KALEX LLC.

Superior Efficiency
Reduced Costs
Viable Alternative Energy

Kalex
Kalina Cycle Power Systems
For Biomass Applications

Kalex LLC's Kalina Cycle
for Biomass Applications

Contents:

Kalex Direct-Fired Power Systems Utilizing Biomass:

Kalex LLC has developed designs for biomass power systems which operate with an overall thermal efficiency of 37 to 45% and have lower projected costs per installed kilowatt than currently existing biomass power systems.

The term "biomass" can refer to a fairly wide set of fuels which are quite different from each other. What these fuels have in common is high wetness and relatively low LHV. Because biomass is difficult and costly to amass in large quantities in one place, power plants utilizing biomass as fuel tend to be of small or medium size (from 10 to 50mW.) For such relatively small plants, it is uneconomical to use complex, expensive methods to increase efficiency. Methods such as multi-stage water heating, or other expensive methods of increasing efficiency, are not likely to be cost effective. Also, the relatively small size of turbines used by the majority of biomass plants mean that these turbines tend to have moderate efficiency, on the order of 85% to 87.5%.

As a result, conventional biomass plants tend to have a low overall efficiency, in the range of 22% to 27%. Such a low efficiency results in increased costs per installed kW and adversely effects the overall efficacy of using biomass as a fuel.

An alternative method of using biomass for power production is to gasify the biomass and burn the resulting low caloric gaseous fuel in a gas turbine. A gas turbine with a bottoming cycle may also be used. The overall thermal efficiency for this approach is on the order of 32 to 33%, however, gasification of biomass for use in a combined cycle is expensive and relatively complex.

Kalex LLC has developed a series of Kalina cycle systems suitable to utilizing biomass as a fuel. These Kalex systems have drastically increased efficiency as compared to Rankine cycle and combined cycle systems. Kalex offers systems both for the production of electricity only and for cogeneration of electricity and heat.

The high efficiency of Kalex systems is based on the advantages of using a variable composition, multi-component working fluid. 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 biomass power systems.

The advantages of the Kalex direct-fired biomass technology is based on several factors.

In conventional power plants, the bulk of the heat produced by fuel combustion is spent on vaporizing the working fluid. This means that high temperature heat is used for a process that occurs at moderate temperatures. This causes substantial thermodynamic losses. In a Kalina cycle, due to the use of multi-component variable composition working fluid, most vaporization can occur recuperativly, resulting in an efficiency that is much higher than is possible in a conventional Rankine cycle system.

Kalex offers three power systems for biomass applications*. These are designated CS-21, CSQ-21f and CS-27.

A flow diagram of system CS-21 is presented in <u>figure 1 & 1a.</u>
A flow diagram of system CSQ-21f is presented in <u>figure 2.</u>
A flow diagram of system CS-27 is presented in <u>figure 3.</u>
A detailed description of systems CS-21 and CSQ-21f are given in <u>appendix A.</u>
A detailed description of system CS-27 is given in <u>appendix B.</u>

(Note that Kalex's biomass systems are also fully suitable for coal-fired or any other direct fired fuel applications.)

Each of these systems use a Kalex designed combustion subsystem called an RCSS (recirculation combustion subsystem.) The RCSS is a simple low cost subsystem developed by Kalex to allow the effective combustion and heat utilization of any type of direct-fired fuels, including (but not limited to) high wetness fuels such as biomass. Unlike conventional combustion systems, the RCSS is able to very precisely control the temperature of the combustion process. In this way the use of an RCSS allows for the complete prevention of the formation of NOx, eliminating the need for NOx scrubbers.
A flow diagram of the RCSS subsystem (and its variants) is presented in <u>figures 4 & 4a.</u>
A detailed description of the RCSS subsystem (in several variants) is given in <u>appendix C.</u>

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 biomass power systems.
A description of Kalex's method for prevention of nitridation is given in <u>appendix D.</u>

System CS-21 is designed for the production of electrical power with no cogeneration of heat. It uses a single, relatively low pressure turbine (with an inlet pressure of 650-690psi) with a high back pressure (of 110-120psi) This sort of turbine is much less expensive and more reliable than a turbine with high inlet pressure and vacuum stages, such as is used for conventional systems of this type.

Another advantage of the CS-21 system is that condensation occurs at a relatively high pressure. This reduces the volume of the working fluid in the condenser, increase the heat transfer coefficient (improving overall efficiency) and allows for smaller and less expensive condensers.

CS-21 utilizes an HRVG (heat recovery vapor generator) with finned tubing as a boiler. An HRVG is much less expensive than a conventional boiler. The utilization of finned tubes in the HRVG, made possible by the use of the RCSS combustion subsystem mentioned above, allows the HRVG in system CS-21 to perform as well as or better than a conventional boiler. The use of the RCSS allows for the very precise temperature control required to make the use of an HRVG with finned tubing practicable. Temperature differences in between flue gasses and the working fluid in the HRVG are substantially smaller than in a conventional boiler, however this is more than compensated by the use of finned tubing, and the overall result is a drastic reduction in the cost of the boiler with no loss of overall efficiency for the system.

System CS-21 has a net thermal efficiency of approximately 37%; this allows for biomass power systems with efficiencies that are otherwise achievable only in very large base-load coal-fired conventional power plants. This represents an improvement of more than 40% over the efficiency of a conventional biomass plant.

For larger biomass power plants, Kalex offers system CSQ-21f. System CSQ-21f is similar to CS-21, but has an additional high pressure turbine. CSQ-21f has a net efficiency of just over 43%*; an efficiency that is superior to those achieved in a Rankine cycle system in supercritical coal-fired base load power plants. The somewhat higher cost of CSQ-21f compared to CS-21 makes CSQ-21f suitable for larger biomass applications.

When comparing the efficiency of systems CS-21 and CSQ-21f to the efficiency of large base-load conventional Rankine power systems, note that the base-load conventional Rankine power systems' overall thermal efficiencies are computed based on the use of large steam turbines with isentropic efficiencies in the range to 91% to 93%. Kalex CS-21 and CQ-21f systems' overall thermal efficiencies are computed based on the use small relatively inexpensive turbines with isentropic efficiencies of approximately 87%. Thus, it can be seen that Kalex's efficiency advantage can be achieved in spite of the use of smaller, less efficient (and less expensive) turbines. *For applications using larger, more efficient turbines, (with isentropic efficiencies in the range of 93%,) system CSQ-21f can attain a net efficiency somewhat in excess of 45%.

For cogeneration of electrical power and heat Kalex offers system CS-27. The efficiency of cogeneration systems depends on the ratio of power production and heat production; the more heat produced in terms of the total output of the system, the higher the overall thermal efficiency of the system, irrespective of the actual efficiency of such a system if it was set to produce only electrical power. Thus, for example, if a system were to be designed to give 100% of its output as heat, that system would have a thermal efficiency of 100%. However this would not automatically mean that such a system would be efficient in producing power. This fact makes statements about the thermal efficiency of cogeneration systems difficult to make clear. However, comparing system CS-27 with a conventional Rankine system designed to give the same output of power and heat would show that CS-27 requires 20% - 25% less fuel for the given output.

In general, due to their higher efficiency, Kalex systems require the processing of less heat for a given output. As a result, the size of the combustion / boiler subsystem is drastically reduced, leading to substantial cost savings.

Likewise the higher efficiency of these systems reduces the fuel used, leading to large savings in both the cost of fuel and the cost of fuel handling. The reduction in fuel use and costs in on the order of 30% or better. (In the case of CSQ-21f, the reduction in fuel use is about 39%.) This savings also corresponds to a reduction of CO_2 emissions for a given power output.

Overall, Kalex systems for biomass open up the possibility of using biomass fuels at efficiencies that are as high or higher than base-load coal-fired power plants and with per-kilowatt installation costs that are comparable to or lower than the per-kilowatt installation costs of base-load coal-fired power plants.

Appendix A:
CS-21 and CSQ-21f

System CS-21 is designed for small and medium power plants (up to 50mW.) CS-21 is a power system designed for the utilization of heat sources with high or medium initial temperatures (from 1076 to 400 deg. F) in medium and small scale power plants.

A flow diagram of system CS-21 is presented in **figure 1** (attached.)

This system utilizes a multi-component variable composition (water-ammonia mixture) working fluid.

The system operates as follows:

Fully condensed working fluid (basic solution,) with parameters as at point 1, is pumped by a feed pump, P1, to required high pressure, and obtains parameters as at point 2. The stream with parameters as at point 2 then passes through a heat exchanger (preheater), HE2, in counterflow with a returning stream 26-27, (see below) and obtains parameters as at point 3, corresponding to a state of saturated or slightly subcooled liquid.

The stream with parameters as at point 3 then passes through a heat exchanger (recuperative boiler-condenser), HE3, where it is heated and vaporized in counterflow with a returning condensing stream 11-14, (see below,) obtaining parameters as at point 8, which corresponds to a state of wet vapor.

Meanwhile a stream of lean saturated liquid with parameters as at point 24 (see below) enters into a pump, P2, where it is pumped to a required high pressure, and obtains parameters as at point 9. Stream 9 is then sent into a heat exchanger, HE4, where it is heated in counterflow with a condensing stream, 12-13, (see below,) and obtains parameters as at point 10.

The stream with parameters as at point 8 passes through a heat exchanger, HE5, where it is heated in counterflow by a stream of de-superheating vapor, 30-31, and obtains parameters as at point 4.

The stream with parameters as at point 10 is meanwhile sent into a heat exchanger, HE6, where it is heated in counterflow, and partially vaporized by a stream of de-superheating vapor, 32-33, and obtains parameters as at point 5, corresponding to a state of a vapor-liquid mixture.

Thereafter, streams 4 and 5 are combined, forming a stream of working solution with parameters as at point 7, corresponding to a state of a liquid-vapor mixture.

Stream 7 is then sent into a gravity separator, S2, where it is separated into a stream of saturated vapor with parameters as at point 6 and saturated liquid with parameters as at point 35.

Streams 6 and 35 are then sent through a heat recovery vapor generator, HRVG, where they are heated in counterflow by a stream of heat source flow *(flue gas or other heat source,)* 500-502 (see below,) and obtain parameters as at point 16 and 15 respectively, which correspond (for both) to a state of superheated vapor.

Thereafter streams 15 and 16 are combined, forming a stream of superheated vapor with parameters as at point 17.

The stream with parameters as at point 17 then passes through a turbine, T1, where it expands, producing power and obtains parameters as at point 18. The stream at point 18 is in a state of a superheated vapor.

Thereafter the returning stream 18 is divided into two substreams with parameters as at point 30 and 32 respectively. Streams 30 and 32 then pass through heat exchangers HE5 (for stream 30) and HE6 (for stream 32) respectively, where they are both cooled and almost completely de-superheated, providing heat for processes 8-4 and 10-5 respectively (see above,) obtaining parameters as at point 31 (from stream 30) and 33 (from stream 32) respectively. Stream 31 and 33 are then combined forming a stream with parameters as at point 19.

It should be noted that, in an alternate embodiment of the system, streams 10 and 8 can be sent in parallel into a single, dual flow heat exchanger. In such a case stream 18 is not divided, but is instead sent in counterflow to both stream 10-5 and stream 8-4 in the dual flow heat exchanger, exiting with parameters as at point 19.

Stream 19 is in a state of slightly superheated vapor. At this point, stream 19 is mixed with a stream of liquid having parameters as at point 29 (see below,) forming a stream of saturated vapor with parameters as at point 20.

The stream of saturated vapor with parameters as at point 20 is then divided into two substreams, having parameters as at points 11 and 12 respectively. Stream 11 passes through heat exchanger HE3, where it is partially condensed, releasing heat for process 3-8 (see above,) and obtains parameters as at point 14.

Meanwhile stream 12 enters into HE4, where it is where it is partially condensed, releasing heat for process 9-10 (see above,) and obtains parameters as at point 13.

Streams 14 and 13 are then combined forming a stream of partially condensed working fluid with parameters as at point 21 corresponding to a state of vapor-liquid mixture.

Again, it is possible, in an alternate embodiment of the system, to utilize a single, dual flow heat exchanger in place of HE3 and HE4; in such a case, stream 20 is not divided, instead being sent in counterflow to both streams 9-10 and 3-8 in the dual flow heat exchanger, exiting with parameters as at point 21.

*A diagram of such a variant, utilizing two dual flow heat exchangers, is presented in **figure 1a** (attached.)*

Stream 21 is then sent into a flash tank (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 concentration of the low boiling component *(ammonia, in a water-ammonia working fluid)* at point 22 is slightly higher than the concentration of the low boiling component at point 1 (see above.)

Stream 23 is then divided into three substreams, having parameters as at points 24, 25 and 28 respectively. The stream with parameters as at point 28 is then sent into a pump, P3, where it is pumped to an increased pressure, and obtains as at point 29. Stream with 29 is then mixed with stream 19 to form stream 20, (see above.)

Stream 24 is meanwhile sent into a pump, P2, where its pressure is increased to a high pressure, and obtains parameters as at point 9, (see above.)

Meanwhile, stream 25 is mixed with the stream of vapor having parameters as at point 22, and forms a stream of basic solution with parameters as at point 26. The stream with parameters as at point 26 is then sent into HE2 where it is further condensed, providing heat for process 2-3, (see above,) and obtains parameters as at point 27.

Stream 27 is then sent into a heat exchanger (final condenser), HE1, where it is cooled and fully condensed by a stream of cooling medium (air or water) 51-52, (see below,) and obtains parameters as at point 1. The cycle is closed.

The cooling medium used for process 27-1, which has initial parameters as at point 50 is pumped to a higher pressure by a pump, P4, obtaining parameters as at point 51 before it is sent into HE1 as described above.

Note that the flow rate of the working fluid passing through turbine T1 is substantially higher than the flow rate of the working fluid passing through the condenser (HE1.) The flow rate passing through the turbine is equal to the sum of flow rates of the stream of basic solution passing through HE1 and the stream of lean recirculating solution (with parameters as at point 24) coming from the separator, (S1.)

The back pressure of the turbine, as well as the pressure of all of the returning streams of working fluid, are controlled by the required pressure for the condensation of the basic working solution in HE1. This pressure is usually quite high (in the range of 100-120psi.) The pressure at point 21 (entering into S1) is only slightly higher than the pressure in HE1. However, because the vapor produced in S1 (stream 22) must be at least slightly richer (having a higher concentration of the low-boiling component) than the basic working solution at point 1, this places a limit on the highest possible temperature at point 21.

At the same time, the temperature at point 3 (corresponding to the boiling point of the basic solution,) must be colder than the temperature at point 21. This limits the pressure at point 3. As a result, the pressure at point 17 (entering the turbine) is limited to a pressure at least slightly lower than the pressure at point 3. (The pressure at point 17 is usually in the range of 650-700psi.)

As a result, the turbine, T1, has a moderate inlet pressure and a substantial back pressure.

For installations larger than 30mW, a system called CSQ-21f is available.
A flow diagram of system CSQ-21f is presented in **figure 2** (attached.)

The operation of CSQ-21f differs from CS-21 as follows:

In distinction from the operation of CS-21, in CSQ-21f, the stream of lean solution with parameters as at point 10 is mixed with a stream the same composition, having parameters as at point 32 (see below,) forming a stream with parameters as at point 33.

Stream 32 is in a state of a vapor-liquid mixture, whereas stream 10 is in a state of a subcooled liquid. As a result of mixing, stream 32 is fully absorbed by stream 10, and the resultant stream 33 is in a state of a saturated liquid.

Thereafter, stream 33 is pumped by a pump, P5, to a desired high pressure, obtaining parameters as at point 30.

Stream 30 is then sent into a recuperative heat exchanger, HE6, where it is heated in counterflow by a stream of returning working solution vapor, 32-33, (see below,) and obtains parameters as at point 5.

Stream 5 then passes through a heat recovery vapor generator, HRVG, where it is further heated, vaporized and superheated by a heat-source stream 500-502, obtaining parameters as at point 36, corresponding to a state of superheated vapor.

In the process of heating stream 5 in the HRVG, stream 5 is initially heated to its boiling point, (point 41) and is then completely vaporized and superheated.

Stream 36 is then sent through an admission valve, TV, where its pressure is reduced, obtaining parameters as at point 37. Stream 37 is then sent through the high pressure turbine, T1. Here it is expanded, producing power, and exits the turbine with parameters as at point 38, corresponding to a state of superheated vapor.

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

Stream 31 is now sent into HE6, where it is cooled, providing heat for process 30-5 (see above,) and obtains parameters as at point 32. Stream 31 is initially de-superheated in HE6 to is dew point (point 7) and then partially condensed, (exiting HE6 with parameters as at point 32.)

Meanwhile, stream 38 is now mixed with a stream of rich solution having parameters as at point 4, forming a stream with parameters as at point 34.

Stream 34 is now sent back into the HRVG, where it is heated by the heat source, 500-502 (see above,) and obtains parameters as at point 17.

Stream 17 is now sent into a low-pressure turbine, T2, where it is expanded, producing power, and obtains parameters as at point 18, corresponding to a state of superheated vapor. (In distinction from CS-21, all of the stream having parameters as at point 18 passes through HE5.)

From point 19 on, the operation of CSQ-21f is the same as CS-21.

Note that the high pressure of the stream of lean solution after P5 (stream 30, and correspondingly stream 37) can be chosen to be relatively higher or lower. The higher the pressure chosen, the higher the efficiency of the system. In actual operation this pressure will be determined by, (or the chosen pressure will determine,) the characteristics of the turbine, T1.

Appendix B:
CS-27

CS-27 is designed for the simultaneous production of electrical power and heat. The assumption of this description is that heat is produced in the form of hot water, but as will be made clear from the description, the heat produced can be used in other ways as well.

A flow diagram of CS-27 is presented in **figure 3**, (attached.)

The system operates as follows:

Fully condensed working fluid (of a rich solution; i.e., a mixture with a high content of the light-boiling component,) having parameters as at point 1 enters into a feed pump, P1, where it is pumped to a specified high pressure and obtains parameters as at point 2.

Stream 2 is then sent into a preheater / heat exchanger, HE2, where it is heated in counterflow by a returning condensing stream 26-44, (see below) and obtains parameters as at point 3, corresponding to or close to a state of saturated liquid.

Thereafter the stream of rich solution with parameters as at 3 passes through a recuperative boiler condenser / heat exchanger, HE3, where it is heated in counterflow and substantially vaporized by a returning condensing stream, 11-14 (see below) and obtains parameters as at point 8, corresponding to a state of a vapor-liquid mixture.

Stream 8 is then sent into a recuperative heater / heat exchanger, HE5, where it is fully vaporized and superheated in counterflow with a returning de-superheating stream of working solution vapor, 30-31 (see below,) and obtains parameters as at point 4, corresponding to a state of superheated vapor.

Thereafter stream 4 is combined with a stream of recirculating lean solution (a mixture with a high content of the low-boiling component,) having parameters as at point 5, and forms a stream of working solution with parameters as at point 7.

The stream of working solution at point 7 is in a state of a liquid-vapor mixture. Stream 7 then enters into a gravity separator, S2, where it is separated into two streams; a stream of saturated vapor with parameters as at point 6 and a stream of saturated liquid with parameters as at point 35.

This separation allows for the easy distribution of the respective streams into multiple tubes passing through the Heat Recovery Vapor Generator, HRVG.

Inside the HRVG, stream 6 is superheated in counterflow by a stream of hot flue gas, 500-502, obtaining parameters as at point 16. Meanwhile stream 35 is heated, fully vaporized and superheated in counterflow by the flue gas, 500-502, obtaining parameters as at point 15.

Stream 15 and 16 are then combined, forming a stream of working solution with parameters as at point 17, corresponding to a state of high pressure, superheated vapor.

Stream 17 is then sent into a turbine, T1, where it is expanded, producing power, and obtains parameters as at point 18, corresponding to a state of superheated vapor.

Stream 18 is then divided into two substreams, with parameters as at point 30 and 32.

The stream with parameters as at point 30 now passes through HE5, where it is de-superheated, providing heat for process 8-4, (see above,) and obtains parameters as at point 31.

Meanwhile the stream with parameters as at point 32 is sent into a heat exchanger, HE6, where it is de-superheated, providing heat for process 10-5 (see below,) and obtains parameters as at point 33.

The parameters of the streams and points 31 and 33 are practically identical. Thereafter streams 31 and 33 are combined forming a stream of working solution with parameters as at point 19.

The stream of working solution at point 19 is in a state of slightly superheated vapor. Stream 19 is then mixed with a recirculating stream of cold lean solution, having parameters as at point 29, and forms a stream of condensing solution (which is slightly leaner than the working solution) having parameters as at point 20.

Stream 20 is then divided into three substreams, having parameters as at points 11, 12 and 34.

Stream 11 passes through HE3, where it is partially condensed, providing heat for process 3-8 (see above,) and obtaining parameters as at point 14.

Stream 12 passes through a heat exchanger, HE4, where it is partially condensed, providing heat for preheating a stream of lean solution in process 9-10 (see below,) and obtains parameters as at point 13.

Stream 34 is sent into a water heater / heat exchanger, HE11, where it is partially condensed, releasing heat which is used for heating water (process 54-55, see below,) for cogeneration of heat, and obtains parameters as at point 36.

The parameters of the streams at point 13, 14 and 36 are substantially similar. Streams 13, 14 and 36 are then combined, forming a stream of condensing solution with parameters as at point 21.

Stream 21 is then separated into two substreams with parameters as at point 26 and 37.

Stream 26 passes through HE2, where it further condensed and cooled, providing heat for process 2-3, (see above,) and obtaining parameters as at point 44.

Stream 37 is further divided into two more substreams with parameters as at points 38 and 39.

Stream 38 is then sent into a water heater / heat exchanger, HE10, where it is further condensed, providing heat which is used for heating water (process 53-54, see below,) for cogeneration of heat, and obtains parameters as at point 40.

Meanwhile, stream 39 passes through a heat exchanger, HE9, where it is further condensed, providing heat for the reheating of a stream of lean solution (process 43-9, see below,) and obtains parameters as at point 41.

At this point, the streams with parameters as at points 44, 41 and 40 each have different temperatures. Streams 44, 41 and 40 are now combined, forming a stream of condensing solution with parameters as at point 42.

Stream 42 is then sent into a gravity separator, S1, where it is separated into a stream of rich saturated vapor having parameters as at point 22 and a stream of saturated liquid having parameters as at point 23.

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

Stream 25 is then combined with the stream of vapor having parameters as at point 22, forming a stream of rich solution with parameters as at point 27.

Stream 27 is then sent into a final condenser / heat exchanger, HE1, where it is cooled and fully condensed in counterflow with a stream of cooling air or water, 51-52, and obtains parameters as at point 1, corresponding to a state of fully condensed liquid (see above.)

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

At the same time, stream 24 enters into a feed pump, P2, where its pressure is increased to a required high pressure, forming a stream of recirculating lean solution having parameters as at point 43, corresponding a state of subcooled liquid.

Stream 43 then passes through HE9, where it is pre-heated by a condensing stream 39-41, (see above) and obtains parameters as at point 9.

Stream 9 is then sent into HE4, where it further heated by a condensing stream 12-13 (see above,) obtaining parameters as at point 10.

Stream 10 is then sent into HE6, where it is further heated and partially vaporized by a stream of retiring working solution vapor, 32-33 (see above,) obtaining parameters as at point 5.

Thereafter stream 5 (lean recirculating solution) is combined stream 4 (rich solution,) forming a stream of working solution with parameters as at point 7 (see above.)

Heat released in HE11 and HE10 by streams of condensing solution 34-36 and 38-40 respectively, is used to heat a stream of water, 53-54-55, for purposes of heat cogeneration (see above.)

System CS-27 somewhat resembles system CS-21, but differs from it in the following: in system CS-21 the quantity of recirculating lean solution is limited by the quantity of heat used for the vaporization of rich solution in heat exchanger HE3 and for preheating recirculating lean solution in HE4. In CS-27, the quantity of recirculating lean solution can be increased to whatever extent is desired. This is possible because the additional heat released by the returning condensing stream, between points 20 and 21, can be used in HE10 for cogeneration.

Increasing the quantity of the recirculating lean solution will increase the quantity of heat output and reduces the power output as a proportion of the total output of the system.

However, it should be noted that due to the fact that the recirculating lean solution is fully vaporized and passes through the turbine, producing power, before it is condensed, producing heat, increasing the heat output of the system causes only a relatively small reduction of the system's power output.

The ratio between power and heat output in CS-27 is controlled by the ratio of the weight-flow rates of the recirculating lean solution to the weight-flow rate of rich solution.

In system CS-21, the inlet pressure into the turbine is limited by the fact that the returning condensing stream exiting heat exchangers HE3 and HE4 must have a temperature low enough that, being separated into liquid and vapor, it is able to produce vapor which is at least slightly richer than the basic rich solution. This in its turn requires that the temperature at point 3 should be lower than the temperature at point 21. As a result, this limits the pressure at point 3 and correspondingly the inlet pressure into the turbine at point 17. However, in CS-27, there is an additional heat load at temperatures which are lower than the temperature at point 3, i.e., in heat exchangers HE9 and HE10. Separation of the condensing solution (at point 42) is thus performed at a substantially lower temperature than the temperature at point 21. Therefore, the temperature at point 3 in CS-27 can be increased and correspondingly, this will increase the inlet pressure into the turbine at point 17.

However, this increase in the inlet turbine pressure is not unlimited. The initial temperature of condensation of the returning condensing stream must be higher than the temperature at which the upcoming rich solution will start to boil. The more the temperature at point 20 is higher than the temperature at point 3, the more vaporization of the rich solution is performed in a recuperative manner. Therefore, on one hand, increasing the turbine's inlet pressure increases the rate of expansion in the turbine (increasing power output,) but on the other hand increasing the turbine's inlet pressure decreases the rate of recuperative boiling of the rich solution in heat exchanger HE3 (thus reducing the over efficiency of the system.) The actual inlet pressure in the turbine should be chosen as a result of optimization of output based on given boundary conditions.

For example, with a given amount of fuel of a given quality, where CS-21 would produce 10 megawatts of electrical power output, CS-27 (with a recirculation ratio of lean solution to lean solution of 3 to 1,) would produce 7.51 megawatts of power and 15.55 megawatts of heat, with a thermal efficiency of 85.42%. With the same fuel, but with a recirculation ratio of 5 to 1, CS-27 produces 7.125 megawatts of electrical power and 17.23 megawatts of heat, with a thermal efficiency of 90.22%.

It should be noted that these high thermal efficiencies are a consequence of producing heat rather than electricity. If CS-27 were to produce electrical power only (at which point it would be identical to CS-21) its efficiency would be 37.1%.

It should also be noted that a conventional Rankine cycle system cogeneration plant can achieve similar high thermal efficiency, but it would produce a much lower electrical power output than CS-27.

Thus, for the production of a given amount of heat and electrical power, CS-27 would consume 20 to 25% less fuel than a conventional Rankine cycle system.

In general, the advantage of CS-27 over a conventional Rankine cycle system will in proportion to the amount of electrical power output desired from the system.

Appendix C:
RCSS

The Kalex RCSS is a modular combustion subsystem applicable to any fuel type. It allows for all the advantages of fluidized bed combustion without loss of efficiency. Moreover, the RCSS does away with much of the maintenance expense associated with conventional combustion systems.

In the process of the combustion of fuels, the minimum quantity of air that is theoretically necessary for complete combustion is such that all oxygen contained in the air is completely consumed. Such a process is referred as *stoichiometric* process, and corresponds to the highest possible temperature in the combustion zone with a combustion process that uses air as an oxidant.

In actual operation, all current combustion systems operate with some excess of air, which is necessary to assure the complete combustion of the fuel. This results in a lowering of the temperature of combustion. The greater this excess of air, the lower the temperature of combustion.

However, the greater the excess of air, the greater the flow rate of the produced flue gasses. Because the flue gas cannot be cooled to a temperature equal to the initial temperature of air, the quantity of heat rejected into the atmosphere by the flue gas increases with the increase in amount of excess air. This results in a reduction of the efficiency of the combustion system.

Therefore, in conventional combustion systems, in order to operate with a minimum excess of air, the tubes in which the boiling of the working fluid occurs, (so-called "waterwall" tubes), are located directly in the combustion zone. This allows the heat of the combustion to be partially absorbed by the boiling of the working fluid, and thus controls the temperature in the combustion zone. Such systems are known as conventional boiler combustion systems. These systems are, perforce, expensive and complex structures that require a high degree of maintenance, especially due to the fact that the waterwall tubes are subjected to very high thermal stresses.

Conversely, in fluidized bed combustors the excess of air is usually very high due to the fact that there is a substantial flow of air needed to maintain the fluidized bed. As a result, fluidized bed boiler/combustors have substantially reduced efficiencies.

In concept, it would be extremely desirable, and simpler, if combustion were to be performed in a separate combustion chamber without the need for internal cooling by waterwall tubes, while at the same time operating with a minimum excess of air. All heat produced by the combustion would thus be accumulated in a stream of hot flue gas which could then be utilized in a heat recovery steam generator or a heat recovery vapor generator, (HRSG or HRVG.) HRSG and HRVG systems are relatively simple heat exchangers which are substantially less expensive than conventional boilers. A combustion system with such a structure would be substantially more reliable and less expensive than a conventional boiler/combustion system.

But in such a case, the temperature in the combustion chamber would become so unacceptably high that the materials of the combustion chamber would be unable to withstand them. Moreover, the flue gases produced would have such a high temperature that they could not be used directly to provide heat to the heat exchangers of a power system, especially if these heat exchangers were intended to superheat vapor.

Separate combustion chambers, without internal waterwall cooling, have in some cases been used for the combustion of low quality fuels, particularly those with high water contents, such as biomass. However, even in these cases, the temperature of the flue gas produced is too high to be directly used in the heat exchangers of a power system.

Usually in such cases, the hot flue gas is used to heat an intermediate heat carrying fluid, which in its turn is then used to provide heat to the heat exchangers of the power system. However such an arrangement results in the addition of substantial complications to the entire system.

The intent of the RCSS is to act as a combustion system which allows the combustion of fuels with a minimum excess of air (in order to attain a high efficiency) and at the same time allows the effective control of temperatures in the combustion chambers and temperature of the produced flue gas (in order to allow direct utilization of heat in HRSG or HRVG type heat exchangers.)

There are three variants of the Kalex RCSS: The most basic variant of the RCSS allows for the preheating of the air used for combustion. The next variant of the RCSS allows for the same preheating as well as the production of hot water for district heating. The final variant of the RCSS allows for the preheating of air used for combustion and some drying of wet fuel. All three variants of the RCSS can work with FD fans (i.e., under slight pressure,) or with exhaust fans (i.e., under a slight vacuum.)

A computational flow diagram of a basic RCSS is given in figure 4 (attached.)

The RCSS operates as follows;

Atmospheric air with parameters as at point 560 enters into an air fan, FA, where its pressure is increased, obtaining parameters as at point 561. This air, with parameters as at point 651 is then sent into a recuperator, HE30, where it is heated in counterflow by a stream of returning flue gas, 553-555, and obtains parameters as at point 562. The air with parameters as at point 562 is then mixed with precooled flue gas having parameters as at point 570 (see below), forming an air/flue gas mixture with parameters as at point 571. This air-flue gas with parameters as at point 571 always contains a sufficient quantity of oxygen to provide for effective combustion of fuel. Stream 571 is then sent into the combustor. Any type of combustor can be use in the RCSS.

Combustors used with the RCSS are comprised only from a combustion chamber, having a wall of thermal resistant materials (i.e. refractive walls.) No tubes of any kind, with any sort of working fluid are used in such a combustion chamber. As a result, problems of the burning, blocking, or replacement of boiler tubes are utterly absent. This makes the maintenance of such a combustor extremely easy.

Meanwhile, fuel is fed into the combustor. In the diagram provided, for computational purposes, the fuel feed is shown as three separate feeds; dry, ash-free fuel shown as stream 587; ash, shown as stream 586; and moisture, shown as stream 585.

Due to the fact that heat released in the process of combustion is used, not only for the heating of newly produced flue gas, but for the heating of the precooled recirculating flue gas as well (stream 570, see above,) the temperature in the flame zone and the temperature of the flue gas produced is substantially reduced, while at the same time the volume and flow rate of the flue gas produced are substantially increased.

The flue gas produced in the combustor, with parameters as at point 600, then leaves the combustor and is sent into the power generation system (the CSQ system.)

It is evident that the temperature at point 600 can be easily controlled by increasing or decreasing the flow rate of the recirculating flue gas (stream 570.) The flow rate of the recirculating flue gas is chosen in such a way that the controlled temperature at point 600 is always lower that the temperature at the formation of NOx occurs.

The stream of flue gas with parameters as at point 600 passes through the heat acquisition heat exchangers of the power system (the CSQ system) where it is cooled and obtains parameters as at point 550. Thereafter, stream 550 is divided into two substreams with parameters as at points 551 and 552 correspondingly. The flow rate of stream 551 is the same as the flow rate at point 570. This flow rate is dictated by the requirement to control the temperature at point 600.

The stream with parameters as at point 551 enters into a recirculating fan, FR, where its pressure is slightly increased, and obtains parameters as at point 570 (see above.)

The stream with parameters as at point 552 represents the stream of flue gas newly created in the process of combustion. This stream is then redesignated as stream 553, and is sent into HE30 where it is cooled, providing the heat necessary for the air preheating process (561-562.)

The stream of returning flue gas, (553-555) can, in principle, be cooled to a temperature slightly exceeding the temperature at which the precipitation of moisture would begin. The quantity of heat that would be available from the flue gas due to such cooling is substantially higher than the quantity of heat which is necessary for the preheating of the air in process 561-562 (see above.) Therefore, if this stream of flue gas (555) is sent into the stack, then some portion of usable heat will be lost, thus reducing the overall efficiency of the system.

However, this excess of heat can be utilized for drying and preheating fuel. Such an arrangement is incorporated the RCSS variant designated RCSS-DR, and is shown in **figure 4a** (attached.)

In RCSS-DR, the stream of flue gas with parameters as at point 555 is sent into a fuel drier, D1, where it is further cooled and absorbs moisture from the fuel, and then finally, having obtained parameters as at point 557, is sent into the stack.

Appendix D:
Prevention of Nitridation

Nitridation (or nitriding) 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).

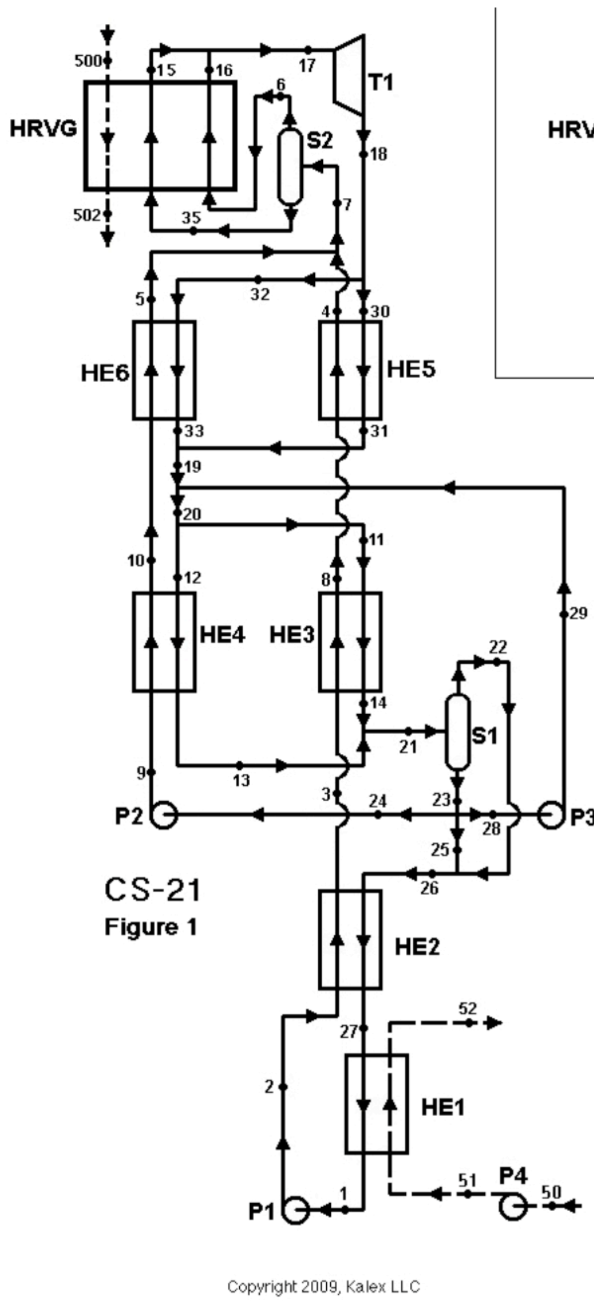
Sulfuric corrosion is **not** a risk using this method because the concentration of sulfur used is far below threshold concentration at which sulfuric corrosion can begin to occur. None the less, additional catalytic poisons may be added to secure against any hypothetical possibility of sulfur corrosion

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.

Figures 1 & 1a:
System CS-21



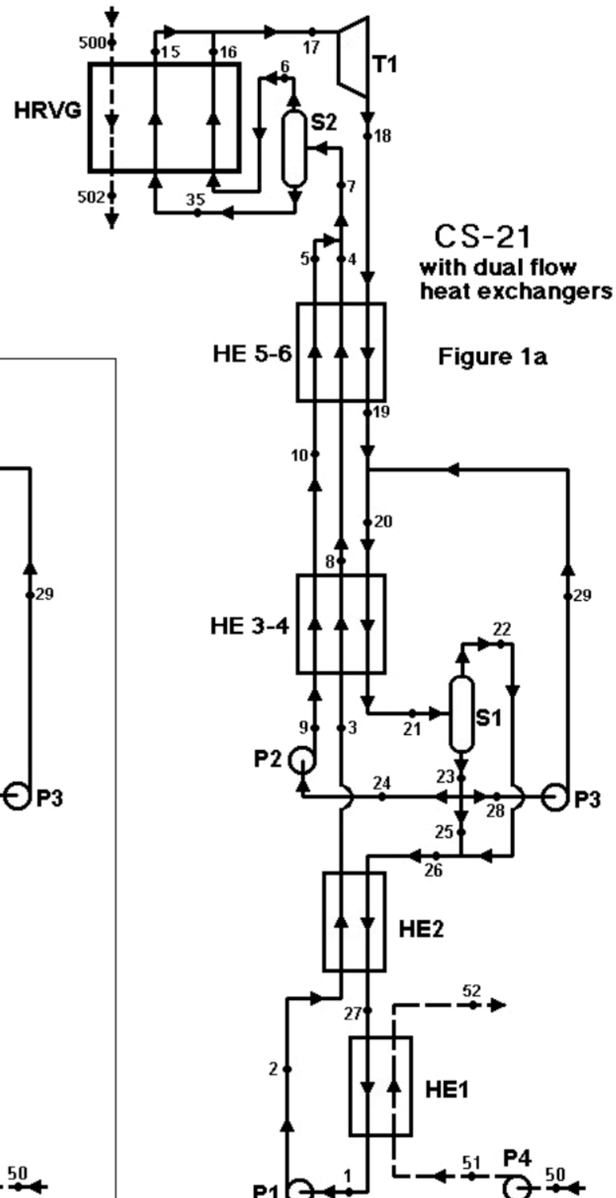


Figure 2:
System CSQ-21f

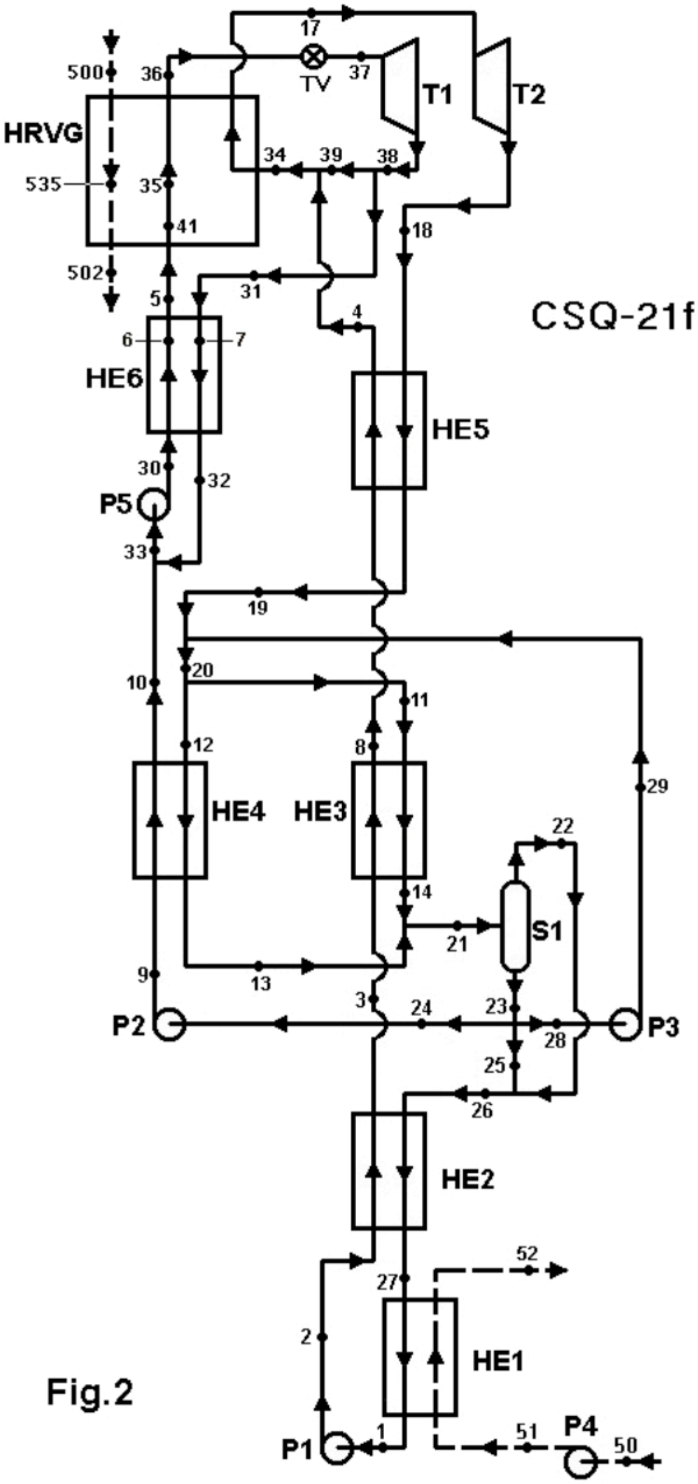
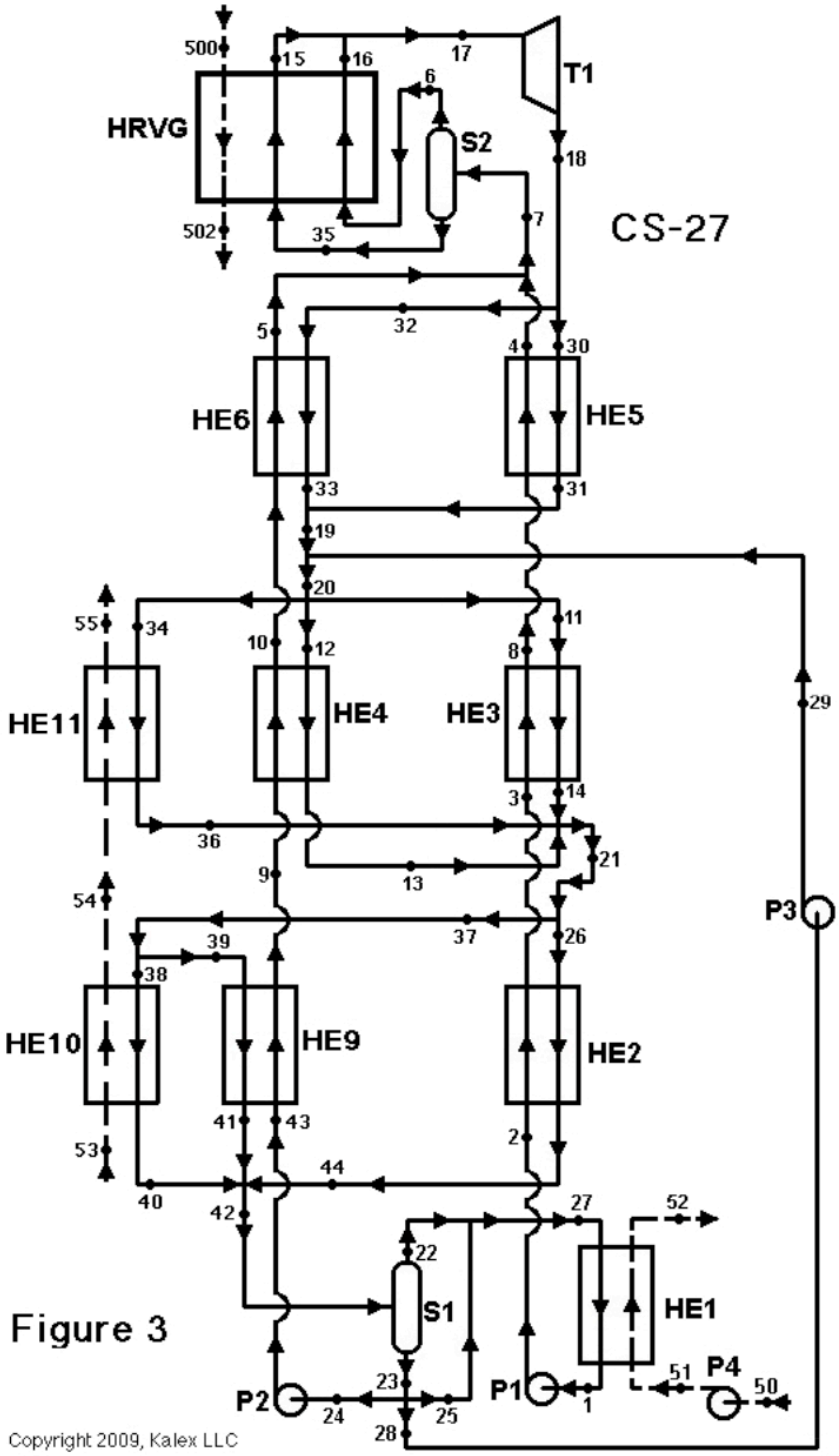
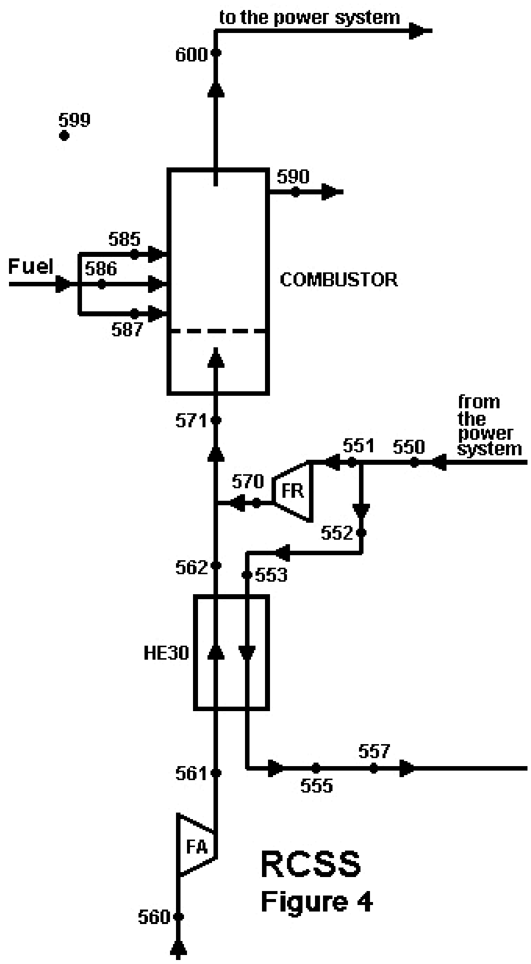


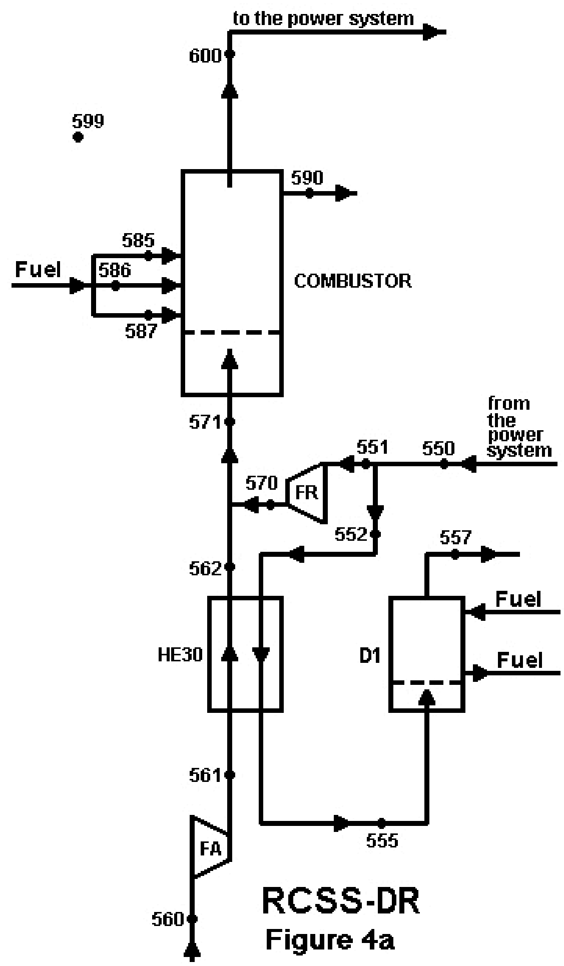
Fig.2

Figure 3:
System CS-27



Figures 4 & 4a:
RCSS





599; adiabatic flame point.
590; ash.
585; moisture.
586; ash.
587; dry ash-free fuel.

For further information, please see Kalex LLC's website, at **www.kalexsystems.com**
Technical and business inquiries may be sent to **information@kalexsystems.com**