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A combined power and cooling cycle modified to improve resource utilization efficiency using a distillation stage

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Abstract

A combined power and cooling cycle is being investigated. The cycle is a combination of a Rankine cycle and an absorption refrigeration cycle. The vapor exiting the turbine in this cycle is cold enough to extract refrigeration output. This combined cycle is being proposed for application with lower temperature heat sources such as solar, geothermal, and industrial waste heat with the primary objective of producing power. In this paper, the goal is to optimize the cycle configuration for maximum resource utilization efficiency (RUE). Based on an exergy analysis, the cycle configuration has been modified to improve the RUE. A thermal distillation scheme, similar to those found in some Kalina cycles has been implemented in the cycle. An optimization method is used to maximize the RUE of the modified configuration. A significant improvement in the efficiency of more than 25% was achieved with the improved configuration. Larger pressure ratios are obtained across the turbine. Increased efficiencies can also be obtained for the cases where only work output is desired.

1. Introduction

A combined cycle yielding power and refrigeration for low-temperature heat sources was proposed by Goswami [1] in 1995. The cycle can be considered to be a combination of the Rankine cycle and an absorption refrigeration cycle. The proposed cycle is suitable as a bottoming cycle using waste heat from conventional processes or utilizing relatively low-temperature solar or geothermal renewable resources.

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Nomenclature

E	exergy (kW)
h	specific enthalpy (kJ/kg)
$\dot{m}$	mass flow rate (kg/s)
s	specific entropy (kJ/kg K)
T	temperature (K)
W	work output (kW)
η	efficiency
Subscripts	
c	cooling
hs	heat source
in	input
r	resource utilization efficiency
ref	refrigeration cycle
I	first law
II	second law

A binary mixture of ammonia and water has been found to be a good thermodynamic choice as the working fluid and is used in the initial studies.

Fig. 1 shows a schematic diagram of the cycle. A mixture of ammonia and water is pumped to a high pressure (state 2) and partially boiled in a boiler. The vapor generated (state 5) consists mostly of the more volatile component, ammonia. A rectifier-condenser is used to increase the concentration of the volatile component in the vapor, by partially condensing water out of the vapor from the boiler. The resulting purified vapor (state 7) is superheated (state 8) and expanded in a turbine to low temperatures (state 9). This low-temperature vapor can be used to obtain cooling (states 9 and 10). The weak ammonia solution from the boiler (state 11) is passed through a heat exchanger to recover heat from it, and throttled back to the absorber (states 11–14). The vapor and the weak solution are used to regenerate the strong solution in the absorber along with the rejection of heat from the cycle. The condensed liquid in the rectifier is also throttled back into the absorber in this schematic. Alternatively, it could be recirculated back to the boiler.

Detailed discussions on the performance of this new cycle and an optimization method used to improve cycle output are discussed in past publications [2–6]. Simulation results [6] show that depending on the heat source temperatures, as much as half of the output may be obtained as refrigeration under optimized conditions, with refrigeration temperatures as low as 205 K being achievable. Analysis performed with organic working fluids showed that ammonia–water mixture is still a superior choice [7].

2. Efficiency definition

Defining an efficiency for a combined power and cooling cycle such as this is not straightforward, since the output consists of two different quantities—power and refrigeration, while the only input is

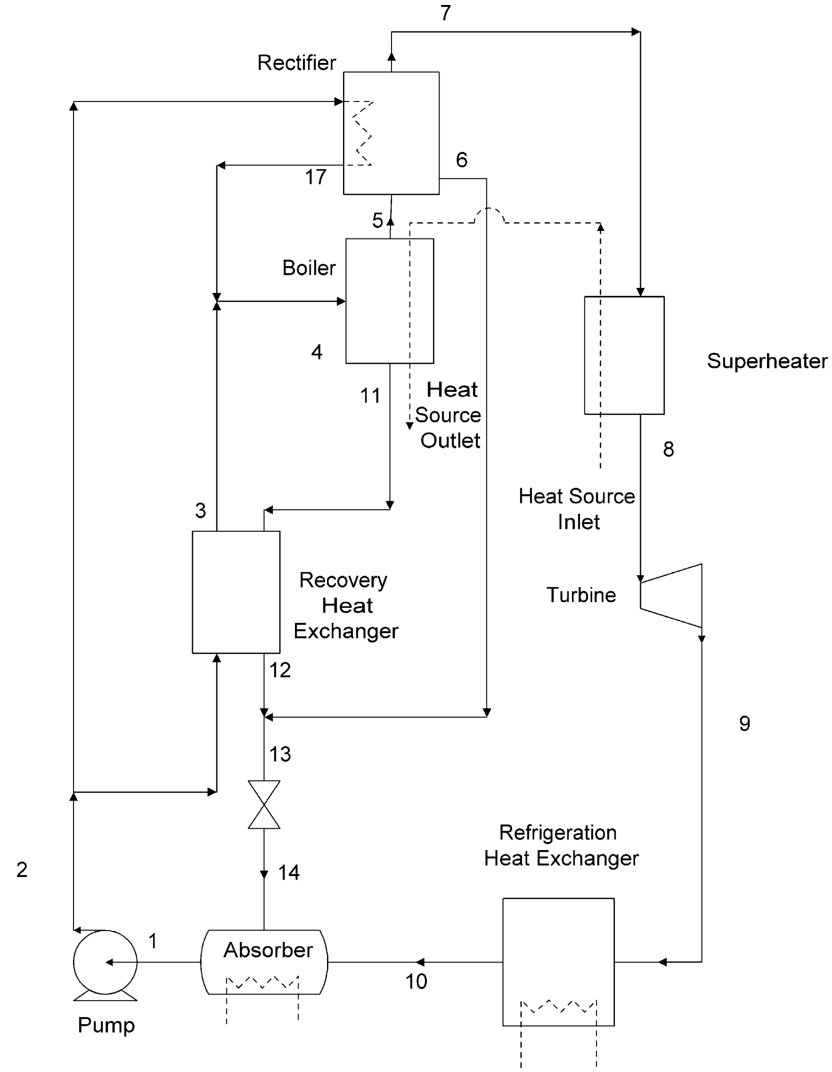

Fig. 1. The basic cycle configuration.

heat. A proposed second law efficiency for this cycle is the resource utilization efficiency (RUE) defined as [8]:

$$\eta_{\mathrm{r}} = (W_{\mathrm{net}} + E_{\mathrm{c}})/(E_{\mathrm{hs,in}}) \tag{1}$$

Here E_c is the exergy associated with the cooling output, while $E_{hs,in}$ refers to the exergy available in the heat source fluid. The definition above accounts for the cooling output using the equivalent exergy associated with it. The exergy of cooling corresponds to the minimum work required to produce that amount of cooling, which would be the case if a reversible refrigeration cycle were to be used.

However, this would not be true in practice. A better definition of the second law efficiency then would use an effective RUE defined as [8]

$$\eta_{r,eff} = (W_{net} + E_c/\eta_{II,ref})/(E_{hs,in}) \tag{2}$$

Here the exergy of cooling is weighted with a second law efficiency of refrigeration, which can be chosen so as to realistically account for the refrigeration output. For example, a typical second law efficiency value of a vapor compression refrigeration system could be a possible choice for $\eta_{II,ref}$. Similar definitions can be used for the first law efficiency also

$$\eta_I = (W_{net} + E_c)/(Q_{in}) \tag{3}$$

$$\eta_{I,eff} = (W_{net} + E_c/\eta_{II,ref})/(Q_{in}) \tag{4}$$

3. Method of analysis

An optimization program coupled with the cycle simulation program is used to determine the optimum RUE values. The cycle is simulated using simple energy and mass balances across the devices. Pressure drops and heat losses are neglected in the calculations. The working fluid leaving the absorber is assumed to be a saturated liquid and the temperature is set at 5 K above the ambient, which fixes the working fluid state there. Equilibrium conditions are assumed to exist in the boiler and rectifier to calculate the state points. Isentropic efficiencies of 0.85 and 0.80 were assumed for the turbine and pump, respectively. A detailed description of the simulation process can be found elsewhere [9]. Ammonia–water mixture properties are calculated using the method described by Xu and Goswami [10]. The cycle simulation program calculates all the state points in the cycle and then evaluates the objective function and the constraint functions. The objective function in this paper consisted of the RUE (Eq. (1)) or effective RUE (Eq. (2)).

Constraint functions are used to keep the cycle operating parameters physically feasible. Minimum approach and pinch point temperatures of 5 K are specified for heat exchangers in the constraint equations. The turbine vapor exit condition is constrained to keep the vapor more than 90% dry. A full list of constraints is given in Table 1. In addition, an ambient temperature of 298 K was assumed, which

Table 1
Constraints used in the optimization

Description of constraint	Lower limit	Upper limit
Measure of superheating	-1×10^{30}[a]	0
Heat exchanger approach temperatures	5	1×10^{30}
Pinch point for boiling	5	1×10^{30}
Boiler reflux mass flow rate	0	1×10^{30}
Vapor mass fraction at turbine exit	90	100
Vapor mass fraction at boiler exit	1×10^{-5}	100
Liquid mass fraction	1×10^{-5}	100
Cycle cooling	0.1	1×10^{30}
Efficiency percentage (objective)	0	100

[a] Limits of 1×10^{30} and -1×10^{30} denote infinity.

was also used as the dead state temperature in all exergy calculations. A 270 K temperature limit was set at the turbine exit for useful cooling output. The working fluid is assumed to exit the refrigeration heat exchanger at 5 K below ambient.

The optimization of the cycle is carried out using a commercially available program called GRG2, which is based on the GRG (Generalized Reduced Gradient) method. GRG is a search method, which requires the specification of an initial guess, which is used as a starting point for the search. The search proceeds in the direction of an improved value for the objective function until either a mathematical termination condition, such as the Kuhn–Tucker condition is satisfied, or when the search algorithm is unable to find a new feasible point where the objective function is improved. While the search will almost always terminate at a point that is an improvement over the initial guess, it is possible that the termination occurs at a local optimum. With several initial guesses and sensitivity analysis, it is possible to be reasonably confident that a global optimum has been obtained. The theory of the GRG method is covered in detail in standard textbooks [11].

The cycle has been found to work best at lower heat source temperatures. At higher heat source temperatures, expansion through the turbine does not get the vapor to cold enough temperatures. The target heat sources for this cycle include low and medium temperature solar collectors, geothermal heat as well as low-temperature waste heat. Based on these criteria and past work [2–6], heat source temperatures of 360, 400, 440 and 480 K were selected. The heat source fluid was assumed to have the properties of water at the corresponding saturation pressure.

4. Basic cycle configuration optimization

The basic cycle configuration, shown in Fig. 1, was optimized for maximum resource utilization efficiency (RUE) using Eqs. (1) and (2) as the objective functions. The RUE is a good choice in evaluating the second law efficiency in applications, such as geothermal power where the energy in the geofluid is lost once it leaves the cycle. Optimizing for RUE ensures that maximum use is made of the energy source or fuel.

The RUE of the cycle (Eq. (1)) can be written as:

$$\eta_{\mathrm{r}} = \frac{(W_{\mathrm{net}} + E_{\mathrm{c}})}{\dot{m}_{\mathrm{hs}}[h_{\mathrm{hs,in}} - h_0 - T_0(s_{\mathrm{hs,in}} - s_0)]} \tag{5}$$

The specific enthalpy h_{o} and specific entropy s_{o} are calculated at the ground state, while the specific enthalpy and entropy of the heat source are calculated at the inlet conditions. It can be seen from Eq. (5) that in order to maximize the RUE at a heat source inlet temperature, the numerator has to be increased, while the mass flow rate of the heat source will be decreased. The objective function, therefore, is quite sensitive to the flow rate of the heat source.

The results for four different cases are plotted in Fig. 2. It is seen that the RUE calculated from Eq. (1) results in the lowest maximum values. Using Eq. (2), where a larger weight is assigned to the refrigeration output by considering the second law efficiency of a refrigeration cycle, results in higher maximum efficiency values. A lower value of $\eta_{\mathrm{II,ref}}$ in Eq. (2) assigns more weight to the cooling output and therefore, higher effective RUE. Optimizing for the case where only work output is required from the basic configuration gives the best values for RUE at all the heat source temperatures

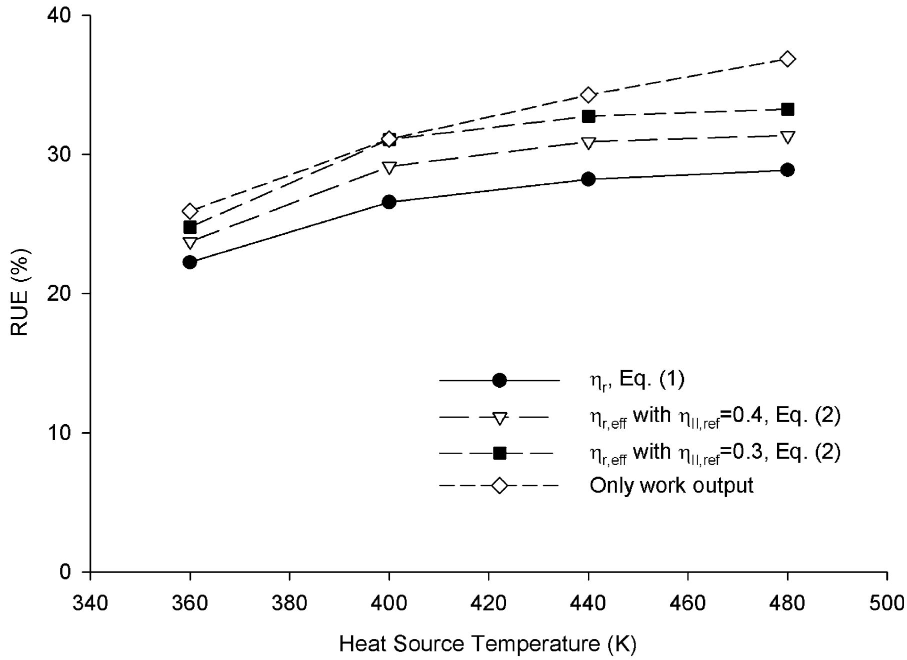

Fig. 2. Optimized RUE for the basic cycle configuration.

considered. Note that the cycle low pressure was constrained to be less than 5 bar for the work maximization simulations. Without this constraint, the optimum conditions are seen to be at conditions where the cycle operates using nearly pure ammonia as a Rankine cycle. The absorber model used is not valid at those conditions. Further, the pressures in the cycle are too high to be practical at those conditions.

Overall, the RUE values obtained at optimized conditions are quite low. An analysis was performed to study the exergy destruction in the cycle. For the conditions of optimized RUE using Eq. (1), the major exergy losses in different parts of the cycle are plotted in Fig. 3. The largest loss is through the unrecovered exergy of the heat source fluid leaving the cycle. Exergy destruction during the heat addition is high pointing to a large ΔT between the hot and cold streams in the heat exchangers. Heat rejection in the absorber is accompanied by irreversible mixing leading to large exergy destruction.

In order to get low temperatures at the turbine exit in this cycle, the vapor has to have a high concentration of ammonia. This limits the fraction of working fluid vaporized in the boiler and consequently the temperature glide of the working fluid (Fig. 4). Further, a large quantity of hot weak solution returns from the boiler to the absorber. Much of the energy in this stream is used to preheat the strong solution in the recovery heat exchanger. As a result, the basic solution enters the boiler at a high temperature (usually slightly above the bubble point—Fig. 4).

Referring to Eq. (5), at maximum RUE conditions, the mass flow rate of the heat source is minimized. Consequently, a pinch point exists at the heat source exit of the boiler. The heat source exit temperature is limited by the working fluid inlet temperature, which is high in this configuration. Higher vapor fractions in the cycle will lower the weak solution mass flow rate and lower the boiler inlet temperature. Therefore, optimum conditions tend towards higher strong solution concentration (higher absorber pressure) and a lower boiler pressure resulting in low cycle pressure ratios (Fig. 5). The poor temperature match in the boiler (Fig. 4) leads to large exergy destruction during heat addition (Fig. 3).

When the objective function is defined to only consider the work output in the RUE, high ammonia concentration in the vapor is no longer important. Therefore, at optimum conditions, the vapor fractions and temperature glide in the boiler are higher, rectification is not required and boiler temperature match is improved. This can be seen from Figs. 4 and 6 which show the temperatures and vapor fraction in the boiler at optimized RUE conditions with and without cooling. Note that the pinch point in the boiler is at the heat source exit and the heat source inlet temperatures are plotted on the abscissa in Figs. 4 and 6.

The inflection in Fig. 6 on increasing the heat source temperature from 400 to 440 K is due to a shift in the optimum parameters. If the trend of increasing vapor fraction seen between 360 and 400 K heat source temperatures is to continue, there would be insufficient weak solution to preheat the strong solution close to the bubble point before the boiler. Therefore, the heat source fluid would have to preheat the strong solution. However, this condition requires a larger heat source flow rate which lowers the RUE of the cycle.

Fig. 7 shows the corresponding first law efficiency at optimized RUE conditions. The values are seen to be fairly low, even considering the low heat source temperatures. First law efficiencies tend to be low when a cycle is optimized for RUE. However, in this case the efficiency is even lower because of the low-pressure ratios in the turbine.

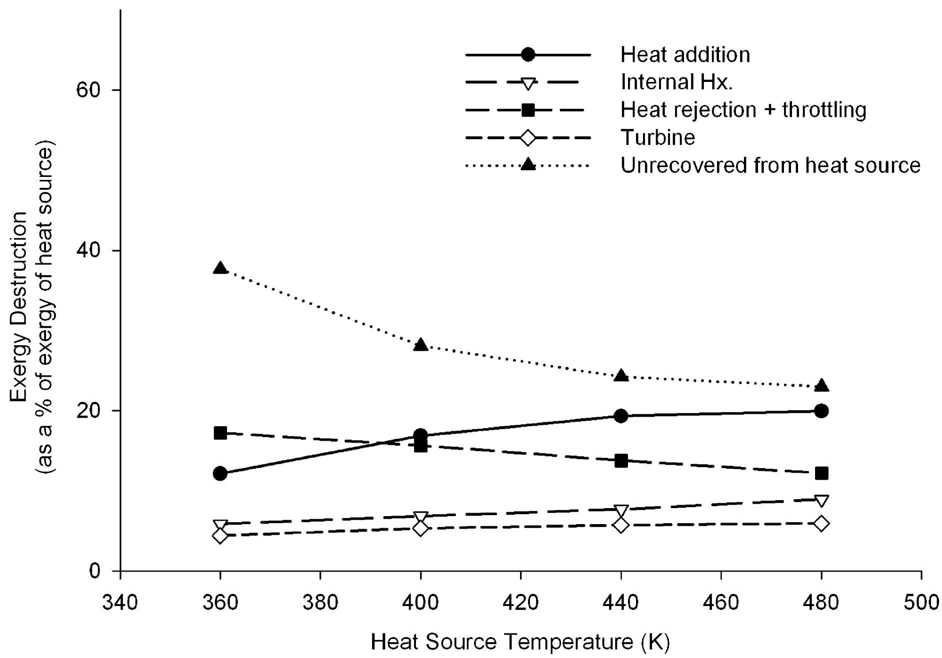

Fig. 3. Exergy destruction in the basic cycle at optimized RUE case with combined power and cooling output.

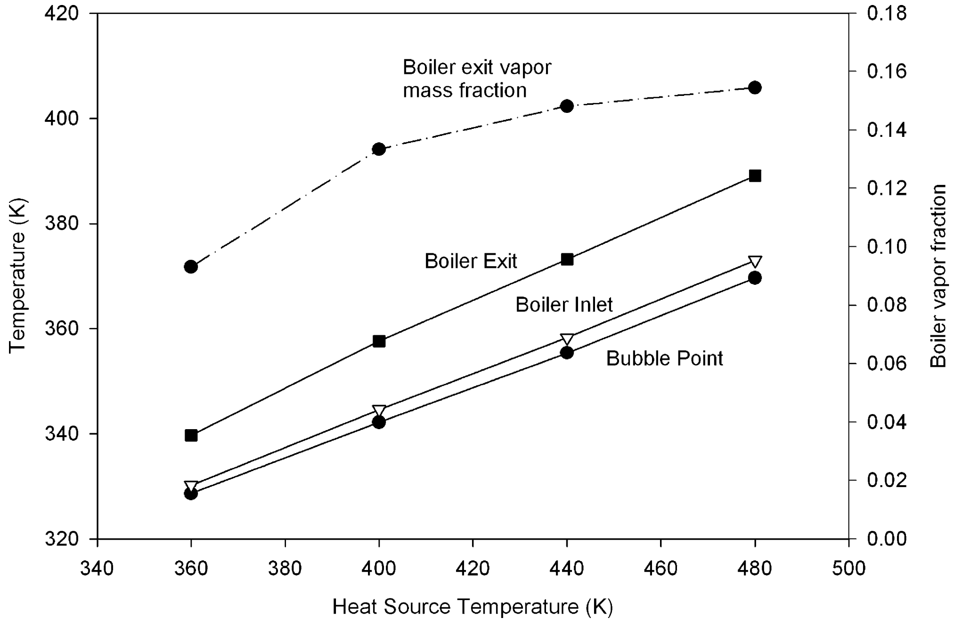

Fig. 4. Working fluid temperatures and vapor fractions in boiler: optimized for maximum RUE and with cooling. Due to the pinch point, the heat source boiler exit temperature is 5 K more than the boiler inlet temperature.

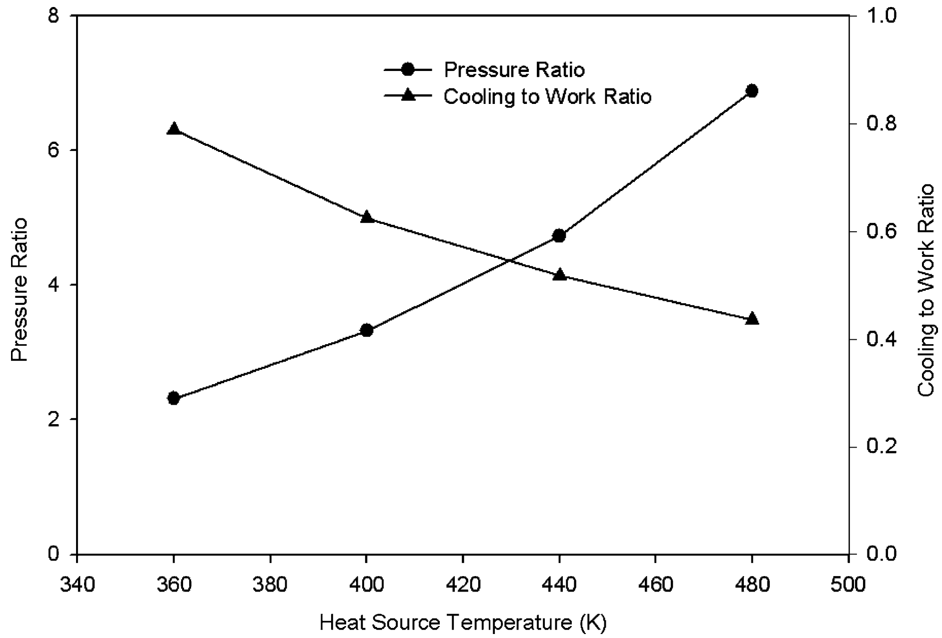

Fig. 5. Pressure ratio and cooling to work ratio in the basic cycle configuration for the case: optimized for maximum RUE with cooling.

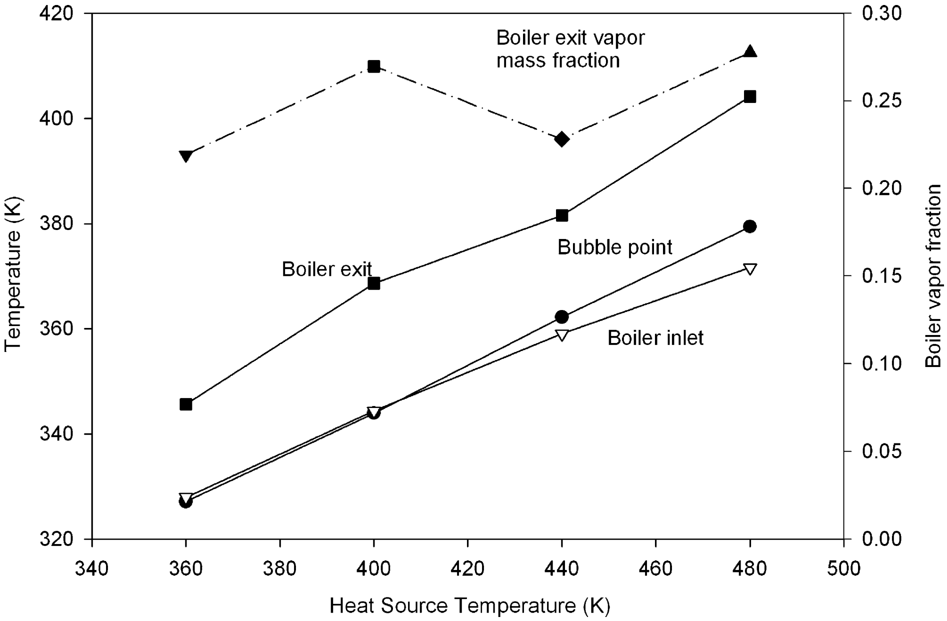

Fig. 6. Working fluid temperatures and vapor fractions in boiler: at maximum RUE with pure work output. Due to the pinch point, the heat source boiler exit temperature is 5 K more than the boiler inlet temperature.

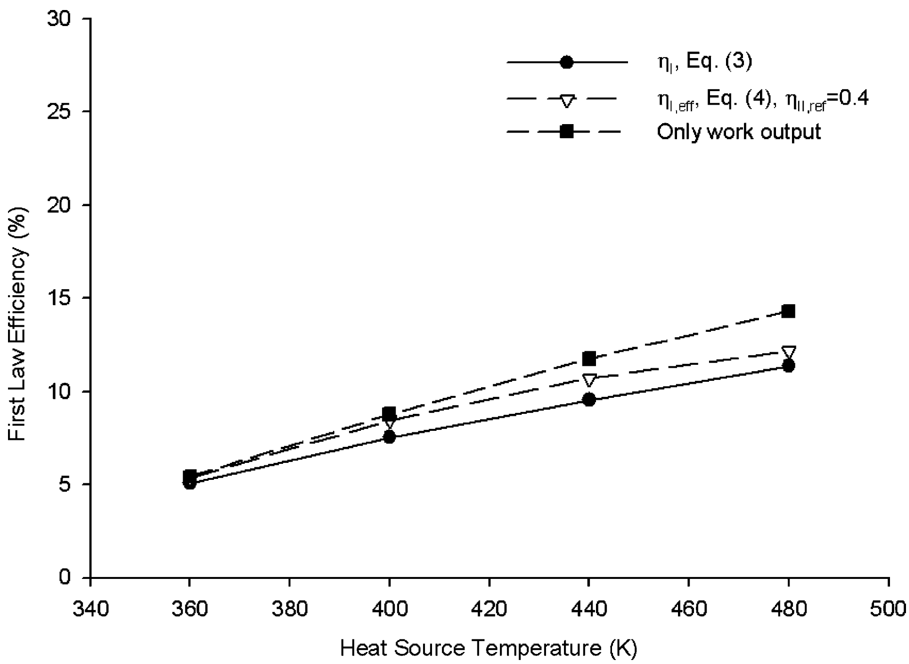

Fig. 7. First law efficiency at optimum RUE conditions.

5. Improved configurations for higher RUE

The maximum RUE of the basic cycle configuration is limited by two conflicting requirements. A low turbine exit pressure is desirable in order to increase the pressure ratio across the turbine (increase work) and to achieve low temperatures at the turbine exit (increase cooling). A higher absorber pressure is also required in order to have a strong solution with a higher ammonia concentration to generate more vapors with a low-temperature heat source. Therefore, a method of decoupling the absorber pressure from the turbine exit pressure should improve the optimum RUE of the cycle. Accomplishing this using a jet pump in which the high pressure weak solution is used to pump vapor into a higher pressure absorber, was studied by the authors [12]. However, improvements were found to be small. Alternate methods of achieving the same effect, using distillation (thermal compression) methods are described in the following sections.

5.1. Distillation (thermal compression) methods

Kalina [13] proposed several cycles using ammonia–water mixtures and featuring extensive internal heat exchange. In several of these cycles, a distillation arrangement is used as part of the heat recovery. As an example, hot vapor from the turbine exhaust is used in a distiller to generate ammonia-rich vapor. The vapor is then used to increase the concentration of the strong solution going to the boiler. This method has been called a ‘thermal compression’ method by some authors. Two different cycle configurations are proposed here that incorporate the ‘thermal compression’ feature in the combined power and cooling cycle.

5.1.1. Configuration I—using heat source

The cycle configuration discussed in this section is shown in Fig. 7. This configuration has an additional absorber, recovery heat exchanger, pump and a distiller when compared with the basic cycle (Fig. 1). Some of the ammonia–water mixture in the low-pressure absorber is pumped into the high-pressure absorber at a slightly higher pressure (state 23). The remaining solution is sent to the distiller where the heat source exiting the boiler is used to generate vapor rich in ammonia (state 22). This vapor is routed into the high-pressure absorber where it recombines with the solution pumped there to generate a higher concentration strong solution (State 1). This strong solution is used in the boiler, turbine and refrigeration heat exchanger (states 6–11) as in the basic cycle. The weak solution from the boiler (state 12) is routed back to the low-pressure absorber through a recovery heat exchanger. The liquid return streams from the boiler and distiller (states 15 and 21) recombine with the vapor (state 11) in the LP absorber. In this configuration, the turbine exit pressure is close to the low-pressure absorber pressure, which limits the solution concentration at state 16. The boiler, however, gets a higher concentration solution that is limited by the pressure at state(1). Also note that the heat addition in the cycle can take place in the boiler and superheater, as well as the distiller.

The optimized RUE values obtained using this cycle configuration is plotted with heat source temperature in Fig. 8. Three objective functions were used:

1. RUE using Eq. (1), or in other words, Eq. (2) with $\eta_{\mathrm{II,ref}}=1$
2. Eq. (2) with $\eta_{\mathrm{II,ref}}=0.4$
3. Eq. (1) with only work output considered (work domain).

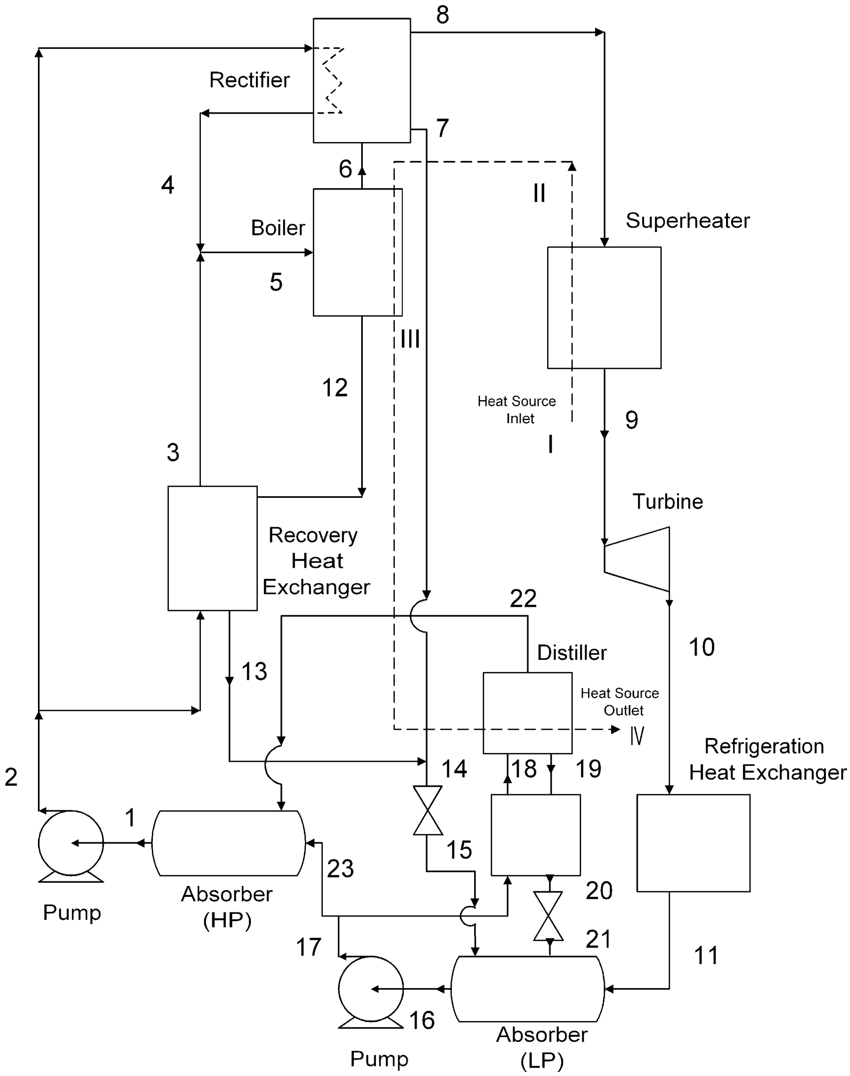

Fig. 8. Cycle configuration using the heat source to produce vapor in the distiller.

Corresponding optimized values of the basic cycle configuration are also plotted for comparison. The modified configuration improves the optimum RUE of the cycle, with the largest improvement being seen at lower heat source temperatures. Again the optimum RUE in the work domain is substantially higher. Just as for the basic cycle configuration, the requirement of expanding to low temperature in the turbine requires a vapor with high ammonia fraction and lower temperatures at the turbine inlet leading to additional irreversibilities (Fig. 9).

Since the heat source is used in the distiller following the boiler, the temperature of the fluid leaving the cycle is lower than in the basic cycle. Hence, the exergy lost unrecovered from the heat source is much lower as seen in Fig. 10. Figs. 10 and 11 show the exergy destruction in the major processes in

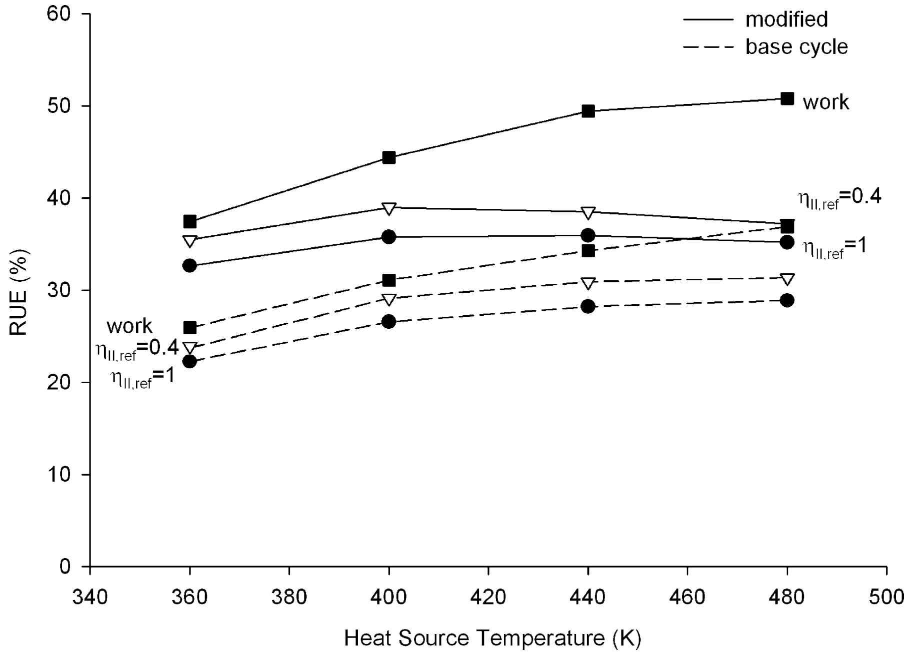

Fig. 9. Maximum RUE of the cycle configuration with heat source fluid powered thermal compression modification.

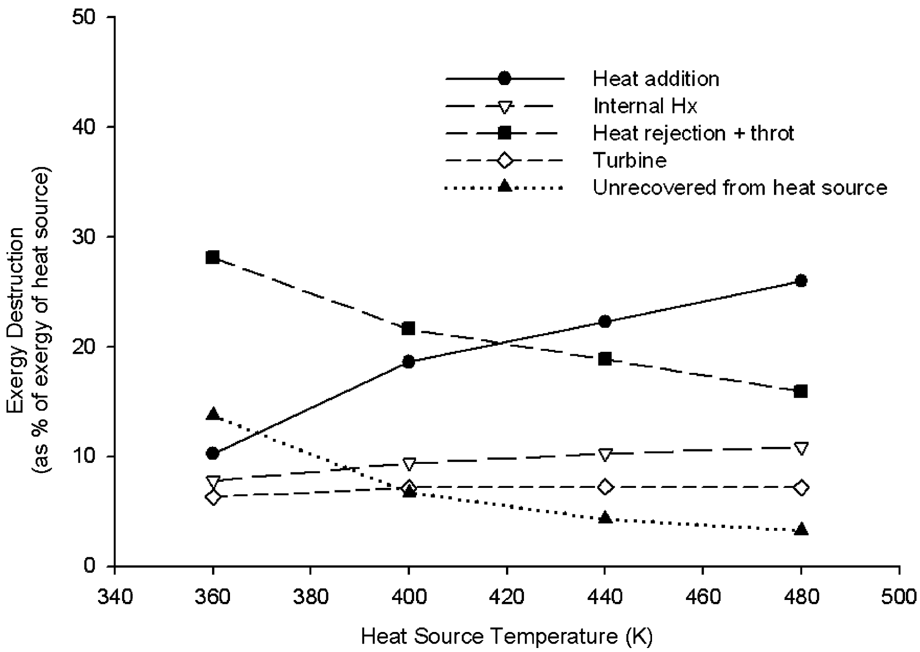

Fig. 10. Exergy destruction in the cycle with heat source powered thermal compression modification, when operated to provide power and cooling.

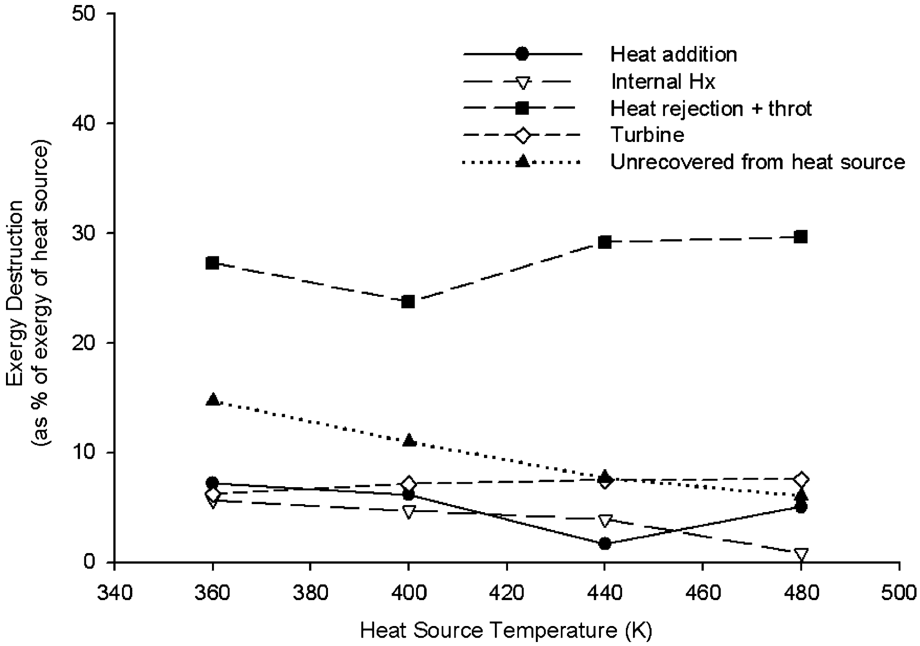

Fig. 11. Exergy destruction in the cycle with heat source powered thermal compression modification, when operated to provide only power output.

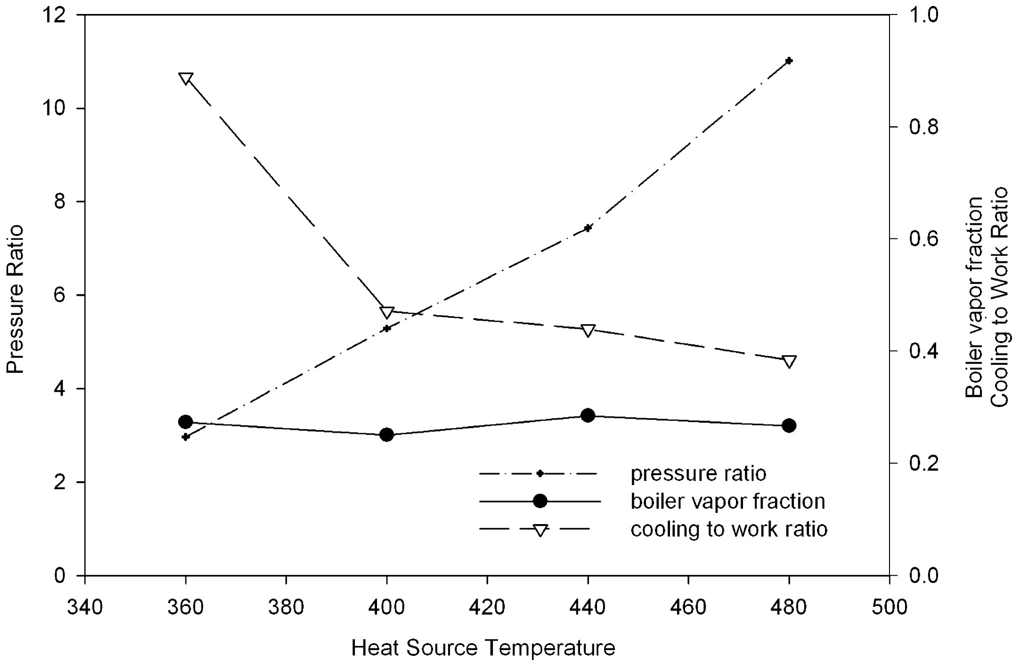

Fig. 12. Pressure ratio, boiler vapor fraction and cooling to work ratio for optimized RUE conditions in the cooling domain for the cycle configuration discussed in Fig. 8.

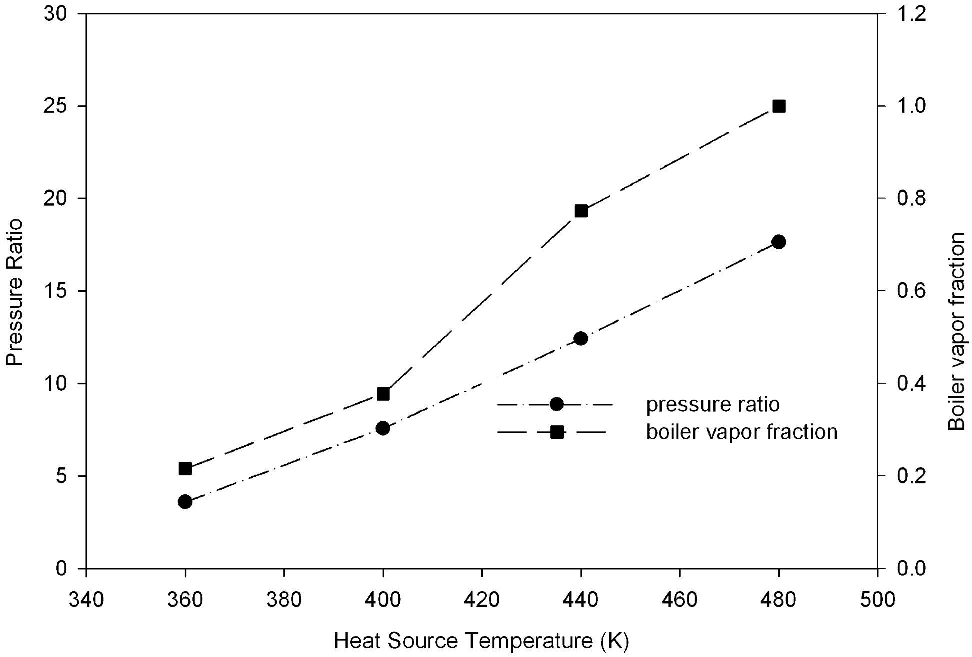

Fig. 13. Pressure ratio and boiler vapor fraction for optimized RUE conditions in the work domain for the cycle configuration discussed in Fig. 8.

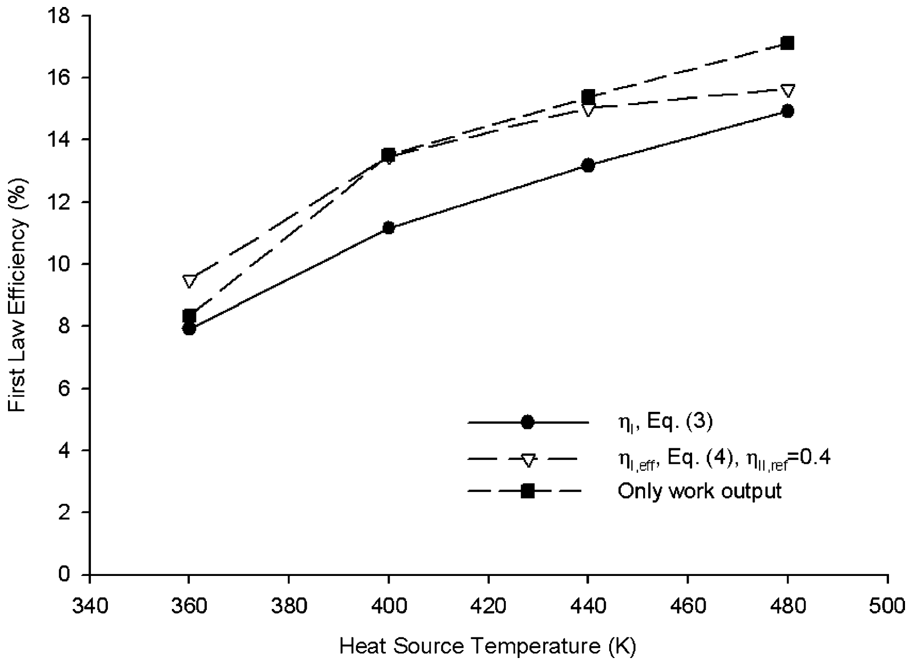

Fig. 14. First law efficiency at maximum RUE conditions for configuration with heat source powered thermal compression modification.

the cycle optimized for RUE (Eq. (1)) with and without cooling output, respectively. The boiler vapor fractions in the modified configuration are up to three times that in the basic cycle along with increased pressure ratio across the turbine (Fig. 12). The larger vapor fraction results in higher boiler exit temperatures and better temperature matching in the boiler. While the exergy destruction in the boiler itself (data not shown) is lower in the modified configuration, since heat is added in the distiller also, the total exergy destruction during heat addition is larger.

Fig. 13 shows the pressure ratios and boiler vapor fractions for the optimum conditions in the work domain. The vapor fractions observed are substantially higher in the work domain, since the vapor does not have to be expanded to low temperatures and therefore, can have a lower concentration of ammonia.

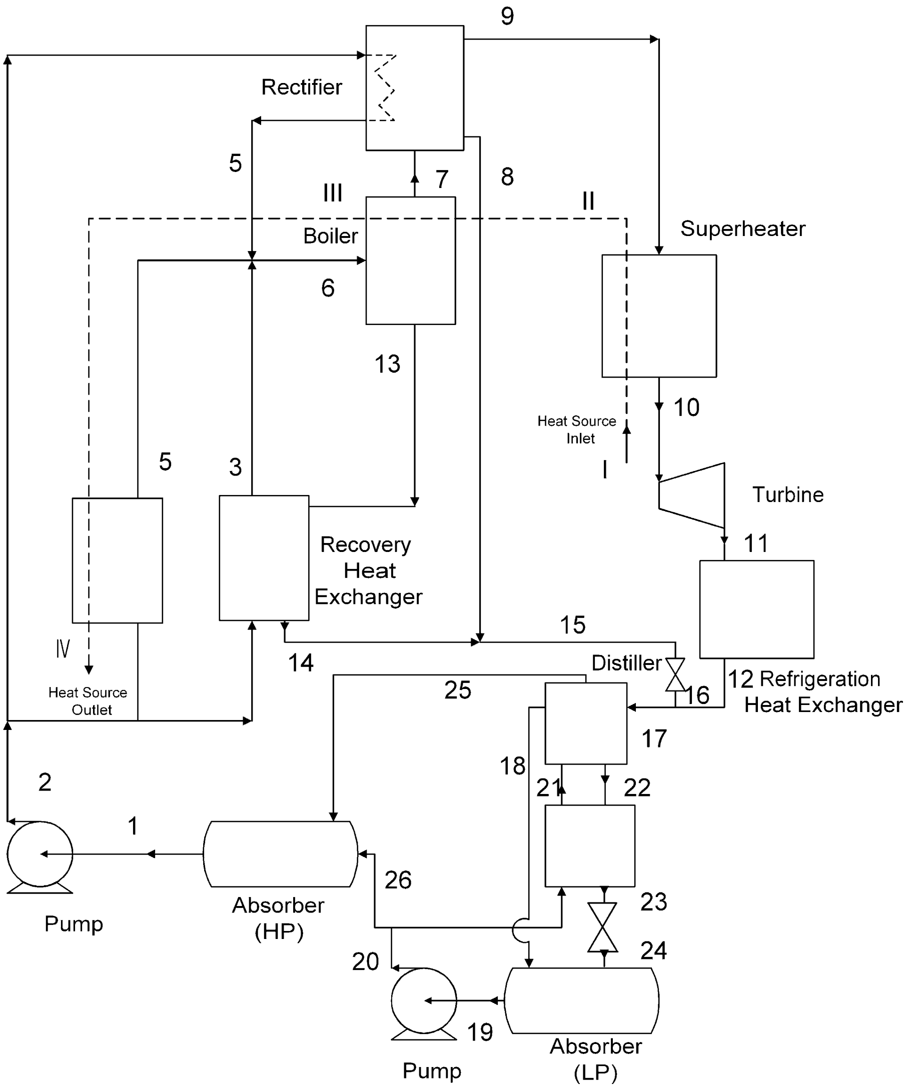

Fig. 15. Cycle configuration using heat of condensation to produce vapor in the distiller.

In fact, at higher temperatures (440 and 480 K), it is seen that most of the strong solution is boiled off in the boiler. The corresponding first law efficiency of the modified cycle at optimum RUE conditions (Fig. 14) also shows a significant improvement over the basic cycle configuration, due to the higher pressure ratios in the turbine and the resulting additional expansion leading to larger work output.

5.1.2. Configuration II—using absorber heat recovery

The heat supply in the distiller can come from different sources. In the previous configuration, heat was supplied by the heat source. The weak solution is another available source, however, pinch point effects limit the potential vapor generation using this sensible heat source. However, if the weak solution were to be mixed with the vapor, the heat of condensation of the two-phase mixture would match well with the requirements in the distiller. The challenge is in mixing the vapor and liquid streams. A jet pump like device (without a diffuser section) could be used to accomplish the mixing. Other configurations possible include absorber designs built so as to partially recover the heat of condensation from it such as in a GAX cycle.

In the simulations performed for the current configuration, the mixing process is modeled as isenthalpic. A diagram of this configuration is shown in Fig. 15. The optimized RUE values are plotted for the new configuration in Fig. 16. This configuration is also seen to be an improvement over the basic

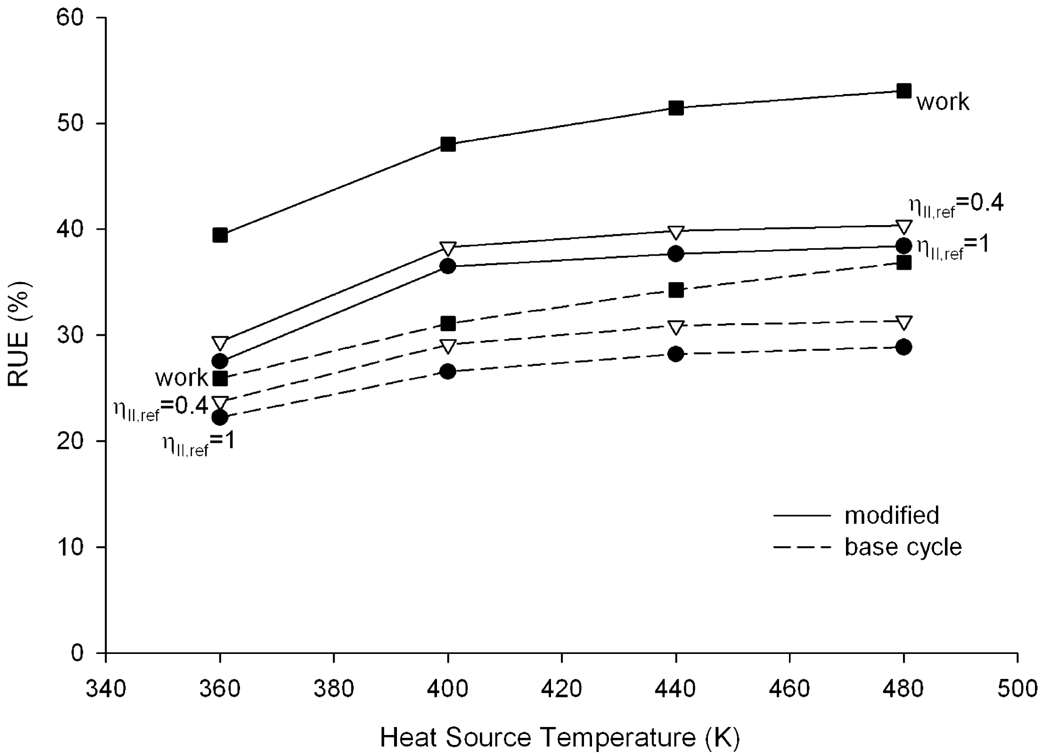

Fig. 16. Maximum RUE of the cycle configuration modified with a condensing mixture providing heat of distillation.

cycle. Figs. 16–21 plot the performance and trends seen with this configuration. The overall effect of a thermal compression stage is similar in this case also.

In the cooling domain, using the heat source for the thermal compression gives higher RUE at lower source temperatures while the present configuration has slightly better RUE at higher temperatures. The improvement in pressure ratios over the basic cycle is lower in this case. The first law efficiency is poor and is comparable to the values obtained for the basic cycle configuration. The larger specific turbine output due to the increase in pressure ratio does not sufficiently offset the additional heat input in the preheater.

In the work domain, the optimum efficiency of the configuration using the condensing mixture as the heat source is better at all heat source temperatures. The boiler vapor fraction at optimum conditions is very large at 440 and 480 K. This means that there is very little weak solution stream returning from the boiler and the heat input in the distiller is predominantly from hot vapor exiting the turbine. Therefore, at higher heat source temperatures (greater than 480 K) in the work mode, once the redundant components such as the rectifier, recovery heat exchanger, and refrigeration heat exchanger are removed, optimized cycle configuration is roughly a Kalina cycle incorporating distillation condensation.

Examples of cycle state points at optimized conditions for all the configurations discussed in this paper can be found in Appendix B of Vijayaraghavan [9].

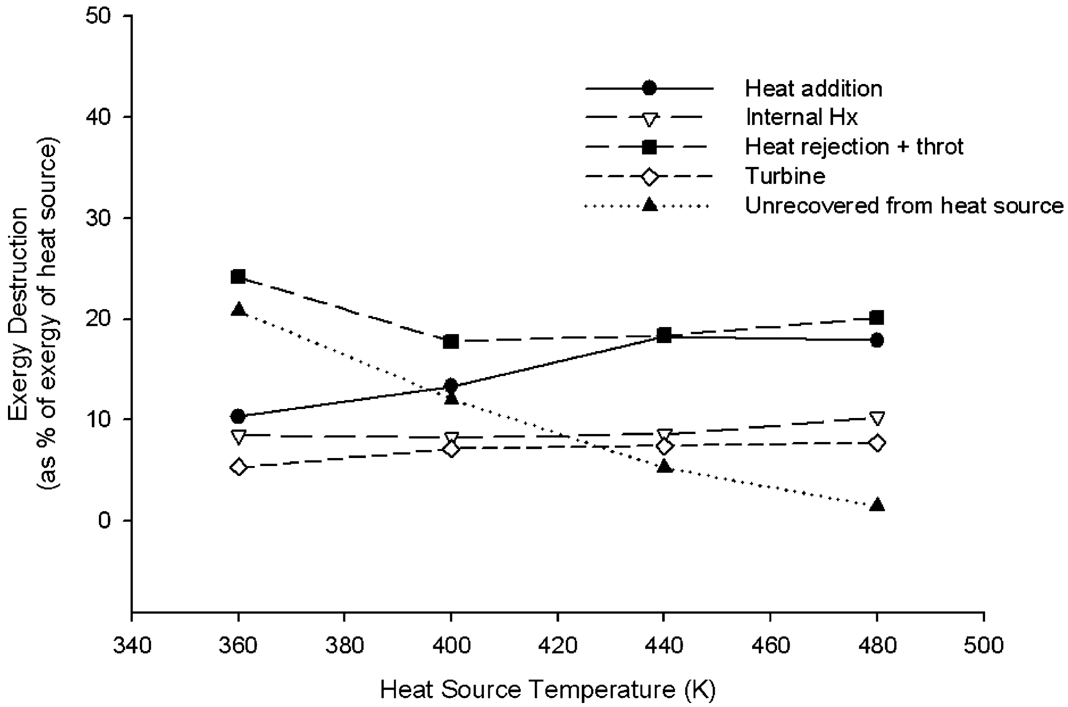

Fig. 17. Exergy destruction in cooling domain in modified cycle with a condensing mixture providing heat of distillation.

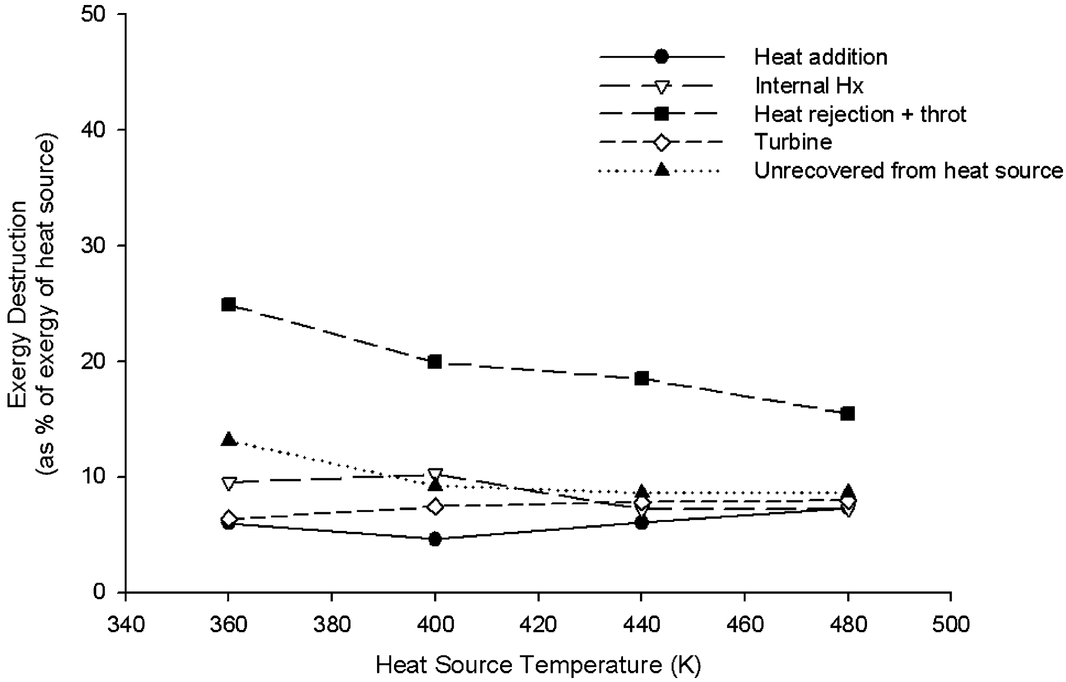

Fig. 18. Exergy destruction in work domain with a condensing mixture providing heat of distillation.

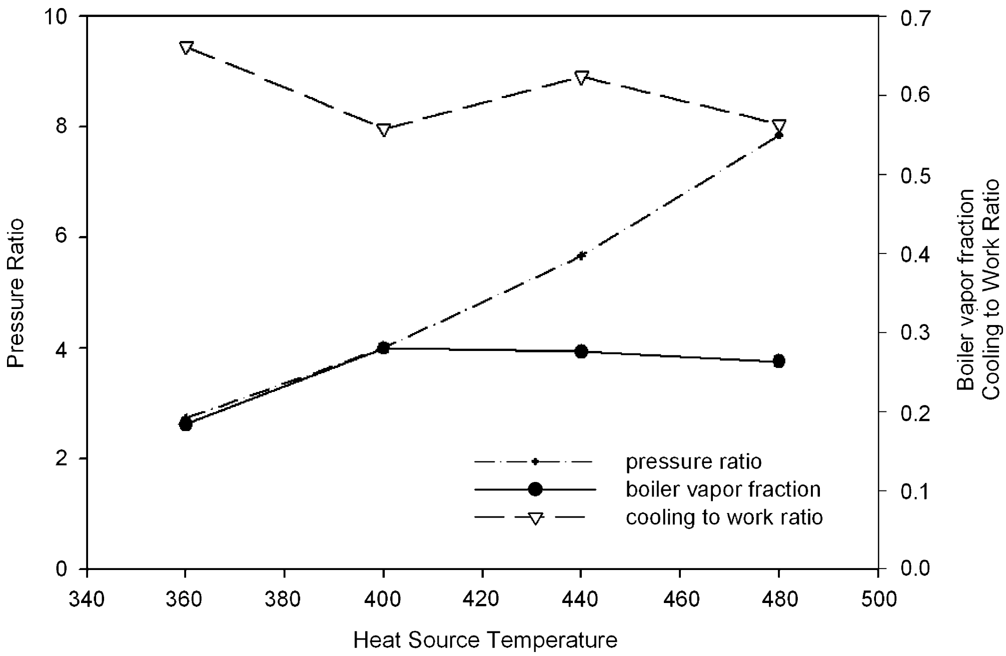

Fig. 19. Pressure ratio, boiler vapor fraction and cooling to work ratio at optimized RUE conditions (Eq. (1)) in the cooling domain for the modified cycle with a condensing mixture providing heat of distillation.

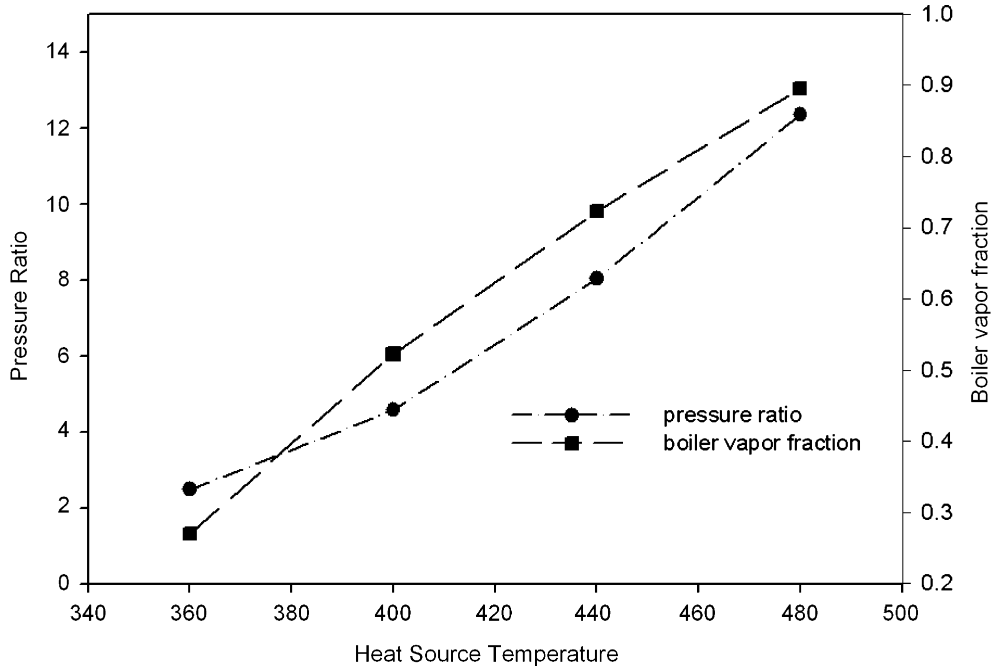

Fig. 20. Pressure ratio and boiler vapor fraction at optimized RUE conditions (Eq. (1)) in the work domain for the modified cycle with a condensing mixture providing heat of distillation.

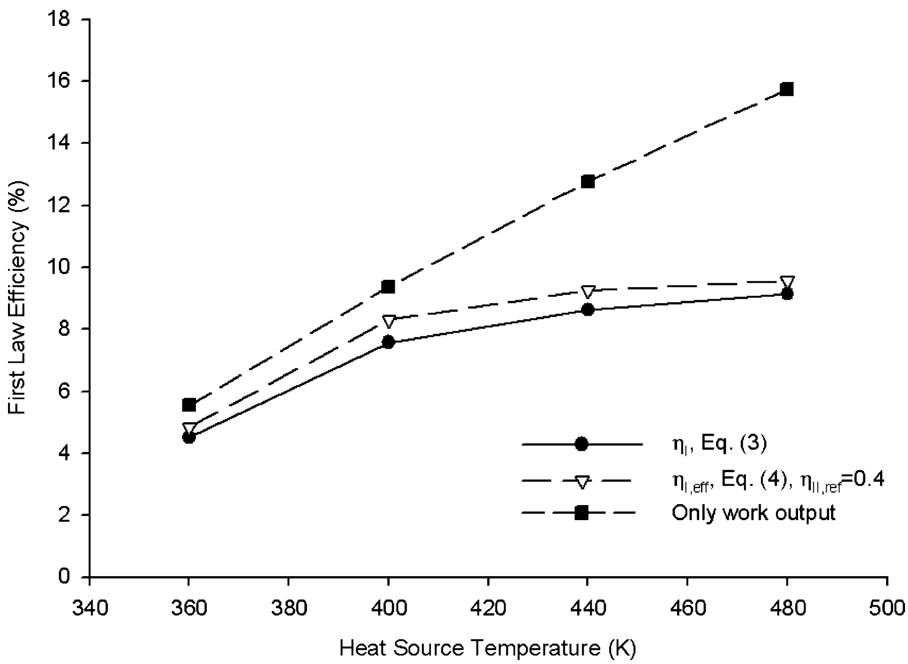

Fig. 21. First law efficiency at maximum RUE conditions for configuration using heat of condensation to produce vapor in the distiller.

6. Conclusions

Two configurations implementing a distillation method to decouple absorber and turbine exit pressures in a combined power and cooling cycle have been studied. In one case, the heat source fluid is used for the distillation, while in the other case, a condensing mixture of weak solution and vapor is used. Based on the simulation and optimization results, both methods improved the resource utilization efficiency over the optimized basic cycle configuration. The configuration where the heat source is used for distillation has slightly higher RUE at lower heat source temperatures. The first law efficiency is also consistently higher at optimized RUE conditions for this cycle. Higher first law efficiency normally means that components used for heat addition and rejection could be smaller for a cycle with the same output. From that point of view, this configuration is a better choice.

As expected, higher-pressure ratios are seen in the turbine and also higher vapor fractions in the boiler. When optimized for maximum RUE considering only the work output, both cycles tend to have very high vapor fractions in the boiler, especially at higher heat source temperatures. The RUE values are better in all cases when the output from the cycle is only work. At higher heat source temperatures and when the output from the cycle is only work, the modified configurations discussed in this paper roughly become Kalina cycles.

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