



Superior Efficiency
Reduced Costs

Viable Alternative Energy

Kalex
Kalina Cycle Power Systems
For Solar-Thermal Applications

**Kalex LLC's Kalina Cycle
for Biomass Applications**

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Kalex has developed power systems and auxiliary technology that greatly improve the efficiency and reduce the costs of solar-thermal power, reducing the required area of solar collectors for a given power output by as much as 15 to 30%.

Solar-thermal power is a growing segment of the alternative power industry. Solar-thermal power is clean, renewable and has the potential to produce more power than any other alternative power sources, as it is not limited by the availability or cost of the energy source, as is the case with geothermal and biomass power. Solar-thermal power systems tend to be fairly large; 50 MW or larger, with many in excess of 100 MW. At sizes less than 50 MW, solar-thermal is not an attractive alternative to solar photo-voltaic power.

While the actual energy in solar-thermal power, sunlight, is free, the construction and operation of solar-thermal power is far from free; the capital costs of solar-thermal power plants are high. Of these costs, the greatest expense tends to be the cost of the solar collection apparatus, i.e., the solar field.

This means that to be worthwhile, any improvement of the efficiency of solar thermal power systems must result in a decrease of capital costs for the solar-thermal power system as a whole. In particular, improvements that lead to a decrease in the costs of the solar field are desirable.

There are many forms of solar collectors used in solar-thermal applications, but three have emerged as the most common and promising and Kalex LLC has focused its efforts these three:

- Heliostat power towers.
- Solar parabolic troughs.
- Fresnel reflectors.

All solar-thermal power systems involve three sequential processes. First, sunlight is collected and concentrated onto an absorber. Then the concentrated radiated solar energy is converted to heat in the absorber. Last, this heat is used in a power cycle to produce electrical power.

In the process of conversion of solar energy to thermal energy in the absorber, a substantial portion of energy is lost due to thermal radiation from the absorber. These losses, radiated away from the absorber, are in the range of from 25 to 40% (or more) of the total solar energy delivered to the absorber. These losses are proportional to the size of the absorber and to the fourth power of the absolute temperature (temperature above absolute zero) at the absorber. The ratio of thermal energy transferred by the absorber to the power system, to the total solar energy delivered to the absorber by the solar collector apparatus is referred to as the thermal efficiency of the absorber.

It is thus desirable to reduce these losses, thus improving the thermal efficiency of the absorber. This can be done by reducing the mean temperature of heat acquisition of the system; since the temperature at the absorber is largely the same as the mean temperature of heat acquisition by the power system, reducing this mean temperature results in a reduction of the temperature at the absorber and a substantial proportional reduction in radiated losses from the absorber.

At the same time, it is desirable to improve the thermal efficiency of the power system, allowing it to produce more power from a given amount of heat processed. Note that though an increase in

the mean temperature of heat acquisition is one way to increase the thermal efficiency of a power system, such an increase in the mean temperature of heat acquisition will at the same time decrease the thermal efficiency of the absorber.

Thus, an accurate measure of a solar-thermal power system's total efficiency is not only the thermal efficiency of the power system but rather the product of the thermal efficiency of the power system and the thermal efficiency of the absorber. This is termed the "conversion efficiency" and describes what proportion of the solar power collected is actually converted to output from a solar-thermal power system.

There are, therefore, two ways in which Kalex LLC's technology is able to improve the overall efficiency of a solar-thermal installation. The first is to directly improve the thermal efficiency of the power system. The second is to improve the thermal efficiency of the absorber by reducing the mean temperature of heat acquisition.

Kalex LLC has developed two Kalina cycle power systems for solar-thermal applications. These Kalex systems have increased efficiency as compared to Rankine cycle systems. Moreover, Kalex Kalina cycle power systems attain their high efficiency using heat at lower mean temperatures, thus substantially reducing the losses to radiation at the absorber and in turn providing very substantial cost savings in the solar-collector apparatus.

The high efficiency of Kalex systems is based on the advantages of using a variable composition, multi-component working fluid. By separating and mixing the multi-component working fluid throughout the system, the characteristics of the working fluid at each point in the system can be controlled, greatly increasing the system's thermal efficiency without requiring exotic components, extreme temperature or extreme pressure. The use of a water-ammonia working fluid allows Kalex systems to operate with conventional components, steam turbines and heat exchanger apparatus. In fact, Kalex systems use only proven "off-the-shelf" components that are widely available in the power industry. By avoiding the use of experimental or specialized high cost components, Kalex systems can attain their high efficiency at low cost, while maximizing reliability and minimizing technological risk. The result is a series of power systems that have better efficiency and lower cost than conventional solar-thermal power systems.

In current solar-thermal applications, the power system used for all three of the more common collector systems is a Rankine cycle power system.

To maximize efficiency in these Rankine cycle systems, recent solar-thermal applications have used Rankine cycles with five stages of feed water heating and two steam reheats. Such a system requires three turbines for high pressure, intermediate pressure and low pressure operations. In particular, the low pressure turbine is required to operate with a substantial vacuum at the outlet, making this turbine particularly expensive. Though this approach does improve the overall efficiency of a Rankine power system, it greatly increases the capital costs of such a system.

Currently, the most efficient (subcritical) Rankine cycle systems, (such as those used for base-load coal fired power plants,) have an efficiency of approximately 37% at ISO conditions (i.e., with a cooling water temperature of 11 deg.C.) For solar-thermal applications, where cooling is

usually performed by air with a temperature of at least 30 deg.C., the best efficiency of such a system is the range of 36%.

Proposals exist for improving the efficiency of Rankine cycle power systems for solar-thermal applications to as high as 40% or more. However these proposals involve either the use of extremely high temperatures and thus require ultra-efficient exotic materials for turbines, or else involve using supercritical Rankine cycles, (operating at pressures as high as 4,750psi,) which are extremely expensive and potentially unreliable. Moreover, the increase in the thermal efficiency of a supercritical Rankine power system comes from an increase in the mean temperature of absorbed heat, which in turns results in a decrease in the thermal efficiency of the absorber and thus would tend to offer only modest improvements in the overall conversion efficiency of a solar-thermal installation.

Kalex LLC offers systems designated CSQ-21h and CS-34s for solar thermal applications.

- A flow diagram of system CSQ-21h is presented in **figure 1**.
- A flow diagram of system CS-34s is presented in **figure 2**.
- A detailed description of systems CSQ-21h **appendix A**.
- A detailed description of system CS-34s is given in **appendix B**.

System CSQ-21h provides thermal efficiencies that are an average of 20% (or more) higher than the thermal efficiencies of Rankine cycle systems. At the same time, CSQ-21h offers substantial reduction in the mean temperature of heat acquisition, thereby reducing radiated losses at the absorber and increasing the thermal efficiency of the absorber.

System CS-34s (which is a simpler and lower cost system) provides only moderate improvements in thermal efficiency (as compared to a Rankine cycle,) but has drastically lower mean temperature of heat acquisition, leading to greatly reduced radiated losses at the absorber and thus greatly increasing the thermal efficiency of the absorber.

--Operational parameters of systems CSQ-21h and CS-34s as compared to a Rankine cycle system are given in **table 1** and the associated graphs; **graph 1, graph 2 & graph 3**.

--A comparison of conversion efficiency and of the reduction in required solar collector area for CSQ-21h, CS-34s and a Rankine cycle system, assuming 20% radiated losses, is given in **table 2** and the associated graph; **graph 4**.

--A comparison of conversion efficiency and of the reduction in required solar collector area for CSQ-21h, CS-34s and a Rankine cycle system, assuming 40% radiated losses, is given in **table 3** and the associated graph; **graph 5**.

(Radiated losses of 20% are an optimistic estimate, sometimes attained by heliostat central tower units. Radiated losses of 40% are more accurate for most solar-trough and Fresnel-reflector solar-thermal units.)

In most existing solar-thermal power systems, heat generated by the solar collectors is transferred to a heat transfer fluid, or HTF, (such as molten salt, *Therminol*, etc.) Some designs propose to use a direct transfer of heat to the working fluid of the power cycle, avoiding the use of HTF. It is possible to use Kalex's power systems and Central Axis Solar Trough for either approach (with HTF or for direct transfer of heat to the working fluid.)

Solar-thermal power systems using direct heating of the working fluid of the power system do not allow storage of accumulated energy to provide operation when there is no sun. On the other hand, in a system that uses HTF, it is possible to store the hot HTF and then operate the power system at night or at other times when there is no sun.

Kalex LLC has developed a new, low cost Multi-Tank Heat Storage System for use with solar-thermal power systems that work with HTF.

--A conceptual diagram of the Multi-tank Heat Storage System is presented in **figure 4**.

--A detailed description of the Multi-tank Heat Storage System is presented in **appendix D**.

In general, energy storage systems of this sort consist of two large tanks, one of which is thermally insulated and designed to store hot HTF and the other of which is designed to store cooled HTF. This design means that at the end of the operation cycle of this storage system one tank is full of HTF and the other is empty. This means that the total volume of tanks required is equal to almost two times the total volume of all the HTF in the system. (Note that since cooled HTF is denser than hot HTF, the tank for cooled HTF requires slightly less volume than the tank for hot HTF.)

In Kalex's multi-tank storage system, the HTF is stored in multiple insulated tanks. When one of the tanks in the multi-tank system is emptied of hot HTF, it can be immediately filled with cooled HTF. As a result, the total volume of tanks required can be as low as 10% to 20% (depending on the total number of tanks used) greater than the total volume of all HTF in the system. This is a major reduction in the total volume of tanks and represents a substantial cost savings.

Because these systems use a water-ammonia working fluid at high temperatures, a process for the prevention of nitridation is part of the Kalex technology for solar-thermal power systems.

--A description of Kalex's method for prevention of nitridation is given in **appendix E**.

In summary, Kalex's solar-thermal technology allows for higher power outputs from a given solar collector field or for a much lower cost per installed kilowatt of solar-thermal power from the ground up. Kalex technology also allows for savings in overall operation and maintenance costs, since a Kalex solar-thermal system will tend to operate at lower working temperatures and pressures.

Appendix A: System CSQ-21h

The system CSQ-21h is designed for solar-thermal applications.

The system operates as follows:

Fully condensed rich working solution (having a high concentration of low-boiling component, usually ammonia in the case of a water-ammonia working fluid) with parameters as at point 1, corresponding to a state of saturated or slightly subcooled liquid, is pumped by a feed pump, P1, to an intermediate pressure, and obtains parameters as at point 2, corresponding to a state of a subcooled liquid.

Stream 2 is now sent into a preheater, HE2, where it is heated in counterflow by a stream of returning condensing rich working solution, 26-27, (see below,) obtaining parameters as at point 3, corresponding to a state of saturated or slightly subcooled liquid.

Stream 3 is sent into a recuperative boiler-condenser, HE3, where it is further heated, and substantially vaporized, in counterflow with a returning stream of condensing intermediate solution, 11-14 (see below,) and obtains parameters as at point 8, corresponding to a state of wet vapor, (i.e., a biphasic mixture of vapor and liquid.)

Stream 8 now enters into a recuperative heat exchanger, HE5, where it is heated in counterflow by a stream of vapor of mixed working solution, 18-19, (see below,) and obtains parameters as at point 4, corresponding to a state of superheated vapor.

Meanwhile, a stream of lean working solution, with parameters as at point 24 is pumped by a pump, P2, to a pressure exceeding, but close to, the pressure at point 4, (see above,) and obtains parameters as at point 9, corresponding to a state of subcooled liquid.

Thereafter, stream 9 enters into a recuperative preheater, HE4, where it is heated in counterflow with a returning stream of condensing intermediate solution, 12-13, (see below,) and obtains parameters as at point 10, corresponding to a state of subcooled liquid.

Stream 10 is now mixed with a circulating stream of the same lean working solution, having parameters as at point 32, corresponding to a state of a biphasic liquid-vapor mixture, forming a stream of lean working solution with parameters as at point 33, corresponding to a state of slightly subcooled liquid.

Stream 33 is now pumped by a booster pump, P5, to a desired high pressure, and obtains parameters as at point 30, corresponding to a state of subcooled liquid.

Thereafter, stream 30 enters into a recuperative heat exchanger, HE6, where it is heated in counterflow by a recirculating stream of lean working solution, 31-32 (see below,) and obtains parameters as at point 5, corresponding to a state of subcooled liquid.

Stream 5 now enters into a heat exchanger, HE7, where it is further heated and fully vaporized in counterflow with a stream of heat transfer fluid, HTF, 506-507 (see below,) obtaining parameters as at point 41, corresponding to a state of saturated vapor.

Stream 41 now enters into a heat exchanger, HE8, where it is again heated in counterflow with a stream of HTF, 501-503 (see below,) and obtains parameters as at point 36, corresponding to a state of superheated vapor.

Stream 36 now passes through an admission valve, TV, where its pressure is reduced, obtaining parameters as at point 37, corresponding to a state of superheated vapor.

Stream 37 now enters into a high pressure turbine, T1, where it is expanded, producing work, and obtains parameters as at point 38.

At point 38, the stream of lean working solution is usually in a state of superheated vapor. In the case that the maximum temperature of the heat source (point 501,) is low, then the stream at point 38 could be in a state of saturated or even wet vapor.

Stream 38 is now divided into two substreams, having parameters as at points 31 and 39.

Stream 31 is now sent into HE6, where it is de-superheated (in the case where stream 38, and 31, corresponded to a state of superheated vapor,) and is partially condensed, obtaining parameters as at point 32 and providing heat for process 30-5, (see above.)

Stream 32 is now combine with stream 10 (see above,) forming stream 33. Stream 10 is in a state of a subcooled liquid and fully absorbs stream 32 upon being mixed, producing stream 33, which corresponds to a state of slightly subcooled liquid (see above.)

Meanwhile, stream 39 is combined with stream 4, (see above,) forming a stream of mixed working solution with parameters as at point 34, corresponding to a state of superheated vapor.

Stream 34 now enters into a heat exchanger, HE9, where it is heated in counterflow by a stream of HTF, 502-504, and obtains parameters as at point 17, corresponding to a state of superheated vapor.

Stream 17 is now sent into a low pressure turbine, T2, where it is expanded, producing work, and obtains parameters as at point 18, corresponding to a state of superheated vapor.

Stream 18 is now sent into HE5, where it is de-superheated, providing heat for process 8-4 (see above,) and obtains parameters as at point 19, corresponding to a state of slightly superheated vapor.

Stream 19 is now mixed with a stream of lean liquid, having parameters as at point 29 (see below,) forming a stream of intermediate solution with parameters as at point 20, corresponding to a state saturated vapor.

Stream 20 is now divided into two substreams with parameters as at points 11 and 12.

Stream 11 is now sent into HE3 and stream 12 is sent into HE4. Both streams are partially condensed and obtain parameters as at points 14 and 13 respectively.

Thereafter streams 13 and 14 are combined, forming a stream of intermediate solution with parameters as at point 21, corresponding to state of a liquid-vapor (biphase) mixture.

Stream 21 is then sent into a gravity separator, S1, where it is separated into a stream of saturated vapor with parameters as at point 22 and a stream of saturated liquid with parameters as at point 23. The composition of stream 22 must be the same, or slightly richer, than the composition of rich working solution (see above.)

Stream 23 is now divided into three substreams, having parameters as at points 28, 24 and 25.

Stream 28 is now pumped by a recirculating pump, P3, to a pressure equal to the pressure at point 19, and obtains parameters as at point 29. Stream 29 is then combined with stream 19, forming stream 20 (see above.)

Stream 24, meanwhile, is sent to P2, where it is pumped to an elevated pressure, obtaining parameters as at point 9 (see above.)

Meanwhile, stream 25 is combined with stream 22, forming a stream of rich working solution with parameters as at point 26, corresponding to a state of vapor-liquid (biphase) mixture.

Stream 26 is now sent into HE2, where it is condensed, providing heat for process 3-2 (see above,) and obtains parameters as at point 27, corresponding to a state of vapor-liquid (biphase) mixture.

Stream 27 now enters into the final condenser, HE1, where it is cooled and fully condensed in counterflow by a stream of coolant, 51-52, (water or air) and obtains parameters as at point 1.

The cycle is closed.

The system described used two loops of HTF.

In the first loop, HTF with parameters as at point 505, enters into a solar collector / absorber, SC2, where it is heated and obtains parameters as at point 500. Stream 500 is then divided into two substreams, having parameters as at points 501 and 502.

Stream 501 passes through HE8, providing heat for process 41-36 (see above,) and obtains parameters as at point 503.

Stream 502 passes through HE9, providing heat for process 34-17 (see above,) and obtains parameters as at point 504.

Streams 504 and 503 are now combined, forming a stream of HTF with parameters as at point 505 (see above.)

In the second loop, HTF with parameters as at point 507 enters into a solar collector / absorber, SC1, where it is heated, obtaining parameters as at point 506.

Stream 506 now passes through HE7, providing heat for process 5-41 (see above,) and parameters as at point 507 (see above.)

The rate of circulation for stream 506-507 is substantially higher than the rate of circulation for stream 505-500. Such an arrangement provides for the minimization of the mean temperature of heat acquisition by the system.

In some cases, where it is not desirable to have two HTF loops, it is possible to have a single HTF loop in the system. In this case, stream 505 is sent into HE7, obtains parameters as at point 507, and stream 507 is then sent into a single combined solar collector / absorber. In such a case, the flow rate of the combined stream of circulating HTF will be higher than the flow rate of stream 505-500, but lower than the flow rate of stream 506-507.

The mean temperature of heat acquisition will be higher with a single HTF loop than with two HTF loops.

The working fluid of the system, upon entry into HE7, HE8 and HE9 (i.e., at points 5, 41 and 34, correspondingly,) is in a state of a single phase fluid. Therefore it is possible to have a variant of the system where the working fluid is sent directly into the solar collector / absorbers and no HTF is used. In such a case, the mean temperature of heat acquisition is the lowest possible.

Appendix B: System CS-34s

Kalex designation CS-34s operates as follows:

Working fluid of a basic rich solution (i.e., a solution with a high concentration of ammonia,) with parameters as at point 1, corresponding to a state of saturated liquid, is pumped by a feed pump, P1, to a required elevated pressure, attaining parameters as at point 2, corresponding to a state of subcooled liquid.

Thereafter stream 2 passes through a pre-heater / heat exchanger, HE2, where it is heated in counterflow by a returning, condensing stream of the same rich solution, 26-27, (see below,) and obtains parameters as at point 3, corresponding to a state of a saturated liquid.

Thereafter, stream 3 passes through a recuperative boiling condenser / heat exchanger, HE3, where it is almost fully vaporized in counterflow by a returning stream, 11-14 (see below,) and obtains parameters as at point 8.

Thereafter stream 8 is divided into two substreams, having parameters as at points 6 and 7.

Stream 6 is then sent through a de-superheater / heat exchanger, HE5, where it is fully vaporized and superheated in counterflow by a returning stream of working solution 18-19, (which has a lower concentration of ammonia than the basic rich solution, see below,) and obtains parameters as at point 4, corresponding to a state of superheated vapor.

The returning stream of working solution with parameters as at point 19 corresponds to a state of slightly superheated vapor. Stream 19 now mixed with a stream of liquid, having parameters as at point 29, corresponding to a state subcooled liquid, forming a stream with parameters as at point 20, corresponding to a state of saturated vapor.

Stream 20 is not divided into two substreams, with parameters as at points 11 and 12. Stream 11 now passes through HE3, where it is partially condensed, provides heat for process 2-8, (see above,) and obtains parameters as at point 14, corresponding to a state of vapor-liquid mixture.

At the same time, stream 12 passes through a heat exchanger, HE4, where it is likewise partially condensed in counterflow with a stream of upcoming lean solution, (having a very low concentration of ammonia,) 9-10, (see below,) and obtains parameters as at point 13, corresponding to a state of vapor-liquid mixture.

Streams 13 and 14 are not combined, forming a stream with parameters as at point 21, which is now sent into a gravity separator, S1.

In S1, stream 21 is divided into a stream of saturated vapor with parameters as at 22, and a stream of saturated liquid with parameters as at point 23.

Stream 23 is now divided into three substreams, having parameters as at point 24, 25, and 28.

Stream 28 now sent to a circulating pump, P3, where its pressure is increased to a pressure equal to the pressure at point 19 (see above,) and obtains parameters as at point 29. Stream 29 is then with stream 19, forming a stream with parameters as at point 20 (see above.)

Meanwhile, stream 24 is pumped by a circulating pump, P2, to a elevated pressure, obtaining parameters as at point 9, corresponding to a state of subcooled liquid.

Thereafter stream 9 passes through HE4, where it is heated in counterflow by stream 12-13 (see above,) and obtains parameters as at point 10, corresponding to a state of subcooled liquid.

When the stream of rich solution with parameters as at point 8 was divided into streams 6 and 7, the flow rate of stream 6 is chosen in such a way that stream 6 can absorb the heat released by the returning stream of working solution, 18-19 in HE5 (see above.)

Excess rich working solution from stream 8 therefore forms stream 7 (see above.)

Stream 7 is then mixed with stream 10 (see above,) is fully absorbed by stream 10, and forms a stream of intermediate solution with parameters as point 15, which is in a state of a subcooled or saturated liquid.

Thereafter, stream 15, passes through an evaporator / heat exchanger, HE6, where it is heated and fully vaporized in counterflow with a stream HTF, 44-45, (see below,) and obtains parameters as at point 5, corresponding to a state of saturated vapor.

Stream 5 and stream 4 are now mixed to form a stream of working solution with parameters as at point 16, corresponding to a state of superheated vapor.

Stream 16 is now sent into a superheater / heat exchanger, HE7, where it is further superheated in counterflow with stream HTF, 40-14 (see below,) and obtains parameters as at point 17, corresponding to a state of superheated vapor.

Stream 17 is now sent into a turbine, T1, where it is expanded producing power, and obtains parameters as at point 18, (see above.)

Meanwhile, the stream of liquid from S1, (after being divided into substreams) with parameters as at point 25, is combined a stream of saturated vapor from S1, having parameters as at point 22, and forms a stream of basic rich solution with parameters as at point 26.

Stream 26 now passes through HE2, where it is further condensed providing heat for process 2-3 (see above,) and obtains parameters as at point 27, corresponding to a state of a vapor-liquid mixture.

Thereafter stream 27 is sent through a final condenser/ heat exchanger, HE1, where it is further cooled by a stream of coolant, 50-51, fully condenses, and obtains parameters as at point 1.

The cycle is closed.

Meanwhile, coolant with initial parameters as at point 50 passes through HE1, where it is heated by stream 27-1 (see above,) obtaining parameters as at point 51. The pressure at point 50 is usually equal to atmospheric pressure; pressure at point 51 is slightly below atmospheric pressure. Stream 51 is then sent through a fan, F, where its pressure is increased to atmospheric pressure, obtaining parameters as at point 52. Stream 52 is discharged into the atmosphere. (This assumes the use of air as a coolant. If water is used as coolant, fan F is replaced by a pump.)

HTF, which provides heat to the working fluid in HE6 and HE7, is heated in solar collectors SC1, SC2 and SC3.

In the preferred embodiment of Kalex's system CS-34s, the flow rate of HTF through HE6 (evaporator) is substantially higher than the flow rate of HTF through HE7 (superheater.) As a result, the mean temperature of the heat acquisition in the absorbers in the solar collectors is substantially reduced, which in turn increases the efficiency of the solar collectors, reducing the required surface area of the collectors.

As a result, the heating of the HTF is performed in series.

Stream 45 (a stream of HTF exiting HE6) is pumped by a circulating pump, CP-1, where its pressure is increased, and obtains parameters as at point 46. Stream 46 then passes through solar collector SC-1, where it is heated, and obtains parameters as at point 44. Thereafter stream 44 passes through HE6, providing heat for process 15-5 (see above,) and again obtains parameters as at point 45.

Stream 41 (a stream of HTF exiting HE7) is pumped by a circulating pump, CP2, where its pressure is increased, and obtains parameters as at point 42. Stream 42 then passes through solar collector SC-2, where it is heated, obtaining parameters as at point 43. Stream 43 then passes through solar collector SC-3, where it is further heated, obtaining parameters as at point 40. Stream 40 is not sent into HE7, where it provides heat for process 16-17 (see above,) and obtains parameters as at point 41.

Solar collector SC-1, which provides heat for HE6 (the evaporator,) can be replaced by two or more consecutive solar collectors.

SC-2 and SC-3 can be replaced by a larger quantity of consecutive solar collectors, or by a larger single solar collector.

It is also possible to provide heat to HE6 and HE7 from a single stream of HTF passing through HE7 and then through HE6. This arrangement simplifies the heat collection apparatus but increases the mean temperature of heat acquisition in the solar collectors, thereby reducing the efficiency of the solar collectors and requiring more solar collector surface area in total.

This arrangement is, however, more convenient if HTF is sent from an energy storage system for the operation of the power system at night.

A further possible option is to use different kinds of collectors in a single system. By using Fresnel collectors to obtain the low temperature heat and solar troughs to obtain high temperature heat, the overall cost of the solar collector apparatus can be reduced with no reduction in the system's efficiency.

For any given project it is possible to choose an optimal combination of solar collectors and a rate of circulation of HTF and working fluid based on economic and operational considerations.

The system described above uses HTF to transfer heat from the solar collectors to the working fluid. In the system described above, two loops of HTF are used to transfer heat from the solar collectors to the working fluid. It is possible to design the system so that there is only a single loop of HTF, though this would increase the mean temperature of heat acquisition and thus reduce the efficiency of the absorber.

It is also possible to design the system to forgo the use of HTF and instead to use the working fluid directly in the solar collectors.

In such a case, the working fluid at points 15 and 16, i.e., at the points of entrance into the corresponding heating apparatuses (HE6 and HE7,) is at each point in a single state; fully liquid at point 15 and fully vapor at point 16. This allows streams 15 and 16 to be sent directly into the absorbers of solar collectors. In such a case, HE6 and HE7 do not exist and the working fluid goes directly from the solar collectors to the turbine.

A substantial cost advantage of the system described is that only a single turbine is used. In contrast, a Rankine cycle system, using multiple-state feed-water heating and two re-heats, requires three consecutive turbines; a high pressure, medium pressure and low pressure turbine.

Moreover, the low pressure turbines used in a Rankine cycle system of this sort have an outlet pressure which requires a vacuum, (i.e., is substantially below atmospheric pressure.)

In contrast, system CS-34s uses only a single turbine with a moderate inlet pressure. The water-ammonia working fluid used by the system requires relatively high pressure for condensation, therefore, the single turbine in the Kalex system CS-34s also has a relatively high outlet pressure (over 120 psia.) This allows for the use of a relatively inexpensive turbine with no loss of performance.

Due to the high pressure of condensation, the pressure in S1 (the gravity separator) is likewise high (slightly higher than the pressure of condensation.) To restore the composition of the basic rich solution, the composition of vapor exiting S1 (point 22) must have a concentration of ammonia which is slightly higher than the concentration of ammonia in the basic rich solution. As a result, the temperature at which S1 operates cannot exceed the temperature at which such a rich vapor can be generated. At the same time, the temperature at which the basic rich solution begins to boil (point 3) must be lower than the temperature at point 21 (the temperature at which S1 operates.) This limits the pressure at point 3 and thus likewise limits the inlet pressure of the turbine, T1, at point 17.

As a result, the turbine in the system CS-34s has a moderate inlet pressure and a relatively high outlet pressure. Such a turbine has a higher efficiency and a much lower cost than the three staged turbines used in a Rankine cycle system for solar-thermal applications.

In system CS-34s, condensation of the working fluid occurs in HE1 at a substantially elevated pressure, which provides for a higher heat transfer coefficient in HE1 and a lower volume of the condensing stream. As a result, the condenser (HE1) of system CS-34s is much smaller and less costly than in the vacuum condenser needed in the prior art.

System CS-34s has a much higher thermodynamic efficiency (i.e., 2nd Law efficiency) than a comparable Rankine cycle system. Thus, the net thermal efficiency of system CS-34s is higher than the net thermal efficiency of the Rankine cycle system used in the prior art, at any given equal maximum temperature of working fluid.

At the same time, the mean temperature of heat acquisition in the solar collectors' absorbers is substantially lower in system CS-34s than in the prior art Rankine cycle system. As previously noted, this means that the system CS-34s will have a higher efficiency of its solar collectors and thus require a smaller surface area of solar collectors, leading to substantial cost savings.

Appendix C: Solar-Thermal Energy Storage System.

The Multi-tank Heat Storage System is a means of storing thermal energy in the form of hot heat transfer fluids (HTF) such as various sorts of molten salts, *Therminol*, *Dowtherm*, or other heat transfer fluids.

Conventional systems for storing thermal energy in the form of hot HTF usually consist of two or more tanks, but always in groups of two. One tank (or one set of tanks) is devoted to the storage of hot liquid, which has been heated in a solar collector (or by other means) and the other for the storage of cooled HTF after it has transferred its heat to the working fluid of a power system.

Because cooled HTF has a slightly lower volume than hot HTF, the tank(s) for the storage of cooled HTF is usually slightly smaller than the tank(s) for storing hot HTF. Nevertheless, the overall volume of the storage tanks is almost twice as large as the maximum volume of HTF that is stored in the system.

The aim of the proposed invention is to substantially reduce the total volume of storage tanks.

The multi-tank heat storage system is shown in figure 5. It is comprised of multiple storage tanks. (6 in the figure, but in actual operation, any number of tanks, three or more, can be used.) The system includes a solar collector (or other heat source) where the HTF is heated and a power system where the HTF is cooled, transferring its heat to the working fluid of the power system in order to produce power.

The power system is connected to the solar collectors by two pipes or pipelines, a hot HFT line and a cooled HTF line. Each of the storage tanks is, in its turn, connected to both of these lines.

The system operates as follows: (In this example, it assumed that six tanks are used.) When the system is fully discharge, i.e., all HTF is cooled, tanks T1, T2, T3, T4, and T5 are filled with cooled HTF and tank T6 is empty.

At the beginning of the system's operation, cooled HTF from tank T5 is sent through the cooled HTF line into the solar collector, where it is heated, becoming hot HTF. This hot HTF is then sent through the hot HFT line to tank T6 (which was empty.) This process continues till tank T5 is emptied of cooled HTF and tank T6 is full of hot HTF.

Thereafter, cold HTF from tank T4 is sent via the cooled HTF line to the solar collector and is converted to hot HTF. This hot HTF (from tank T4) is now sent via the hot HTF line to tank T5 (which was previously emptied.) Again the process continues till tank T5 is full and tank T4 is empty.

This process is then repeated, with cooled HTF from tank T3 heated and stored in the now empty tank T4, and so on, till the cooled HTF in tank T1 is heated and stored in tank T2.

As a result, all HTF in the system is now hot HTF and is stored in tanks T2, T3, T4, T5 and T6, and tank T1 is empty.

At the point when the heat of the stored hot HTF is required by the power system, the process described above is reversed. I.E., hot HTF from tank T2 is sent via the hot HTF line into the power system, where it transfers its heat to the working fluid of the power system, enabling the power system to produce power, and is cooled, leaving the power system via the cooled HTF as cooled HTF. This cooled HTF is now stored in tank T1, which was empty at the start of this process.

After tank T2 is emptied in this manner, hot HFT is sent to the power system from tank T3 and after being converted to cooled HFT is then stored in tank T2, which has been emptied of hot HTF previously.

This process is now repeated with the other tanks until the hot HTF in tank T6 is converted to cooled HTF and stored in tank T5. At the end of this process, all the HTF in the system is cooled HTF, stored in tanks T1, T2, T3, T4 and T5, and tank T6 is empty.

The entire process is now complete and the system is returned to its initial, fully discharge state, ready to send HTF to the solar collector (or other heat source) to repeat the entire process.

The main advantage of the proposed system is the drastic reduction in the total volume of storage tanks. For instance, is the system shown in figure 1 and described above, the total volume of storage tanks is 120% of the total maximum volume of the HFT in the system. In the prior art, the total volume of the tanks would be almost 200% of the total maximum volume of the HTF in the system.

If more tanks are used then the total volume of storage tanks will be further reduced. (For example, seven tanks would have only 116.67% of the total volume of the system's HTF. Eight tanks would have 114.29% of the total volume of the system's HTF, and so on.)

In all cases, the total number of tanks will be equal to $N+1$, where N is the number of tanks filled with the system's HFT.

Such a reduction in the volume of storage tanks allows for a corresponding reduction in the total costs of the entire storage system.

The actual number of tanks used for any given application can be chosen to attain a best case from the point of view of economic and operational considerations for each given project.

Appendix D: Prevention of Nitridation

Nitridation occurs as a catalytic heterogenic process in which metal serves as the catalyst.

In Kalina Cycle applications that use a water-ammonia working fluid, nitridation is thought to cause degradation of metal parts, in particular turbine blades, making metal brittle and prone to breakage.

In low temperature applications, water acts as catalytic poison, preventing nitridation at temperatures up to 750 °F (400 °C.) However, once operating temperatures climb above this temperature, water no longer prevents nitridation. Up till now, this has precluded the use of water-ammonia power cycles with operational temperatures higher than 750 °F (400 °C.)

In Kalex high temperature systems, a patented process is used to prevent nitridation by means of adding a catalytic poison to the working fluid.

After more than 3 years of analysis and experimentation, with more than 1,000 samples of materials and catalytic poisons tested to 1,100 °F (higher than the operating temperature of Kalex high temperature power systems), it was determined that sulfur acts as catalytic poison up to 1100 °F. In theory, sulfur should continue to act as a catalytic poison up to 2800°F (1540 °C).

Sulfuric corrosion is not a risk using this method because the concentration of sulfur used is far below threshold concentration at which sulfuric corrosion can begin to occur. None the less, additional catalytic poisons may be added to secure against any hypothetical possibility of sulfur corrosion

The total operational cost of this method is extremely low, and does not negatively impact the economic viability of Kalex technology.

Kalex has exclusive rights to this patented method (subject to U.S. Patent 6,482,272 B2) and offers this anti-nitridation technology as part of the Kalex technological package.

More information can be had in Kalex's detailed report on prevention of nitridation in Kalina Cycle systems, available upon request.

Table 1: Comparison of Performance of Air Cooled Solar-Thermal Power Systems

Cooling air temperature is set at 30 deg.C (86 deg.F)

Solar Heat Source Maximum Temperature		Kalex System CSQ-21h			Kalex System CS-34s			Rankine Cycle System*	
Centigrade	Fahrenheit	Net Thermal Efficiency	Mean Temperature of Absorbed Heat	Change in Absorber Radiated Losses**	Net Thermal Efficiency	Mean Temperature of Absorbed Heat	Change in Absorber Radiated Losses**	Net Thermal Efficiency	Mean Temperature of Absorbed Heat
580 deg.C	1076 deg.F	42.82%	419.30 deg.C	-33.31%	37.28%	340.03 deg.C	-58.99%	36.02%	493.10 deg.C
560 deg.C	1040 deg.F	42.19%	407.18 deg.C	-33.18%	36.63%	330.50 deg.C	-58.58%	35.29%	479.33 deg.C
540 deg.C	1004 deg.F	41.52%	396.82 deg.C	-32.23%	35.96%	319.23 deg.C	-58.12%	34.54%	465.21 deg.C
520 deg.C	968 deg.F	40.69%	386.58 deg.C	-30.92%	35.27%	309.73 deg.C	-57.65%	33.78%	451.08 deg.C
500 deg.C	932 deg.F	39.97%	377.68 deg.C	-29.56%	34.57%	301.55 deg.C	-57.03%	33.00%	436.68 deg.C
480 deg.C	896 deg.F	39.21%	366.42 deg.C	-28.36%	33.85%	291.59 deg.C	-55.45%	32.20%	422.04 deg.C
460 deg.C	860 deg.F	38.40%	355.41 deg.C	-27.15%	33.11%	280.25 deg.C	-56.12%	31.38%	407.21 deg.C
440 deg.C	824 deg.F	37.54%	344.78 deg.C	-25.60%	32.34%	270.82 deg.C	-55.32%	30.53%	392.19 deg.C
420 deg.C	788 deg.F	36.62%	334.62 deg.C	-23.90%	31.56%	261.42 deg.C	-54.45%	29.65%	377.57 deg.C
400 deg.C	752 deg.F	35.63%	323.52 deg.C	-21.79%	30.74%	252.25 deg.C	-52.81%	28.75%	360.77 deg.C
380 deg.C	716 deg.F	34.41%	311.41 deg.C	-20.00%	29.90%	243.21 deg.C	-51.29%	27.82%	344.93 deg.C
360 deg.C	680 deg.F	33.18%	300.03 deg.C	-17.34%	29.03%	235.96 deg.C	-48.79%	26.85%	328.71 deg.C
340 deg.C	644 deg.F	31.94%	290.23 deg.C	-14.31%	28.12%	229.34 deg.C	-45.77%	25.85%	312.40 deg.C

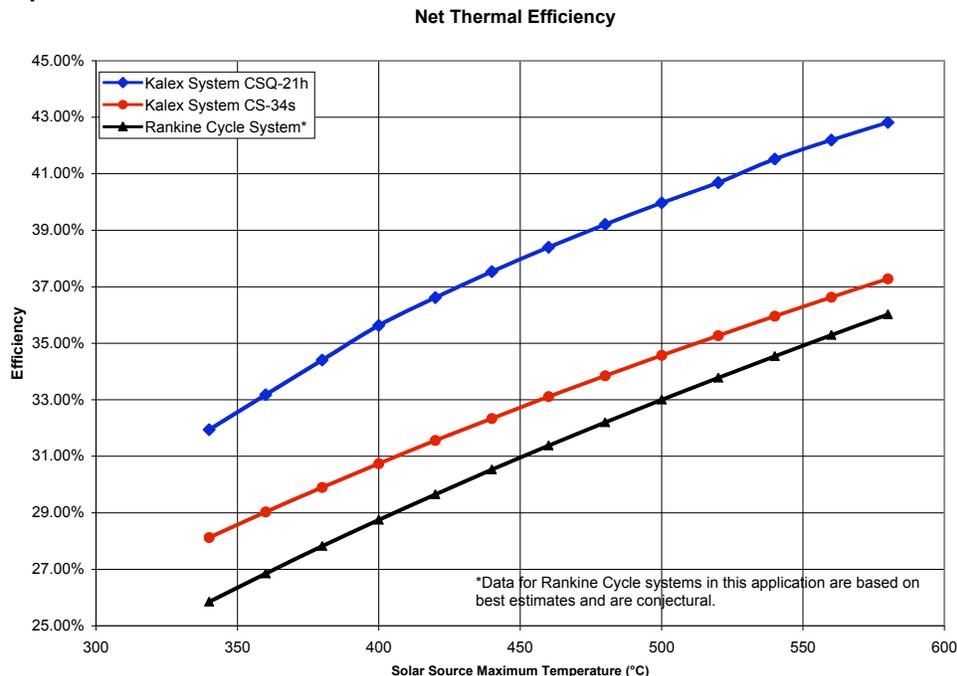
*Data for Rankine Cycle systems in this application are based on best estimates and are conjectural.

**As compared to a base-line based on the use of a Rankine Cycle system.

As a base-line, solar collectors (*depending on type absorber, type of collector and degree of concentration*) lose from 20% to 40% of the concentrated solar energy as radiated heat losses from the absorber. Reducing the mean temperature of absorbed heat at the absorber reduces these losses to a degree given in the table above. (Note that with the use of a Kalex Kalina cycle power system, these reduction in radiated losses from the solar energy absorber come with *increases* in the thermal efficiency of the power system, and thus do not result in any loss of power generation capability; (on the contrary, power generation is increased.)

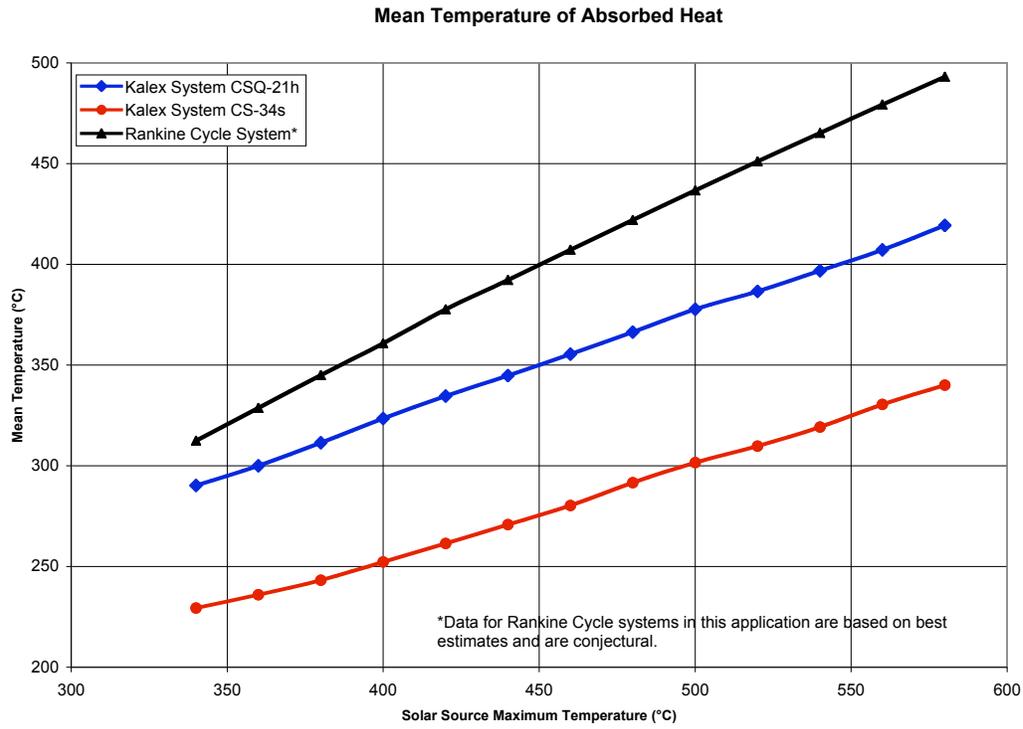
Graphs of Table 1 Data:

Graph 1:



Graphs of Table 1 Data, continued:

Graph 2:



Graph 3:

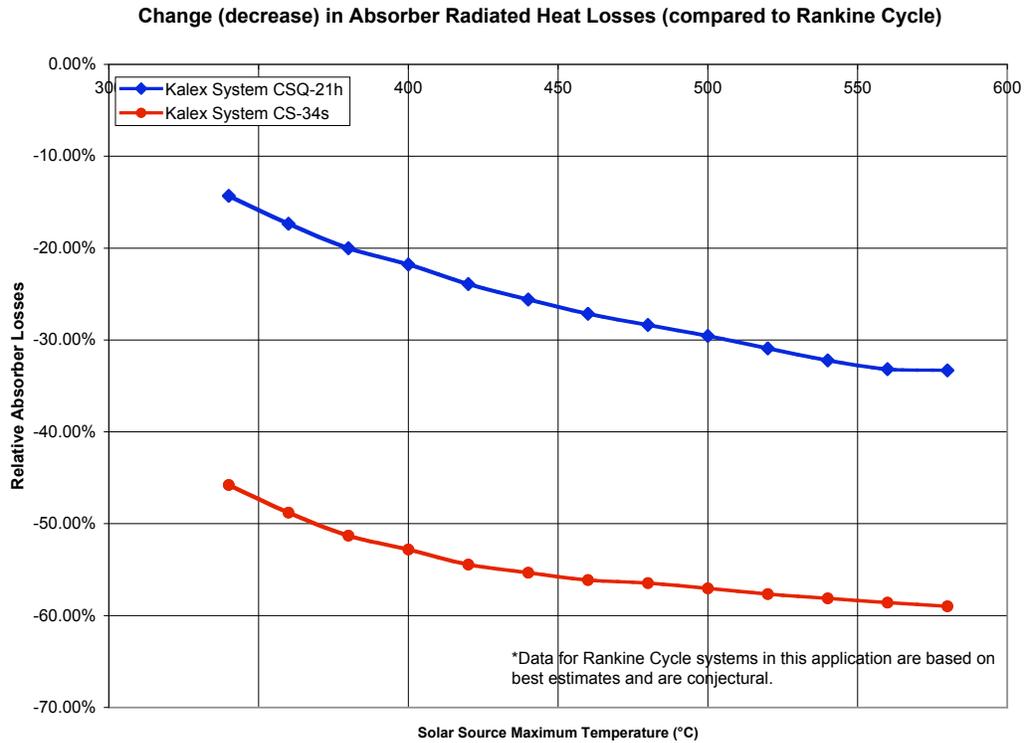


Table 2: Conversion Efficiency and Collector Area Reduction (at 20% radiated losses)

Maximum temperature	Rankine Cycle	CSQ-21h		CS-34s	
	Conversion Efficiency	Conversion Efficiency	Collector Area Reduction	Conversion Efficiency	Collector Area Reduction
580 deg.C	28.82%	37.11%	-22.34%	34.22%	-15.78%
560 deg.C	28.23%	36.55%	-22.76%	33.60%	-15.98%
540 deg.C	27.63%	35.89%	-23.01%	32.95%	-16.14%
520 deg.C	27.02%	35.25%	-23.35%	32.28%	-16.30%
500 deg.C	26.40%	34.34%	-23.12%	31.59%	-16.43%
480 deg.C	25.76%	33.59%	-23.31%	30.90%	-16.63%
460 deg.C	25.10%	32.81%	-23.50%	30.20%	-16.90%
440 deg.C	24.42%	31.95%	-23.57%	29.45%	-17.08%
420 deg.C	23.72%	31.05%	-24.19%	28.68%	-17.92%
400 deg.C	23.00%	30.06%	-23.48%	27.84%	-17.39%
380 deg.C	22.26%	28.90%	-22.98%	26.99%	-17.52%
360 deg.C	21.48%	27.69%	-22.43%	26.06%	-17.57%
340 deg.C	20.68%	26.47%	-21.87%	25.07%	-17.51%

--This table assumes a base-line of 20% radiated losses at the absorber; (*this is an optimistic assessment, as most actual solar-thermal plants have radiated losses of higher than 20%.*)

--Conversion efficiency is the product of absorber efficiency and power system thermal efficiency.

--Collector Area Reduction is as compared to the required collector area for a given power output with a Rankine Cycle power system.

Graph 4; (Graph of Table 2 Data:)

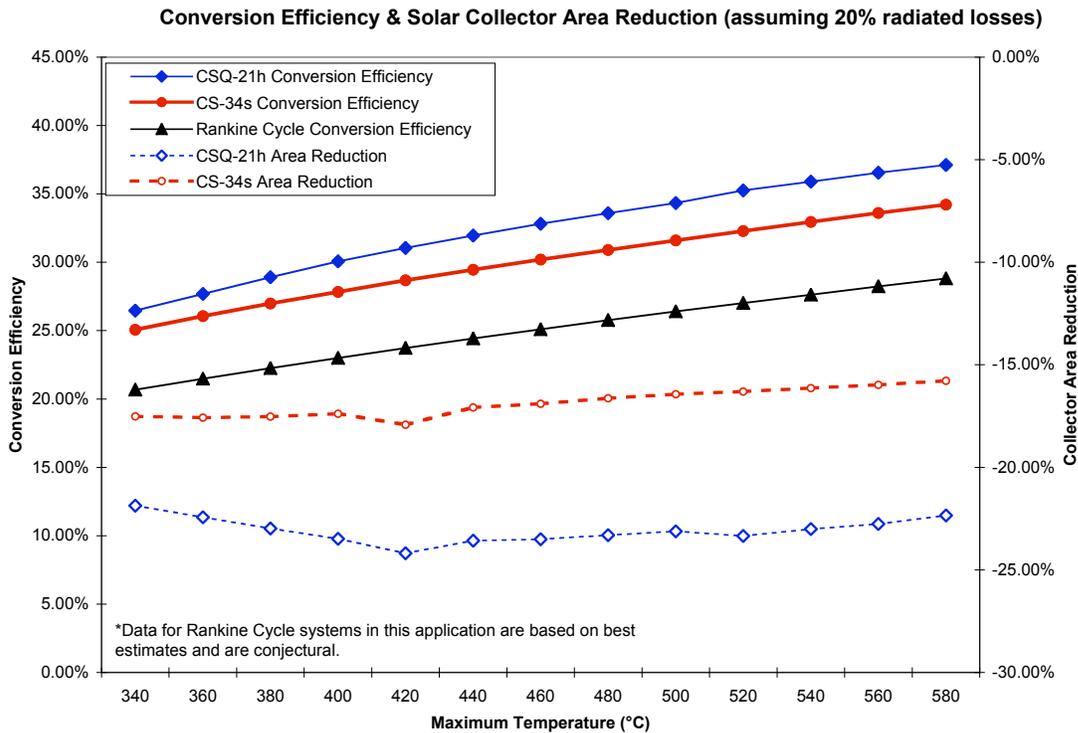


Table 3: Conversion Efficiency and Collector Area Reduction (at 40% radiated losses)

Maximum temperature	Rankine Cycle	CSQ-21h		CS-34s	
	Conversion Efficiency	Conversion Efficiency	Collector Area Reduction	Conversion Efficiency	Collector Area Reduction
580 deg.C	21.82%	31.40%	-30.51%	31.16%	-29.98%
560 deg.C	21.17%	30.91%	-31.52%	30.56%	-30.73%
540 deg.C	20.72%	30.26%	-31.54%	29.94%	-30.79%
520 deg.C	20.27%	29.45%	-31.16%	29.30%	-30.81%
500 deg.C	19.80%	28.71%	-31.03%	28.63%	-30.84%
480 deg.C	19.32%	27.97%	-30.94%	27.82%	-30.54%
460 deg.C	18.83%	27.21%	-30.80%	27.30%	-31.02%
440 deg.C	18.31%	26.37%	-30.56%	26.56%	-31.06%
420 deg.C	17.79%	25.47%	-30.15%	25.81%	-31.07%
400 deg.C	17.25%	24.48%	-29.54%	24.94%	-30.83%
380 deg.C	16.69%	23.40%	-28.67%	24.07%	-30.67%
360 deg.C	16.11%	22.21%	-27.46%	23.08%	-30.20%
340 deg.C	15.51%	20.99%	-26.12%	22.02%	-29.56%

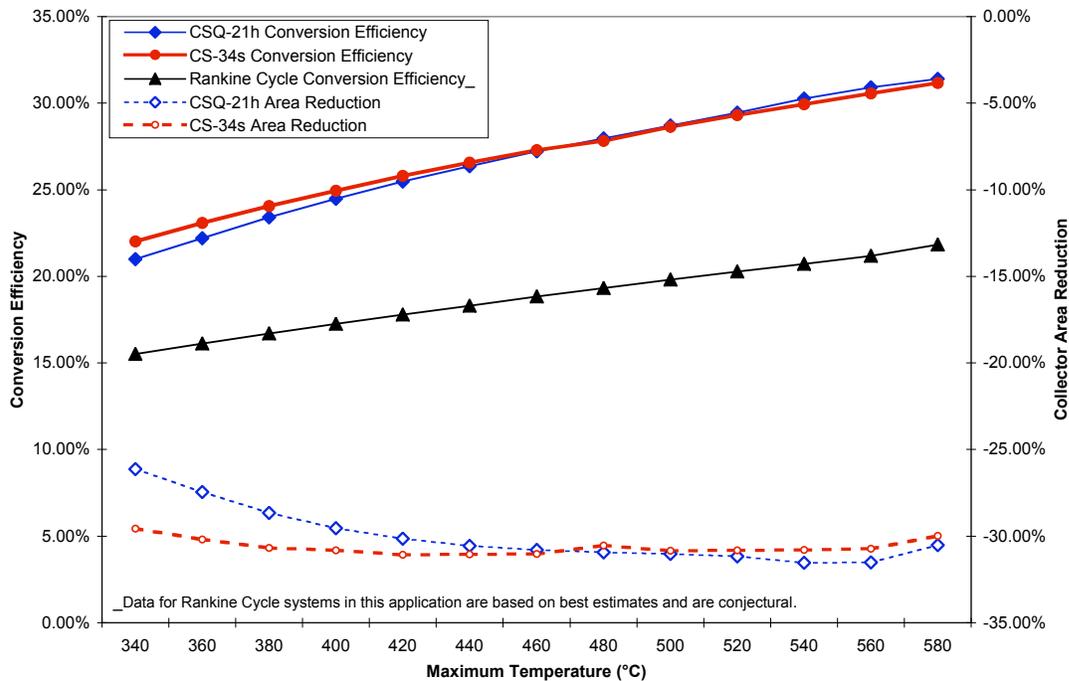
--This table assumes a base-line of 40% radiated losses at the absorber; (*radiated losses on order of 40% can be expected in many solar-trough or Fresnel-reflector solar-thermal plants.*)

--Conversion efficiency is the product of absorber efficiency and power system thermal efficiency.

--Collector Area Reduction is as compared to the required collector area for a given power output with a Rankine Cycle power system.

Graph 5; (Graph of Table 3 Data:)

Conversion Efficiency & Solar Collector Area Reduction (assuming 40% radiated losses)



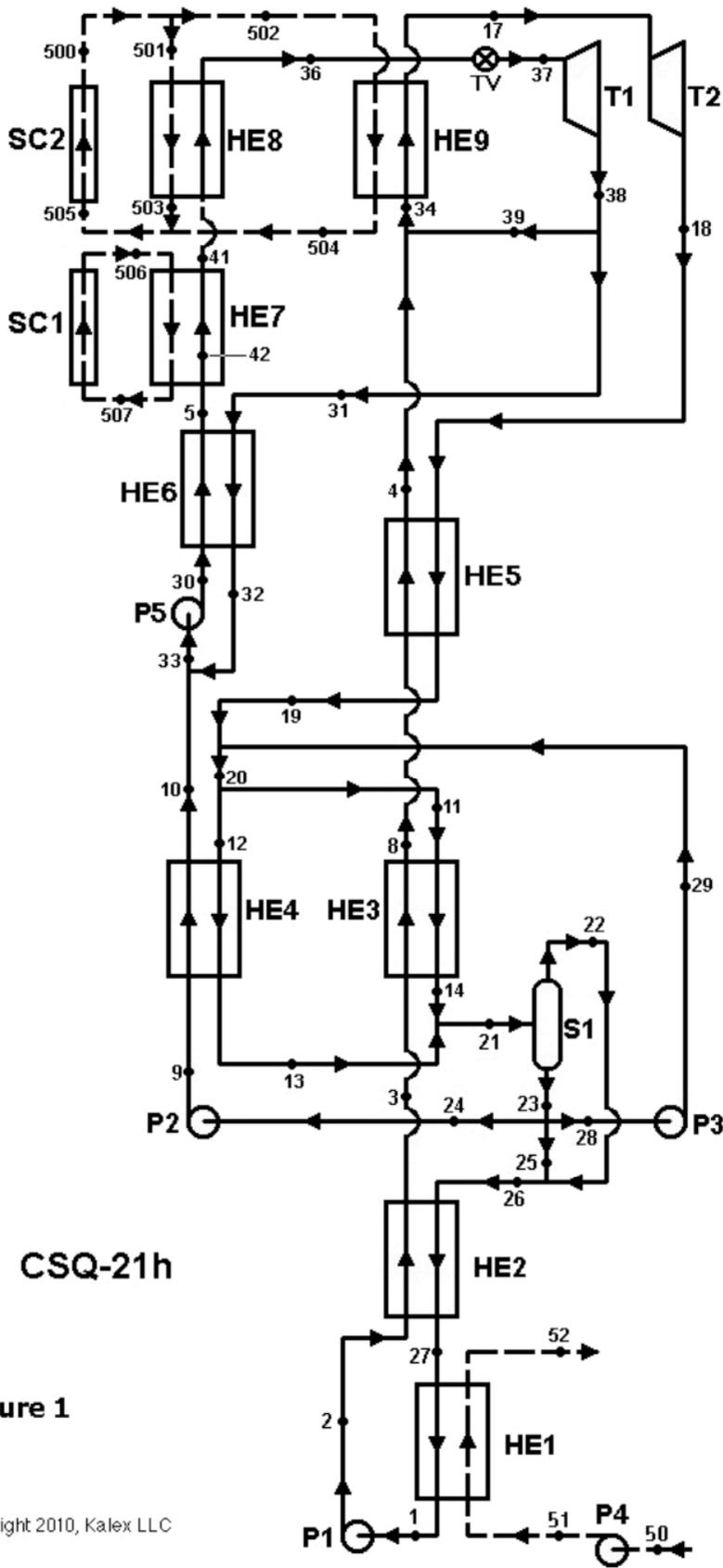
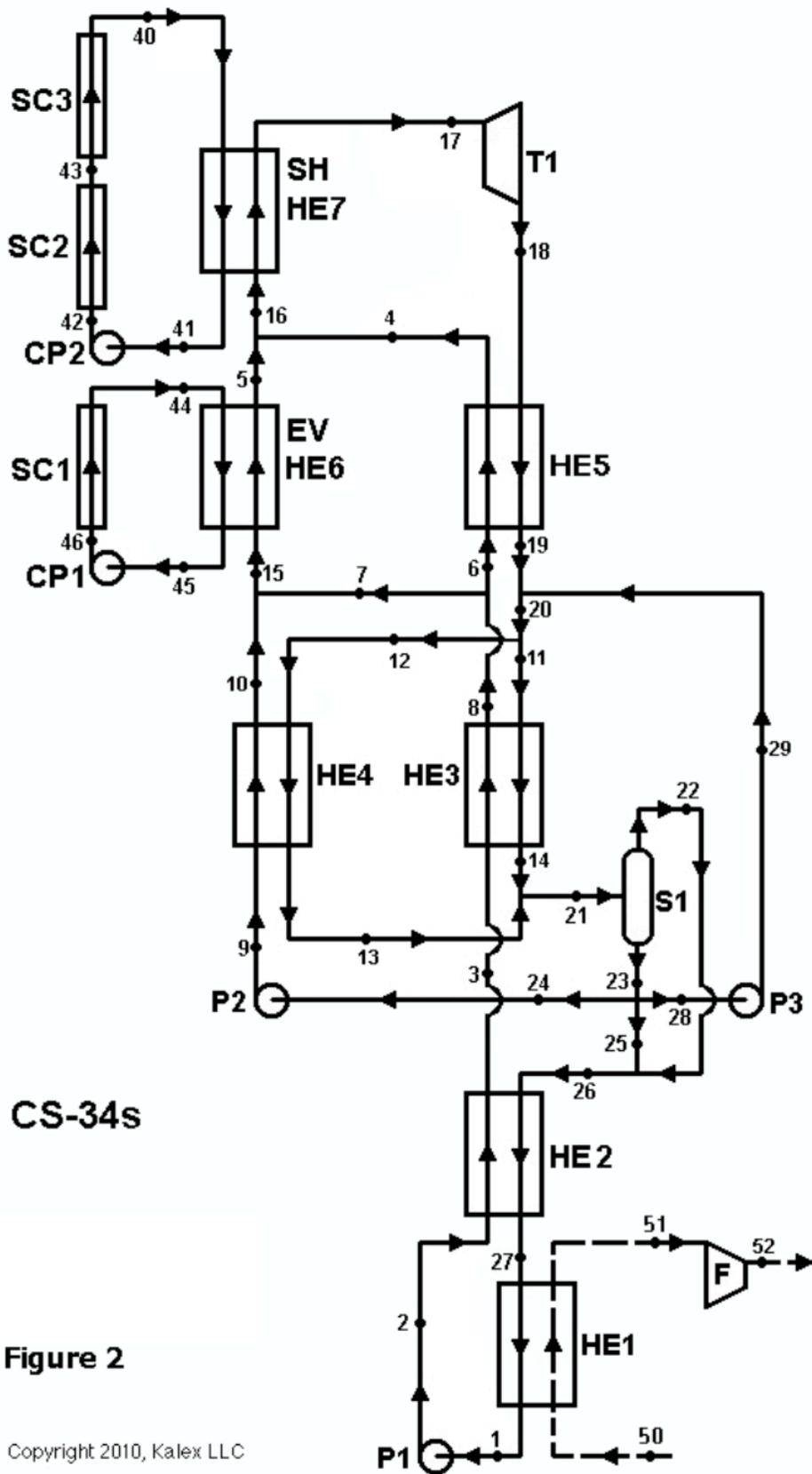


Figure 1

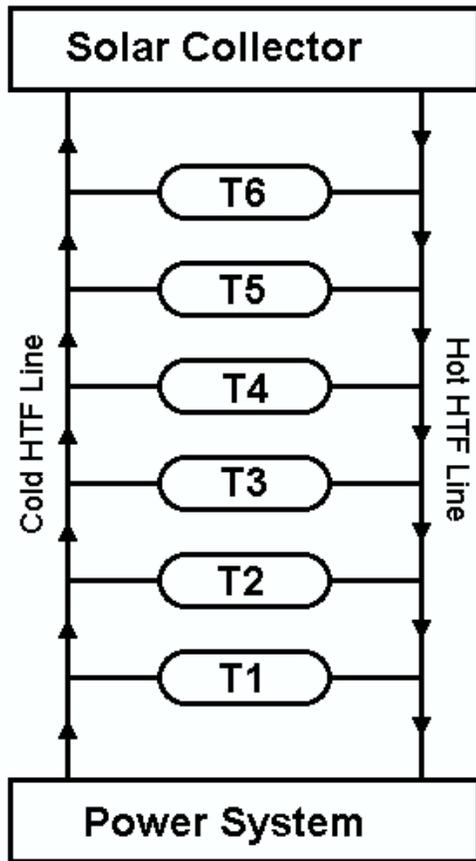
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CS-34s

Figure 2

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Solar Thermal Storage System

Figure 3

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