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FIRST AND SECOND LAW ANALYSIS OF A NEW POWER AND REFRIGERATION THERMODYNAMIC CYCLE USING A SOLAR HEAT SOURCE

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Abstract—The first and second laws of thermodynamics were used to analyze a novel thermodynamic cycle proposed by Goswami in 1995 that uses an ammonia–water binary mixture as the working fluid, while producing both power and refrigeration simultaneously. The thermodynamic performance of the cycle was optimized for maximum second law efficiency using a commercially available optimization program. A maximum second law efficiency of 65.8% was obtained at a heat source temperature of 420 K. An exergy analysis was performed to study losses in different components of the cycle. It is seen that the largest contribution to cycle irreversibility comes from the absorber, with the rectifier and solution heat exchanger also contributing significantly. Irreversibility generation in the boiler is high at very low heat source temperatures, but drops at higher temperatures.

1. INTRODUCTION

A substantial improvement in the performance of power cycles can be achieved by reducing the mismatch in temperature between the working fluid and the heat source fluid during the heat addition process. One approach to reduce this mismatch and irreversibility is by using a multi-component working fluid (Kalina, 1984; Alefeld, 1989; Goswami, 1998). Multi-component working fluids have variable boiling temperatures, thus providing significantly lower irreversibility in the boiler when used with a sensible heat source fluid. An improvement of 20% in the second law efficiency over a steam power cycle was reported in the literature, using the ammonia–water based Kalina cycle (Kalina and Leibowitz, 1987).

A schematic of the cycle considered in this study is shown in Fig. 1. This cycle was first proposed by Goswami (1995, 1998). A mixture of ammonia and water, referred to as the basic solution is pumped from the absorber to a high pressure (state 2). The high pressure liquid is split into two streams, that after recovering waste heat, mix and enter the boiling heat exchanger (state 3). The mixture is partially boiled in the boiler, producing a vapor rich in ammonia and a hot, liquid weak solution. The vapor is partially condensed in the rectifier to increase the ammonia content of the vapor. The rectified vapor is superheated and expanded through a turbine. The exit temperature from the turbine is lower than ambient, allowing refrigeration (cooling) to be extracted from it in the refrigeration heat exchanger. The hot weak solution is cooled in a heat recovery heat exchanger and throttled back into the absorber. The vapor is condensed by absorption into the weak solution in the absorber, accompanied by rejection of heat.

Detailed discussions on the performance of this new cycle and an optimization method used to improve cycle output are discussed in recent publications (Tamm *et al.*, 2002; Lu and Goswami, 2002a,2002b). Simulation results presented in these earlier studies 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. The definition of efficiencies for a combined power

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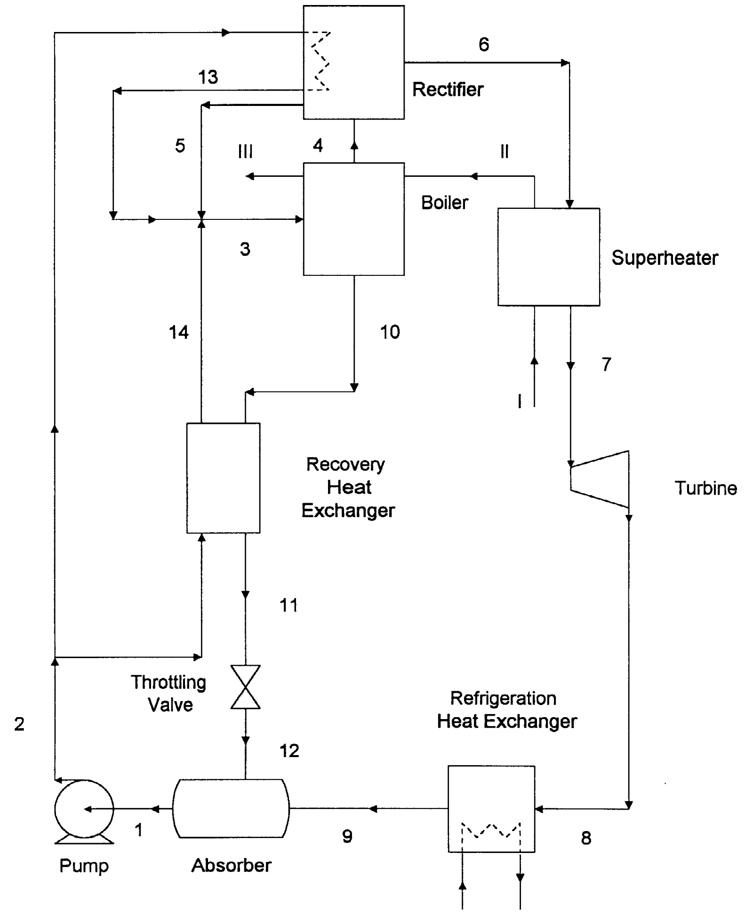


Fig. 1. A schematic drawing of the combined power and refrigeration cycle.

and cooling cycle such as that being considered in this paper is complicated by the two different simultaneous outputs (power and cooling). The optimum conditions of operation are influenced significantly by the choice of an efficiency definition (Vijayaraghavan and Goswami, 2002).

The objective of this paper is to analyze the power and refrigeration cycle using the first and second laws of thermodynamics. A second law efficiency is defined relative to a reversible cycle and is maximized in order to find the optimum operating conditions of the cycle. A simulation program for the cycle was integrated with a commercially available optimization program, GRG2, to establish the optimum operating conditions of the cycle (Lu and Goswami, 2002a). The cycle performance is investigated over a heat source temperature range of 330 K to 470 K. This temperature range covers values that can be obtained from low and mid temperature solar collectors.

2. SENSIBLE HEAT SOURCE

2.1. *Efficiency of Lorenz power cycle*

For sensible heat sources, such as in a solar thermal system or waste heat from hot exhaust gases etc., the Lorenz cycle is the ideal reversible cycle that gives the best performance (Alefeld, 1989; Lee and Kim, 1992; Ibrahim and Klein, 1996). A T–S diagram of the Lorenz cycle is shown in Fig. 2. The efficiency of a Lorenz power cycle can be written as

$$\eta_{\text{Lorenz}} = 1 - \frac{Q_{34}}{Q_{12}} \tag{1}$$

If the heat input and rejection were written in terms of the heat source and heat rejection fluids

$$\eta_{\text{Lorenz}} = 1 - \frac{m_{\text{hr}}(h_{\text{hr,out}} - h_{\text{hr,in}})}{m_{\text{hs}}(h_{\text{hs,in}} - h_{\text{hs,out}})} \tag{2}$$

Knowing that processes 4-1 and 2-3 are isen-

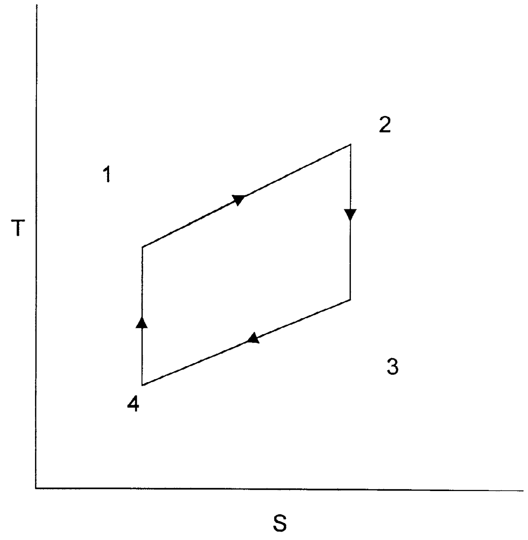


Fig. 2. The T–S diagram for a Lorenz cycle.

tropic, it is easily shown that in terms of specific entropies of the heat source and heat rejection fluids that

$$\frac{m_{\mathrm{hs}}}{m_{\mathrm{hr}}}=\frac{(s_{\mathrm{hr,out}}-s_{\mathrm{hr,in}})}{(s_{\mathrm{hs,in}}-s_{\mathrm{hs,out}})} \tag{3}$$

The efficiency expression for Lorenz cycle then reduces to

$$\eta_{\mathrm{Lorenz}}=1-\frac{(h_{\mathrm{hr,out}}-h_{\mathrm{hr,in}})/(s_{\mathrm{hr,out}}-s_{\mathrm{hr,in}})}{(h_{\mathrm{hs,in}}-h_{\mathrm{hs,out}})/(s_{\mathrm{hs,in}}-s_{\mathrm{hs,out}})} \tag{4}$$

This can be also written as

$$\eta_{\mathrm{Lorenz}}=1-\frac{(\bar{T}_s)_{\mathrm{hr}}}{(\bar{T}_s)_{\mathrm{hs}}} \tag{5}$$

The temperatures in the expression above are entropic average temperatures, of the form

$$\bar{T}_s=\frac{(h_2-h_1)}{(s_2-s_1)} \tag{6}$$

For constant specific heat fluids, the entropic average temperature can also be written as

$$\bar{T}_s=\frac{(T_2-T_1)}{\ln(T_2/T_1)} \tag{7}$$

The Lorenz efficiency can therefore be written in terms of temperatures as

$$\eta_{\mathrm{Lorenz}}=1-\frac{(T_{\mathrm{hr,out}}-T_{\mathrm{hr,in}})/\ln(T_{\mathrm{hr,out}}/T_{\mathrm{hr,in}})}{(T_{\mathrm{hs,in}}-T_{\mathrm{hs,out}})/\ln(T_{\mathrm{hs,in}}/T_{\mathrm{hs,out}})} \tag{8}$$

It is easily seen that if the heat transfer processes were isothermal, like in the Carnot cycle, the entropic average temperatures reduce to the temperatures of the heat reservoir and the heat sink, yielding the Carnot efficiency. The COP of a Lorenz refrigerator similarly, can be shown to be

$$\mathrm{COP}_{\mathrm{Lorenz}}=\frac{(\bar{T}_s)_{\mathrm{cf}}}{(\bar{T}_s)_{\mathrm{hr}}-(\bar{T}_s)_{\mathrm{cf}}} \tag{9}$$

3. SECOND LAW EFFICIENCY

The first law efficiency of thermodynamic cycles measures the fraction of the input energy that is converted to useful energy output. In a second law analysis the actual cycle performance is compared with an ideal reversible cycle, accurately indicating the improvement possible in the performance of the cycle.

The second law efficiency, η_{II}, is defined as the ratio of the actual first law efficiency η_{I} to the maximum first law efficiency,$(\eta_{\mathrm{I}})_{\mathrm{rev}}$, that is obtained from a reversible cycle operating within the same thermal boundary conditions. (Cengel and Boles, 1998; Moran, 1982).

$$\eta_{\mathrm{II}}=\frac{\eta_{\mathrm{I}}}{(\eta_{\mathrm{I}})_{\mathrm{rev}}} \tag{10}$$

In the literature, another definition is often used which gives the second law efficiency as the ratio of the exergy output to the exergy input (Szargut *et al.*, 1988; Bejan *et al.*, 1996). Exergy input could be the exergy of heat source or the exergy change in the heat source as given in Eq. (11). If the heating fluid is discarded to the environment after transferring energy to the working fluid, then its unused exergy is wasted and the efficiency is based on the exergy of the heat source. If the heating fluid is recycled or used again such as in the case of a re-circulating solar system, where the fluid is re-circulated through the collector, the efficiency should be based on the change of exergy of the heat source.

$$\eta_{\mathrm{exergy}}=\frac{E_{\mathrm{output}}}{\Delta E_{\mathrm{input}}} \tag{11}$$

where E_{output} is the output exergy and $\Delta E_{\mathrm{input}}$ is the change in exergy of the heat source.

In this paper, the term second law efficiency will be used for efficiency as given by Eq. (10), and exergy efficiency will be used for efficiency expressed in terms of exergy as given by Eq. (11).

Exergy of a system (or control volume) is defined as the maximum amount of work that can be obtained from the system when it is brought from its initial state to one in equilibrium with the environment. Exergy therefore is defined in relation to the environment or the dead state and a value of exergy can be assigned once the environment state is assigned. The exergy efficiency is affected by the choice of the dead state or reference conditions, while the second law efficiency, as defined in this paper, is free from any assumptions concerning the environment state. Different reference conditions are found in the literature for exergy analysis. For example, reference temperatures of 280, 290, 293, 298 and 300 K have been used by different authors. For work presented in this paper, a reference temperature (environmental temperature or dead state) of 290 K has been assumed.

For the combined power and refrigeration cycle, the first law efficiency is calculated as,

$$\eta_{\mathrm{I}} = \frac{W_{\mathrm{net}} + E_{\mathrm{c}}}{Q_{\mathrm{h}}} \tag{12}$$

where W_{net} is the net work output of the cycle, E_{c} is the exergy associated with the refrigeration output and Q_{h} is the heat addition to the cycle. This definition is different from that used in earlier work published on this cycle (Goswami and Xu, 1999; Lu and Goswami, 2002a), see Eq. (13).

$$\eta_{\mathrm{I}} = \frac{W_{\mathrm{net}} + Q_{\mathrm{c}}}{Q_{\mathrm{h}}} \tag{13}$$

The definition in Eq. (13) does not account for the quality of cooling obtained from the cycle, and it has been found that the values obtained from using this energy efficiency can be misleading.

Irreversibilities in the cycle are calculated from entropy generation using Eq. (14),

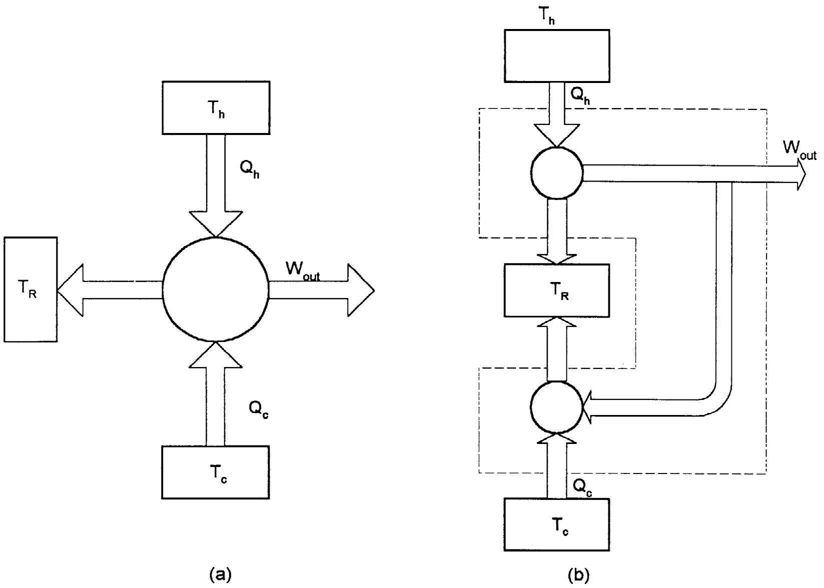
$$I = T_{\mathrm{o}} S_{\mathrm{gen}} \tag{14}$$

where I is the irreversibility, S_{gen} is the entropy generation, and T_{o} is the ambient temperature, which was also assumed to be 290 K.

3.1. Cascaded cycle analogy

An analogy to the combined cycle being considered in this paper is a cascaded power and refrigeration cycle, where part of the work output is directed into a refrigeration machine to obtain cooling. If the heat engine and refrigeration machine were to be treated together as a black box, the input to the entire system is heat, while output consists of work and refrigeration. This represents the new combined power/refrigeration cycle. Fig. 3 shows the analogy, with a dotted line around the components in the cascaded cycle representing a black box.

An ideal reversible combined cycle, then, would be two Lorenz cycle engines cascaded together (Fig. 2b). Assume that the combined cycle and the cascaded arrangement both have the same thermal boundary conditions. This assumption implies that the heat source fluid, chilled fluid and heat rejection fluid have identical inlet and exit temperatures in both cases. Further, assume



(a) (b)

Fig. 3. Thermodynamic representation of (a) combined power/cooling cycle and (b) cascaded cycle.

that the combined cycle and the cascaded arrangement have the same ratio of work to cooling output. The first law efficiency of the cascaded system, using a weight factor f for refrigeration is

$$\eta_{\mathrm{I,sys}} = \frac{W_{\mathrm{out}} - W_{\mathrm{c}} + fQ_{\mathrm{c}}}{Q_{\mathrm{h}}} \tag{15}$$

The weight factor, f is a function of the thermal boundary conditions. Therefore, the first law efficiency of the combined cycle can also be written as

$$\eta_{\mathrm{I}} = \frac{W_{\mathrm{net}} + fQ_{\mathrm{c}}}{Q_{\mathrm{h}}} \tag{16}$$

The work and heat quantities in the cascaded cycle can also be related using the efficiencies of the cascaded device

$$W_{\mathrm{out}} = Q_{\mathrm{h}}\eta_{\mathrm{HE}} \tag{17}$$

$$W_{\mathrm{c}} = Q_{\mathrm{c}}/\mathrm{COP} \tag{18}$$

By specifying identical refrigeration to work ratios (r) in combined cycle and the corresponding reversible cascaded cycle as

$$r = Q_{\mathrm{c}}/(W_{\mathrm{out}} - W_{\mathrm{c}}) \tag{19}$$

and using Eqs. (15) and (17)–(19), one can arrive at the efficiency of the cascaded system as

$$\eta_{\mathrm{I,sys}} = \eta_{\mathrm{HE}}\left[1 + \frac{r(f - 1/\mathrm{COP})}{1 + r/\mathrm{COP}}\right] \tag{20}$$

assuming the cascaded cycle to be reversible, the efficiency expression reduces to

$$\eta_{\mathrm{I,rev}} = \eta_{\mathrm{Lorenz}}\left[1 + \frac{r(f - 1/\mathrm{COP}_{\mathrm{Lorenz}})}{1 + r/\mathrm{COP}_{\mathrm{Lorenz}}}\right] \tag{21}$$

A second law efficiency would then be written as in Eq. (10)

$$\eta_{\mathrm{II}} = (\eta_{\mathrm{I}})/(\eta_{\mathrm{I}})_{\mathrm{rev}} \tag{22}$$

If the cycle and its equivalent reversible cascaded cycle have identical heat input (Q_{h}), the second law efficiency can also be written as

$$\eta_{\mathrm{II}} = \frac{\eta_{\mathrm{I}}}{\eta_{\mathrm{I,rev}}} = \frac{W_{\mathrm{net}} + fQ_{\mathrm{c}}}{W_{\mathrm{net,rev}} + fQ_{\mathrm{c,rev}}} \tag{23}$$

This reduces further to

$$\eta_{\mathrm{II}} = \frac{W_{\mathrm{net}}}{W_{\mathrm{net,rev}}}\frac{(1 + fr)}{(1 + fr)} = \frac{W_{\mathrm{net}}}{W_{\mathrm{net,rev}}} \tag{24}$$

Evidently, the refrigeration weight factor (f) does not affect the value of the second law efficiency. Assuming a value of unity for f simplifies the second law efficiency expression even further. The corresponding reversible cycle efficiency would be,

$$\eta_{\mathrm{I,rev}} = \eta_{\mathrm{Lorenz}}\left[\frac{1 + r}{1 + r/\mathrm{COP}_{\mathrm{Lorenz}}}\right] \tag{25}$$

The resulting second law efficiency equation is a good choice for second law analysis. The expression does not have the drawback of trying to weight the refrigeration with respect to the work output.

4. OPTIMIZATION PROCEDURE

A simulation program written for this cycle was combined with a commercially available optimization code, GRG2, that uses the generalized reduced gradient (GRG) algorithm to optimize cycle performance. The optimization procedure is discussed in more detail by Lu and Goswami (2002a). The variables that were used in the optimization are listed below.

- Boiler exit temperature (states 4 and 10)
- Rectifier exit temperature (state 5 and 6)
- Superheater exit temperature (state 7)
- System high pressure
- Heat source inlet temperature (fixed for each optimization)
- Heat source flow rate
- Effectiveness of recovery heat exchanger
- Flow split ratio at 2 to states 13 and 14

A fixed pressure of 2 bars (0.2 MPa) was set as the system low pressure. The system low pressure influences the turbine size and the size of the absorber condenser. While fixing the low pressure will not yield a global optimum, the results will be more meaningful.

The simulation program neglects pressure drops and heat losses, and treats the turbine and pump as idealized devices. The optimization was performed with some constraints specified to keep the operation practical. Approach temperature limits of 5 K are set for all heat transfer devices, while a pinch point temperature limit of 3 K is set in the boiler, rectifier and recovery heat exchanger. While the working fluid may not begin to boil in the recovery heat exchanger or the rectifier, the pinch point limit is applied if boiling occurs. The simulation was also constrained to produce positive refrigeration output. It was also specified that the turbine exit temperature had to be 280 K or lower with a minimum turbine exit vapor mass fraction (dryness) of 90%, for refrigeration to be obtained. An ambient temperature of 290 K is assumed. In this paper, the ambient temperature and the reference (or dead state) temperature are assumed to have the same value.

5. RESULTS AND DISCUSSION

5.1. Optimum operating conditions

The combined power and refrigeration cycle was mathematically optimized using the second law efficiency given in Eq. (22) as the objective function. The effect of heat source temperatures in the range of 330 to 470 K was studied in 10 K intervals. Both the heat source and chilled fluid were simulated using the properties of liquid water. For a heat source inlet temperature of 400 K, operating conditions obtained from optimizing the cycle for second law efficiency are shown in Table 1. States in the table correspond to locations given in the schematic of the cycle in Fig. 1. The ammonia mass fraction in the vapor leaving the boiler is very high, hence there is only a small load on the rectifier. In addition, there is no superheating under optimized conditions. Ammonia vapor expands through turbine from 2.05 to 0.20 MPa, (pressure ratio of 10.2) to a temperature of 280 K, which is lower than the assumed ambient. Transferring heat from a fluid or space to the working fluid produces refrigeration. The ammonia mass fraction in the basic solution leaving the absorber is 0.437. A large fraction of the basic solution returns from the boiler to the absorber through the solution heat exchanger. Only about 15% of the basic solution goes through the turbine and refrigeration heat exchanger, as vapor, producing power and refrigeration.

Table 2 shows the performance of the cycle at the optimum operating conditions of Table 1. Network output as a percentage of heat input, $W_{\mathrm{net}}/Q_{\mathrm{h}}$, is 16.9% and the corresponding refrigeration, $Q_{\mathrm{c}}/Q_{\mathrm{h}}$ is 1.26%. The combined power and refrigeration cycle has a first law efficiency of 16.9%. Using the old definition of first law efficiency (Eq. (13)), the (energy) efficiency of the cycle is 18.20%. As is evident, the contribution of refrigeration to the efficiency is very small. This arises out of the definition of the efficiency in which a very low weight is given to refrigeration. The first law efficiency defined in Eq. (12) uses the exergy of cooling to evaluate the contribution of the cooling component. Doing so equates the cooling output to the minimum work required to generate that amount of cooling, which will be by using a reversible refrigeration machine. This issue is discussed in greater detail by Vijayaraghavan and Goswami (2002).

From Table 2, it can also been seen that the heat transfer load in the recovery heat exchanger is very large. The large fraction of hot, weak liquid solution returning to the absorber from the boiler results in considerable energy having to be recovered. The small vapor fraction observed is due to the high ammonia concentration required for achieving low temperatures at the turbine exit.

Table 2. Performance of the optimized cycle for 400 K heat source temperature

Parameter	Value
Boiler heat input, kJ/kg basic solution	278.8
Superheater input, kJ/kg basic solution	0.0
Rectifier heat rejection, kJ/kg basic solution	22.5
Refrigeration capacity, kJ/kg basic solution	3.5
Absorber heat rejection, kJ/kg basic solution	235.2
Solution heat exchange, kJ/kg basic solution	362.5
Turbine work output, kJ/kg basic solution	49.4
Pump work input, kJ/kg basic solution	2.2
Net work, kJ/kg basic solution	47.2
Total heat addition, kJ/kg basic solution	278.8
First law efficiency %	16.9
Second law efficiency %	65.4

Table 1. Operating conditions at maximum second law efficiency, heat source at 400 K

State	T (K)	Pressure (MPa)	Entropy (kJ/kg.K)	Concentration (kg ammonia/kg solution)	Flow ratio kg/kg
1	295.0	0.20	0.19	0.437	1.000
2	295.1	2.05	0.19	0.437	1.000
3	377.7	2.05	1.33	0.437	1.009
4	395.0	2.05	4.85	0.927	0.156
5	381.5	2.05	1.35	0.413	0.009
6	381.5	2.05	4.68	0.959	0.146
7	381.5	2.05	4.68	0.959	0.146
8	280.0	0.20	4.68	0.959	0.146
9	285.0	0.20	4.77	0.959	0.146
10	395.0	2.05	1.52	0.348	0.854
11	300.1	2.05	0.30	0.348	0.854
12	300.5	0.20	0.30	0.348	0.854
13	334.6	2.05	0.74	0.437	0.129
14	379.5	2.05	1.33	0.437	0.871

To improve the cycle performance, modifications have to be made to the cycle configuration to increase the vapor fraction achievable in the cycle.

The second law efficiency at the optimum operating conditions of Table 1 is 65.4%. In order to check for possible improvements in the cycle to bring it closer to the ideal Lorenz cycle, a detailed irreversibility analysis is made. Table 3 shows the irreversibilities in the cycle as computed using Eq. (14). In the simulation, it is assumed that both the pump and the turbine are isentropic. At optimum conditions for the 400 K case considered, there is no load in the superheater under optimum conditions, thus its irreversibility is zero. The highest irreversibility is in the absorber (44% of the total irreversibility). Absorption involves the processes of condensation and mixing of the ammonia vapor into a weak liquid solution accompanied by heat rejection. Hence, it is an inherently irreversible process. Heat recovery in the rectifier and solution heat exchanger accounts for about 16 and 24% of the total irreversibility, respectively.

5.2. *Effect of heat source temperature*

Optimized efficiencies as a function of heat source temperature are shown in Fig. 4. The second law efficiency initially rapidly increases at the lower range of heat source temperatures studied. The optimum second law efficiency is a maximum (65.8%) at a heat source temperature of 420 K. Above this temperature the efficiency that can be obtained gradually drops. Fig. 5 shows the normalized heat transfer rates in the cycle as a function of the heat source temperature. It is seen from the figure that the rectifier heat transfer increases significantly at higher temperatures. The net effect is to increase the irreversibility generation in the rectifier.

By definition, the second law efficiency used in this paper attributes a very low weight to the cooling output. This happens due to the use of a reversible refrigeration device in determining a second law efficiency definition. The low weight to refrigeration output results in the cooling obtained being the minimum possible within the constraints applied (see Fig. 6). Note that the optimization was performed by specifying that a non-zero refrigeration output should be obtained.

Fig. 7 shows the pressure ratios obtained for the optimum second law efficiency. The optimum pressure ratio increases at higher heat source temperatures. At a heat source temperature of

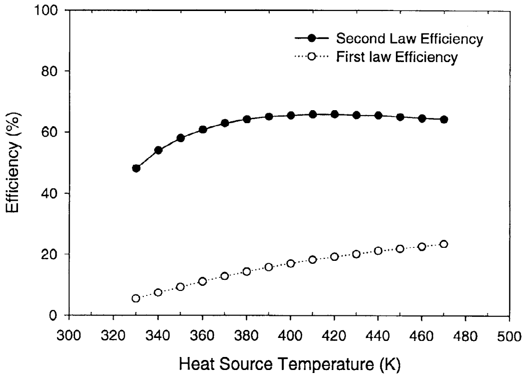


Fig. 4. Optimized first and second law efficiencies as a function of heat source temperature.

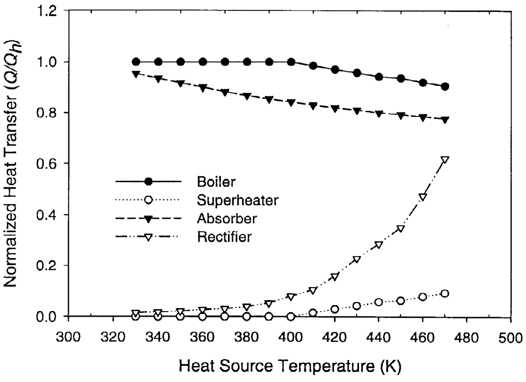


Fig. 5. Normalized heat transfer in the cycle.

Table 3. Cycle irreversibilities, heat source at 400 K, and conditions in Table 1

Component	Irreversibilities (kJ/kg basic solution)	Percent of total
Absorber	10.99	44.1
Pump	0.00	0.0
Boiler	0.69	2.8
Rectifier	4.02	16.1
Superheater	0.0	0.0
Turbine	0.0	0.0
Cooler	0.46	0.2
Heat exchanger	6.00	24.0
Reducing valve	1.74	7.0
Mixing at boiler inlet	1.44	5.8
Total	24.94	100

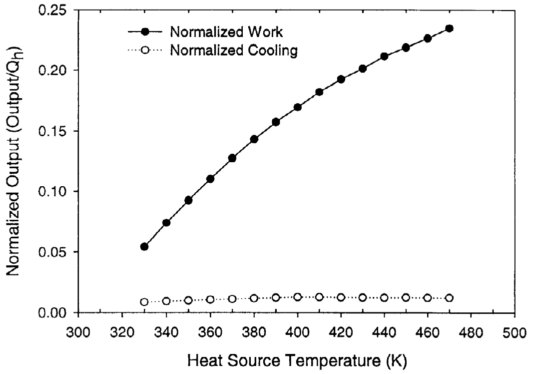


Fig. 6. Normalized (as a fraction of heat input to the cycle) work and cooling output at optimum conditions.

470 K, an optimum pressure ratio of around 27 is obtained. Such high pressure ratios may need more than a single stage expansion device.

Fig. 8 shows the major irreversibilities in the cycle grouped into three categories. Exergy losses in the boiler and superheater constitute the irreversibility of heat addition. Irreversibility generated in heat recovery in the solution heat exchanger and the rectifier along with the losses due to mixing before the boiler are added to form the recovery and mixing losses. The absorption condensation process is the remaining major irreversibility generator. It is seen that while absorber losses dominate at lower heat source temperatures, the total of the recovery and mixing losses become larger at higher source temperatures. In order to improve the performance of the cycle, these losses will have to be lowered. Some strategies that can be used to lower losses could include using alternate binary working fluid combinations and changing the configuration of the cycle to reduce irreversibility generation through better thermal matching in the boiler.

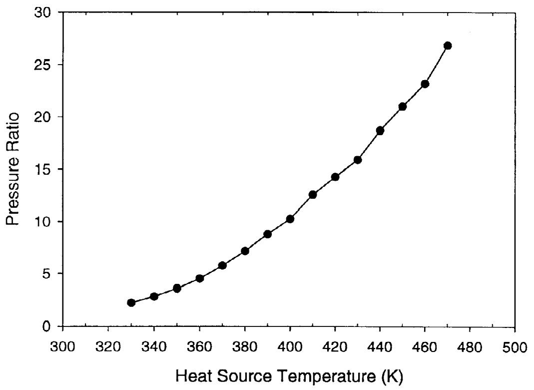


Fig. 7. Cycle pressure ratios at optimum conditions vs. heat source temperature.

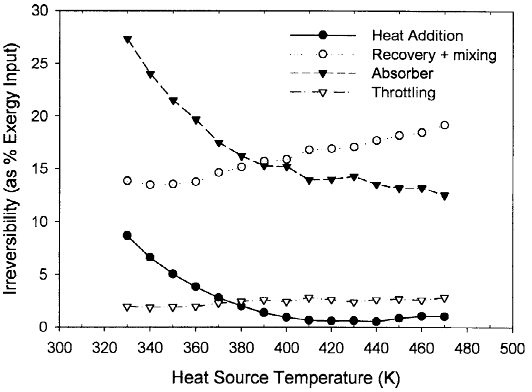


Fig. 8. A plot of major losses in various components of the cycle at optimum conditions corresponding to various heat source temperatures. Heat addition includes the boiler and superheater. Recovery and mixing losses are the sum of irreversibility in the rectifier, recovery heat exchanger and the mixing losses at the boiler entrance.

6. CONCLUSIONS

The maximum performance cycle for the investigated power and refrigeration cycle can be represented with a cascaded Lorenz cycle arrangement. A second law efficiency computed as the ratio of the first law efficiency to the maximum efficiency as obtained from the cascaded cycle arrangement can be used to optimize the cycle operating conditions. The power and refrigeration thermodynamic cycle investigated in this paper can be driven by a sensible heat source such as solar thermal energy. A solar thermal collector at 350 K (77°C) is enough to obtain a second law efficiency greater than 60%.

The combined power and refrigeration cycle can give a second law efficiency as high as 65.8% (under the idealized assumptions of this paper) at a heat source temperature of 420 K. Increasing the heat source temperature above 420 K, does not increase the second law efficiency despite an increase in the first law efficiency. The optimized conditions presented in this paper are based on some idealized assumptions, and yield maximum second law efficiency. An economic or thermoeconomic optimization may give cycle parameters different from those presented here.

Irreversibilities in the cycle are mainly attributed to the absorber, recovery heat exchanger and the rectifier. To improve performance of the cycle, modifications will have to made in order to improve the vapor fraction and volatile fraction concentration in the vapor.

NOMENCLATURE

COP	coefficient of performance
E	exergy, kJ
f	dimensionless weight factor
h	specific enthalpy, kJ/kg
I	irreversibility, kJ
m	mass flow rate, kg/s
Q	heat transfer, kJ
r	dimensionless cooling to work output ratio
S	entropy, kJ/K
s	specific entropy, kJ/kg K
T	temperature, K
$\bar{T}$	average temperature, K
W	work, kJ

Subscripts

c	cold reservoir
cf	cooled fluid
exergy	exergy
gen	generated
HE	heat engine
h	hot reservoir
hr	heat rejection fluid
hs	heat source fluid
in	inlet
I	first law
II	second law
Lorenz	Lorenz cycle
net	net total
o	reference state
out	outlet
R	heat rejection reservoir
rev	reversible
sys	system

Greek

η	efficiency
Δ	change

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