to pre-heat the "cool" working fluid passing between the compressor 16 and the high temperature heat exchanger 18 (stream 23). The system 15 operates in a thermodynamic cycle (S1-S1'-S2-S3-S3'-S4) within a supercritical region SC above the saturation dome A of the working fluid, and the regenerator 22 operates under isenthalpic conditions to induce continuous heat exchange.

The thermodynamic cycle (S1-S1'-S2-S3-S3'-S4) is substantially similar to a Rankine cycle, although it is elevated into the supercritical region SC above the saturation dome A of the working fluid and the states S1-S1' and S3-S3' are under isenthalpic conditions.

By conducting the entire thermodynamic cycle (S1-S1'-S2-S3-S3'-S4) within the supercritical region SC above the saturation dome A, this fully supercritical process reduces efficiency losses associated with the high temperature heat exchanger 18 and the

low temperature heat exchanger 20. That is, in states S1-S1'-S2 and S3-S3'-S4, the working fluid is superheated and condensed at variable temperatures rather than at a constant temperature, as in conventional power cycles. This minimises the temperature mismatch between the geothermal fluid and the working fluid and between the cool fluid and the working fluid. Thus, the entropy in the thermodynamic cycle (S1-S1'-S2-S3-S3'-S4) is reduced, and the efficiency of energy conversion is increased.

In addition, the heat regenerator 22 thus reduces the amount of energy required for the heat addition process performed by the compressor 17, thereby improving the efficiency of the thermodynamic cycle (S1-S1'-S2-S3-S3'-S4) within the supercritical region SC above the saturation dome A of the working fluid.

Furthermore, because the heat regenerating step 14 is performed under isenthalpic conditions, there is always a temperature gradient to act as a driving force for continuous heat exchange between the parallel streams 23 and 24 of the working fluids within the regenerator 22.

The importance of performing the regenerating step under isenthalpic conditions is better illustrated in Figure 5, which shows a plot of the enthalpy difference of the "cold" and "hot" working fluid streams (Δh) as a function of temperature (T) for a pure substance as a working fluid, such as carbon dioxide, under supercritical conditions SC. The resulting curve shows a maximum P_m , at which $\frac{\Delta h}{\Delta T} = 0$, where there is no temperature gradient to drive the heat exchange process. Consequently, the existence of a maximum point P_m for Δh is a detriment to the thermodynamic cycle efficiency, since it prevents continual heat exchange and thus reduces the efficiency of energy conversion.

By contrast, in the preferred embodiment, the temperature is controlled so that the enthalpy in the heat regenerating step is kept constant; i.e. under isenthalpic conditions, and the temperature is such that $\frac{\Delta h}{\Delta T} \neq 0$, ensuring that a temperature gradient exists throughout the whole heat regenerating step 14, resulting in the continuation of the heat exchange process between the "cold" and "hot" working fluid streams 23 and 24 passing through the heat regenerator 22.

Moreover, the improved efficiency derived from method and system 15 is optimised by operating at pressures above the critical point of the working fluid. Consequently, by selecting the appropriate type of working fluid, the system 15 can operate within a desired pressure range to meet any constraints in operational costs or safety requirements. In this context, the operating pressure used in the method and the system is preferably less than 15 MPa, even though operating pressures can be up to 30 MPa. In particular, it has been found that operating pressures between 8 and 12 MPa allows for a greater selection of suitable working fluids. Working in these preferred operational pressures permits application of the embodiments of the invention to greater ranges of temperatures, generally between 100°C and 200°C.

Examples of high temperature heat exchangers include boilers and vapour generators. Suitable low temperature heat exchangers air-cooled or water-cooled condensers.

It is contemplated that this embodiment of the invention is applicable to most heat sources, and in particular may be used a part of a “bottoming” cycle utilising the waste heat from a conventional coal fired power station, or other fossil fuel based power stations, such as peat, oil and gas based power stations.

A particularly preferred embodiment of the invention for geothermal heat sources is illustrated in Figures 6 and 7, where corresponding features have been given the same reference numerals. This embodiment takes into account that for geothermal heat sources, the close proximity of the constant pressure lines P in the supercritical region SC may result in small net power outputs, and the maximum cycle temperatures are relatively low compared with other heat sources. In particular, whilst the embodiment has all the features of the system 15 shown in Figure 4, the heat regenerator 22 has a temperature monitoring system (not shown). The temperature monitoring system includes a plurality of thermocouples located at various positions within the regenerator 22, which are operatively connected to a central processing unit (CPU). The regenerator 22 also includes a plurality of throttle valves 26 provided on the piping 27 of the “hot” working fluid stream 24 between the turbine 19 and the condenser 20, as best shown in Figure 6. The throttle valves are operatively connected to the CPU.

As shown in Figure 7, the thermodynamic cycle according to this method can be described in terms of the transition between states of the working fluid as follows:

	States S1-S1'	Heat recovery at constant enthalpy in the heat regenerator 22
5	States S1'-S2	Heat addition at constant pressure in the high temperature heat exchanger 18
	States S2-S3'	Expansion in the turbine 19 for work output
	States S3'-S3	Heat recovery at constant enthalpy in the heat regenerator 22
	States S3-S3''	Heat recovery at constant enthalpy in the heat regenerator 22
10	States S3''-S4	Heat rejection at constant pressure in the low temperature heat exchanger 20
	States S4-S1	Pressure lift of the condensed working fluid in liquid form by the feed pump 17

Operation of the system 15 according to this embodiment will now be described in detail with reference to Figures 4 and 6. The working fluid, in the form of CO₂ in this example, enters the feed pump 17 at state S4 as saturated (or mildly compressed) liquid and is compressed to the operating pressure of the high temperature heat exchanger 18. The temperature of the CO₂ working fluid increases somewhat during this compression process due to slight changes in its specific volume.

The CO₂ working fluid then enters the heat regenerator 22, which generally operates at the same pressure as the high temperature heat exchanger 18, as a compressed liquid at state S1 and leaves as a heated vapour phase at state S1'. This is a heat recovery process for which the required thermal energy is supplied by the hot CO₂ working fluid leaving the turbine at state S3', slightly above the minimum cycle pressure, and which is subjected to variations in pressure to maintain isenthalpic conditions in the regenerator 22.

Next, the vapour at state S1' enters the high temperature heat exchanger 18 in which its temperature is further increased or superheated such that the CO₂ working fluid becomes a superheated vapour upon exit at state S2. The necessary heat for raising the temperature of the working fluid between states S1' and S2 is supplied by a hot geothermal fluid from a suitable geothermal heat source 16.

The superheated vapour at state S2 then enters the turbine 19 where it expands and produces electricity by rotating the shaft of an electric generator (not shown). During the expansion process, the pressure and temperature of the CO₂ working fluid drops until it reaches state S3'. In this state, the CO₂ working fluid still has considerable thermal energy, which is further exploited by passing it through the heat regenerator 22 to heat up the "cold" CO₂ working fluid entering the heat regenerator 22 at state S1. As a result of the heat exchange in the heat regenerator 22 between the relative hot and cold CO₂ working fluids, the temperature of the hot CO₂ working fluid drops to that of state S3" and the temperature of the cold CO₂ working fluid increases to that of state S1'.

In the regenerator 22, the throttle valves 26 are initially fully open and the hot CO₂ stream 24 exchanges heat with the cold CO₂ stream 23. When the temperature monitoring system, for example using one or more thermocouples, identifies that the temperature of a particular area in the regenerator 22 is approaching the value at which $\frac{\Delta h}{\Delta T} = 0$, the CPU sends a signal to the closest upstream throttle valve, for example valve 26a. In response, the throttle valve 26a is partly closed so that the resulting pressure drop throttles the "hot" working fluid stream 24 into a lower pressure, thus ensuring that the heat regenerator 22 remains in constant enthalpy (states S3'-S3 and S3-S3") so that $\frac{\Delta h}{\Delta T} \neq 0$, and the heat exchange process is continual in the regenerator 22. This adjustment of the localised pressure within the regenerator 22 may be repeated several times, depending on the operating conditions, until the "hot" working fluid stream 24 leaves the regenerator 22 at the minimum cycle pressure in the state S3". During this transition from state S3', at the entry into the regenerator 22, to the state S3", at the exit from the regenerator 22, the working fluid passes through state S3.

The CO₂ working fluid exiting the hot side of the heat regenerator 22 in state S3" is then condensed at a constant pressure in the low temperature heat exchanger 20 by rejecting heat to a cooling medium from the cool medium reservoir 21. The condensed CO₂ working fluid finally leaves the low temperature heat exchanger 20 and enters the feed pump 17 to complete the cycle.

Although this embodiment has been described as being particularly applicable to geothermal heat sources, it may also be used for bottoming cycles, where the heat source is the working fluid of the main power cycle. For example, the low temperature steam exiting the main cycle turbine in coal fired power plants.

- 5 Whilst in the embodiments of the invention, carbon dioxide has been used as the working fluid, other working fluids can be selected, including, n-Pentane (C_5H_{12}), HFC-245ca ($CF_2H-CF_2-CFH_2$), HFC-245fa ($CF_3-CH_2-CF_2H$), HFC-134a (CH_2F-CF_3), refrigerant 125 and pentafluoroethane (F_4CH_2F). The comparative properties of each working fluid are set out in Table 1 below.

Working Fluid	Critical Pressure	Critical Temperature	Safety	Ozone Depletion Potential
Carbon Dioxide (CO_2)	7.38 MPa	30.9°C	Non-flammable	≈ 0
n-Pentane (C_5H_{12})	3.37 MPa	196.6°C	Flammable	≈ 0
HFC-245ca ($CF_2H-CF_2-CFH_2$)	3.86 MPa	178.5°C	Non-flammable	≈ 0
HFC-245fa ($CF_3-CH_2-CF_2H$)	3.64 MPa	157.6°C	Non-flammable	≈ 0
HFC-134a (CH_2F-CF_3)	4.1 MPa	101.2°C	Non-flammable	≈ 0
Refrigerant-125 or Pentafluoroethane (F_4H-CH_2F)	3.7 MPa	66.8°C	Non-flammable	≈ 0

Table 1. Properties of Working Fluids suitable for the invention.

- 10 It has been found that other suitable working fluids can be used with critical pressures of between 3.3 MPa and 7.5 MPa, and a critical temperatures of between 30°C to 200°C. Moreover, in other embodiments of the invention a multi-component working fluid

is employed in the method and system, instead of a working fluid composed of a single component.

The thermal conversion efficiency and exergetic efficiency of the second embodiment of Figures 6 and 7 were calculated as a function of the temperature difference between the geothermal fluid at the production and reject wells, ΔT_{geo} . The thermal conversion efficiency and exergetic efficiency of the preferred embodiment was compared with thermal conversion efficiency and exergetic efficiency of conventional power cycles. The working fluid chosen for the second embodiment was carbon dioxide.

As best shown in Figure 8, the thermal conversion efficiency of the second embodiment (designated as CO₂-RGSC) was higher than the thermal conversion efficiency of conventional power cycles. For the second embodiment, the thermal conversion efficiency ranges from 10% to 18% with an average of 16.5%. By contrast, for conventional power cycles, including the Kalina cycle, the thermal conversion efficiency does not change and plateaus around a nominal value between 11% and 12%. Likewise, the exergetic efficiency of the embodiment was higher than the exergetic efficiency of the conventional power cycles as ΔT_{geo} increases, as best shown in Figure 9. Thus, Figures 8 and 9 indicate that the thermal conversion and exergetic efficiencies of the embodiment are far superior to that of thermal conversion and exergetic efficiencies of conventional power cycles.

These improved thermal conversion and exergetic efficiencies imply that more power can be generated from this embodiment per unit of input energy than from a conventional power cycle. This is illustrated in Figure 10, which shows a graph of the specific power (W_{spc}) against ΔT_{geo} . The amount of work extracted from the geothermal fluid, and hence the amount of power generated, was substantially higher for the embodiment than the conventional power cycles, especially as ΔT_{geo} increases.

Thus, the embodiments of the invention provide a method and system of generating power from a heat source, including a geothermal heat source, with improved efficiency over conventional power cycles and without requiring additional equipment that would add to the complexity of the system or cost of installation. Alternatively or additionally, the invention provides a method to extract more heat economically at a lower temperature than

for conventional technologies, thus expanding the number of potential heat sources that may be exploited commercially.

In other embodiments, the temperature in the regenerating step is directly controlled, instead of controlling the pressure of the working fluid. In one embodiment, 5 the temperature or pressure of the working fluid passing between the expanding step and the condensing step (or the working fluid between the turbine and the condenser) is controlled to maintain isenthalpic conditions.

Although the invention has been described with reference to specific examples, it will be appreciated by those skilled in the art that the invention may be embodied in many 10 other forms.

CLAIMS:-

1. A method of generating power from a heat source, said method including:
compressing a working fluid to increase its temperature;
exchanging heat between said working fluid and said heat source to superheat
5 said working fluid;
expanding said superheated working fluid to drive a turbine, thereby reducing
its temperature;
condensing said working fluid to further reduce its temperature; and
returning said working fluid to said compressing step,
10 the method further including the step of regenerating the heat of said working
fluid wherein working fluid passing between said compressing step and said heat
exchanging step exchanges heat with working fluid passing between said expanding
step and said condensing step;
wherein said steps are performed in a thermodynamic cycle within a
15 supercritical region above the saturation dome of said working fluid, and wherein
said heat regenerating step is performed under isenthalpic conditions to induce
continuous heat exchange.
2. The method of claim 1, wherein the temperature in said heat regenerating step is
controlled to maintain said isenthalpic conditions.
- 20 3. The method of claim 2, wherein the heat regenerating step includes controlling the
temperature of at least one of the working fluid passing between the compressing
step and the heat exchanging step and the working fluid passing between the
expanding step and the condensing step.
4. The method according any one of the preceding claims, wherein the temperature in
25 said heat regenerating step is such that $\frac{\Delta h}{\Delta T} \neq 0$, where Δh is the difference in
enthalpy between the working fluid passing between the compressing step and the
heat exchanging step and the working fluid passing between the expanding step and
the condensing step and ΔT is the difference in temperature between the working
fluids.

5. The method according to any one of the preceding claims, further including the step of monitoring the temperature in said heat regenerating step.
6. The method according to claim 5, wherein the heat regenerating step includes controlling the pressure of at least one of the working fluid passing between the compressing step and the heat exchanging step and the working fluid passing
5 between the expanding step and the condensing step, in response to said temperature monitoring step, thereby controlling the temperature of said at least one working fluid.
7. The method of any one of claims 1 to 4, wherein the pressure in the heat regenerating
10 step is controlled to maintain said isenthalpic conditions.
8. The method of claim 7, wherein the heat regenerating step includes controlling the pressure of at least one of the working fluid passing between the compressing step and the heat exchanging step and the working fluid passing between the expanding step and the condensing step .
- 15 9. The method of any one of claims 6 to 8, wherein the heat regenerating step includes controlling the pressure of the working fluid passing between the compressing step and the heat exchanging step.
10. The method of any one of the preceding claims, wherein the operating pressure is more than the critical point of the working fluid.
- 20 11. The method of claim 10, wherein the operating pressure is less than 30 MPa.
12. The method of claim 11, wherein the operating pressure is less than 15 MPa.
13. The method of claim 12, wherein the operating pressure is between 8 and 12 MPa.
14. The method according to any one of the preceding claims, wherein the operating temperature is between 100°C and 200°C.
- 25 15. The method of any one of the preceding claims, wherein the working fluid has a critical pressure of between 3.3 MPa and 7.5 MPa.

16. The method of any one of the preceding claims, wherein the working fluid has a critical temperature of between 30°C to 200°C.
17. The method of any one of the preceding claims, wherein the working fluid is composed of a single component.
- 5 18. The method of claim 17, wherein the working fluid is selected from the group consisting of carbon dioxide, n-Pentane (C_5H_{12}), HFC- 245ca ($CF_2H-CF_2-CFH_2$), HFC-245fa ($CF_3-CH_2-CF_2H$), HFC-134a (CH_2F-CF_3), refrigerant 125 and pentafluoroethane (F_4CH_2F).
19. The method of any one of claims 1 to 16, wherein the working fluid is a multi-
10 component working fluid.
20. The method of any one of the preceding claims, wherein the heat source includes a geothermal heat source or a waste heat source.
21. The method of claim 20, wherein the geothermal heat source includes a hot-dry-rock reservoir or a hot-water reservoir.
- 15 22. The method of claim 20, wherein the waste heat source includes cooling water or waste steam from a conventional power station.
23. The method of claim 22, wherein the conventional power station includes a coal, peat, oil, gas or other fossil fuel fired power station.
24. A system for generating power from a heat source, said system including:
20 a compressor for compressing a working fluid to increase its temperature;
 a first heat exchanger fluidly connectable to said compressor and said heat source for exchanging heat between said working fluid and said heat source to superheat said working fluid;
 a turbine fluidly connectable to said first heat exchanger for expanding said
25 superheated working fluid, thereby reducing its temperature;
 a second heat exchanger for condensing said working fluid to further reduce its temperature, said second heat exchanger being fluidly connectable to said turbine and said compressor, and

a heat regenerator, said regenerator being fluidly interconnectable between said compressor and said first heat exchanger to pre-heat said working fluid prior to entering said first heat exchanger, and fluidly interconnectable between said turbine and said second heat exchanger to cool said working fluid after exiting said turbine, wherein said working fluid passing between said compressor and said first heat exchanger exchanges heat with said working fluid passing between said turbine and said condenser,

wherein said system operates in a thermodynamic cycle within a supercritical region above the saturation dome of said working fluid, and wherein said regenerator operates under isenthalpic conditions to induce continuous heat exchange.

25. The system of claim 24, wherein the heat regenerator includes means for controlling the temperature within said heat regenerator to maintain said isenthalpic conditions.

26. The system of claim 25, wherein the temperature controlling means controls the temperature of at least one of the working fluid passing between the compressor and the first heat exchanger and the working fluid passing between the turbine and the condenser.

27. The system of any one of claims 24 to 26, wherein the temperature in said heat regenerator is such that $\frac{\Delta h}{\Delta T} \neq 0$, where Δh is the difference in enthalpy between the working fluid passing between the compressor and the first heat exchanger and the working fluid passing between the turbine and the condenser and ΔT is the difference in temperature between the working fluids.

28. The system of any one of claims 24 to 27, further including means for monitoring the temperature within said regenerator.

29. The system of claim 28, wherein the temperature monitoring means includes one or more thermocouples located within the regenerator.

30. The system of claim 28 or 29, wherein the regenerator includes means for controlling the pressure of at least one of the working fluid passing between the compressor and the first heat exchanger and the working fluid passing between the turbine and the

condenser in response to said temperature monitoring means, thereby controlling the temperature of said at least one working fluid.

31. The system of any one of claims 24 to 27, wherein the heat regenerator includes means for controlling the pressure within said heat regenerator to maintain said isenthalpic conditions.

32. The system of claim 31, wherein the pressure controlling means controls the pressure of at least one of the working fluid passing between the compressor and the first heat exchanger and the working fluid passing between the turbine and the condenser.

33. The system of any one of claims 30 to 32, wherein the pressure controlling means controls the pressure of the working fluid passing between the compressor and the first heat exchanger.

34. The system of any one of claims 30 to 33, wherein the pressure controlling means controls the upstream pressure of said at least one working fluid to induce a change in the downstream temperature.

35. The system of any one of claims 30 to 34, wherein the pressure controlling means includes at least one or more valves for controlling the pressure of said at least one working fluid.

36. The system of claim 35, wherein the valves are throttle valves.

37. The system of any one of claims 24 to 36, wherein the operating pressure is more than the critical point of the working fluid.

38. The system of claim 37, wherein the operating pressure is less than 30 MPa.

39. The system of claim 38, wherein the operating pressure is less than 15 MPa.

40. The system of claim 39, wherein the operating pressure is between 8 and 12 MPa.

41. The system of any one of claims 24 to 40, wherein the operating temperature is between 100°C and 200°C.

42. The system of any one of claims 24 to 41, wherein the working fluid has a critical pressure of between 3.3 MPa and 7.5 MPa.
43. The system of any one of claims 24 to 42, wherein the working fluid has a critical temperature of between 30°C to 200°C.
- 5 44. The system of any one of claims 24 to 43, wherein the working fluid is composed of a single component.
45. The system of claim 44, wherein the working fluid is selecting from the group consisting of carbon dioxide, n-Pentane (C_5H_{12}), HFC- 245ca ($CF_2H-CF_2-CFH_2$),
10 HFC-245fa ($CF_3-CH_2-CF_2H$), HFC-134a (CH_2F-CF_3), refrigerant 125 and pentafluoroethane (F_4CH_2F).
46. The system of any one of claims 24 to 43, wherein the working fluid is a multi-component working fluid.
47. The system of any one of claims 24 to 46, wherein the heat source includes a geothermal heat source or a waste heat source.
- 15 48. The system of claim 47, wherein the geothermal heat source includes a hot-dry-rock reservoir or a hot-water reservoir.
49. The system of claim 47, wherein the waste heat source includes cooling water or waste steam from a conventional power station
50. The system of claim 49, wherein the conventional power station includes a coal,
20 peat, oil, gas or other fossil fuel fired power station.
51. A method of generating power from a heat source, substantially as herein described with reference to any one of the embodiments of the invention illustrated in the accompanying drawings and/or examples.
52. A system for generating power from a heat source, substantially as herein described
25 with reference to any one of the embodiments of the invention illustrated in the accompanying drawings and/or examples.