ABSORPTION POWER CYCLES

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Abstract - We present a thermodynamic analysis of the Maloney and Robertson and the Kalina absorption power cycles. The maximum power for specified external conditions is identified and used as a reference to evaluate the performance of these two absorption power cycles. The evaluation focuses on power cycles operating with low temperature heat sources such as geothermal heat, solar energy or waste heat.

INTRODUCTION

Many of the recent proposals for alternative power cycles employ a non-azeotropic binary mixture in a Rankine or absorption-type power cycle. The motivation for using mixtures is that heat transfer can occur at variable temperature while at a constant pressure. The variable temperature heat-transfer processes reduce the temperature mismatch between the hot and cold streams and the cycle, thereby reducing the availability destruction in the heat exchangers.

Maloney and Robertson[1] studied the performance of an absorption cycle. Their conclusion showed that the absorption power cycle has no thermodynamic advantage over the Rankine cycle. More recently, Kalina[2] proposed an absorption power cycle using ammonia-water. In apparent contradiction to Maloney and Ropertson's conclusion, Kalina shows that his cycle has a thermal efficiency which is 30% to 60% higher than comparable steam power cycles. Kalina[3] and Kalina and Leibowitz[4,5,6,7] explained the basic advantages of what has become known as the Kalina cycle technology.

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El-Sayed and Tribus[8] compared the Rankine and Kalina cycles theoretically when both cycles are used as a bottoming cycle with the same thermal boundary conditions. They conducted a first and second law thermodynamic analysis and concluded that the Kalina cycle can have 10 to 30% higher thermal efficiency than an equivalent Rankine cycle. A recent publication by Stecco and Desideri[9] presents the results of an analytical study showing both thermodynamic and practical advantages for the Kalina cycle compared to a Rankine cycle using the exhaust of a gas turbine as energy source. Marston[10] developed a computer model of the cycle analyzed by El-Sayed and Tribus.[8] The results of this model show good agreement with published results of El-Sayed and Tribus.[8]

This paper provides a detailed evaluation of the Kalina and Maloney and Robertson absorption power cycles and a comparison of their performance with the maximum power (MP) cycle. A methodology for providing an engineering evaluation of absorption power cycles is described. The evaluation focuses on power cycles operating with low temperature heat sources such as geothermal heat, solar energy or waste heat.

A case study is considered in which the heat source for the power cycle is a fluid stream with an inlet temperature of 455 K and a thermal-capacitance rate of 10 kW/K. The sink temperature is 286 K. Heat transfer to and from the power cycle occurs through heat exchangers which are described with traditional heat-exchanger relations. In case study, the ratio of the hot-side to cold-side heat-exchanger conductances is 1.0, i.e. $UA_H = UA_L$ in all cycles. Similar results are obtained for other heat-exchanger conductance ratios. The turbines and pumps are modeled as reversible adiabatic processes.

MAXIMUM POWER (MP) CYCLE MODEL

The best cycle that will result in the upper limit of the maximum power for specified external conditions has been studied in Refs. 11-13. The purpose of this section is to identify the MP cycle for given heat-source and sink streams and heat-exchanger characteristics. The MP cycle is necessarily irreversible since zero power results from perfectly reversible cycles. The thermodynamic irreversibilities occurring in real power cycles are primarily the result of heat-transfer processes. To account for these irreversibilities, the MP cycle is modeled as an internally-reversible power cycle coupled to heat source and sink streams through conventional counterflow heat exchangers.

place Fig. 1 here

A MP cycle model is found by recognizing that an internally reversible thermodynamic cycle can be broken into a sequence of Carnot cycles (Fig. 1) having the same total heat interactions with the heat source and heat sink and the same power output as the original cycle.

As the number of cycles in sequence increases, the performance and shape of such a sequence approaches the performance and shape of the MP cycle.

When the sequential Carnot cycles are coupled to a heat source and sink with finite thermal-capacitance rates, the power from the N-Carnot cycles is given by

$$\dot{W}=\sum_{i=1}^{N}\left[\dot{C}_H\,\varepsilon_H\,(T_{H,in,i}-T_{h,i})-\dot{C}_L\,\varepsilon_L\,(T_{l,i}-T_{L,in,i})\right], \qquad (1)$$

where $T_{H,in,i}$ and $T_{L,in,i}$ are respectively the specified source and sink inlet temperatures for a Carnot cycle in the sequence, correspondingly $\varepsilon_{H,i}$ and $\varepsilon_{L,i}$ are the effectivenesses of the hot-side and cold-side heat exchangers of each cycle, and $T_{h,i}$ and $T_{l,i}$ are the high and low temperatures of a Carnot cycle in the sequence, respectively.

The shape of the MP cycle is determined by maximizing $\dot{W}$ with respect to $T_{h,i}$ and $T_{l,i}$ for i=1 to N subject to the entropy balance constraints

$$\dot{C}_H\,\varepsilon_H\,(T_{H,in,i}-T_{h,i})/T_{h,i}\;-\;\dot{C}_L\,\varepsilon_L\,(T_{l,i}-T_{L,in,i})/T_{l,i}=0\;,\; i=1,\,N. \qquad (2)$$

An analytical solution is not apparent for this optimization problem. However, the MP cycle with a finite thermal-capacitance rate heat source and heat sink can be described numerically.

ABSORPTION POWER CYCLES

The Maloney and Robertson[1] and the Kalina[2] absorption power cycles are considered in the following analysis. Ammonia-water mixtures are used as the working medium. The performance of both cycles is investigated and compared to that of the MP cycle.

The Maloney & Robertson Absorption Power Cycle - The Maloney and Robertson cycle is shown in Fig. 2. In this system, the boiler delivers a vapor rich in ammonia (state 6). The vapor is then superheated (state 7). Power is obtained by expanding the vapor to low pressure in a turbine. The turbine exhaust (state 8) is reunited with the weak solution from the distillation unit (state 11) and is used to absorb the rich vapor in ammonia to regenerate the basic solution (state 1). The basic solution is then pumped to a high pressure and then heated and partially boiled before entering the flash tank (state 5) to complete the cycle. Hot and cold fluids are used as the heat source and sink, respectively. The cycle is shown in temperature-enthalpy plane in Fig. 3. The thick solid line represents the basic solution, while the thin solid, and thick dashed lines represent the strong solution, and the weak solution, respectively. The separation and mixing processes are shown with a thin dashed line.

place Fig. 2 here

place Fig. 3 here

The Kalina Absorption Power Cycle - The Kalina cycle is shown in Fig. 4. The Kalina cycle has a slightly different system configuration than the Maloney and Robertson cycle. The strong vapor solution (state 6) from the distillation unit is mixed with a bypass basic solution (state 3). The use of the second condenser adds an extra degree of freedom to the system and allows the distillation unit to operate at a pressure lower than the boiler pressure. The liquid mixture from this condenser is pumped to the boiler pressure (state 9), then heated, boiled, and superheated before it enters a turbine (state 12). Energy is recovered from the turbine exhaust to heat and partially boil the basic solution before it is flashed. The turbine exhaust (state 14) is then reunited with the weak solution from the distillation unit (state 16). The weak solution is used to absorb the rich vapor in ammonia to regenerate the basic solution (state 1). The basic solution is pumped to an intermediate pressure and then split into two streams; about 80% passes to a flash tank after it recovers energy from the turbine exhaust, while the other 20% (state 3) bypasses the flash tank to be mixed with the strong vapor solution (state 6). The combined stream (7) is cooled and condensed in the second condenser (state 8).

In Fig. 5, the cycle is shown in temperature-enthalpy plane. The thick solid line represents the basic solution, while the thin solid line, and thick dashed line represent the strong solution, and the weak solution, respectively. The separation and mixing processes are shown with thin dashed lines.

place Fig. 4 here
place Fig. 5 here

COMPARISON OF ABSORPTION CYCLES

The thermodynamic performance of the Maloney and Robertson and Kalina cycles is compared for different heat-exchanger sizes and thermal-capacitance rates. The effect of the ratio of thermal-capacitance rates on the maximum power for the Maloney and Robertson and Kalina cycles is shown in Fig. 6 for $NTU_H = 10$. Similar behavior can be shown for other values of NTU_H. The curve labeled maximum power cycle gives the upper limit for the power output of any cycle for specified heat-exchanger sizes and external streams conditions. This upper limit was determined as described in Ref. 11. As the ratio of the thermal-capacitance rates increases, the power output increases rapidly first and then levels off for thermal-capacitance-rate ratios greater than 5. A typical power plant operates with a thermal-capacitance-rate ratios between 5 and 10. At a very high thermal-capacitance-rate ratios (e.g., $\dot{C}_L/\dot{C}_H = 20$), the Kalina cycle produces about 80% of the maximum power and the Maloney and Robertson cycle about 70% of the maximum power.

place Fig. 6 here

The effect of heat-exchanger sizes on the power output of the two cycles is shown in Fig. 7 for $\dot{C}_L/\dot{C}_H = 5$. The power output of the MP cycle and the Kalina cycle increases as the hot-side heat-exchanger size increases. The power output of the Maloney and Robertson cycle is greater than the power output of the Kalina cycle for NTU_H less than 5, but as NTU_H increases, the power output of the Kalina cycle surpasses that of the Maloney and Robertson cycle. At very large heat-exchanger conductances (e.g., $NTU_H = 15$), the Kalina cycle produces about 90% of the maximum power and the Maloney and Robertson cycle about 70% of the maximum power.

place Fig. 7 here

The Maloney and Robertson and Kalina cycles are compared with the MP cycle in a T-$\dot{S}$ plane in Figs. 8 and 9. In order to compare the heat-transfer processes of absorption power cycles to the MP cycle, the Maloney and Robertson and Kalina cycles are shown without the internal processes, i.e. the heat recovery, mixing and separation processes. In both cycles, the heat is added and rejected at variable temperature. The Maloney and Robertson cycle has a higher pinch point at the boiler and its heat-transfer processes do not match the MP cycle, particularly at the condenser (process 12-1, Fig. 8). On the other hand, the heat-transfer processes of the Kalina cycle matches the MP cycle more closely.

place Fig. 8 here
place Fig. 9 here

CONCLUSIONS

When a power cycle interacts with source and sink streams having finite thermal-capacitance-rates, the maximum power results when the heat-transfer processes occur at variable temperature paralleling the temperature change of the external streams. The variable temperature at both cooling and heating processes of the MP cycle can be achieved by keeping the pressure constant during the phase change of a non-azeotropic binary mixture. Having different mixtures in the hot-side heat exchanger and the cold-side heat exchanger, adds a degree of freedom in designing power cycles which can approach the performance of the MP cycle. The thermodynamic advantage of the Kalina cycle can be explained in terms of the variable boiling/condensing temperature. Compared to the isothermal boiling/condensing processes occurring in the Ranking cycle, the varying temperature during the heat-transfer processes reduces the thermodynamic irreversibility of heat exchange and the effect of the thermal pinch in the boiler. The thermodynamic advantage of the Kalina cycle occurs at large heat-exchanger sizes; however, at small heat-exchanger sizes the Kalina cycle produces less power output compared to the Maloney and Robertson cycle.

A simple way to evaluate alternative power cycles during preliminary power cycle design is to compare the performance of any new proposed cycle to the cycle which produces maximum for the same external conditions. The ratio of the power output of a proposed cycle to the maximum power is an important criterion for evaluation of new power cycles.

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NOMENCLATURE

$\dot{C}$	thermal capacitance rate (mass flow rate-specific heat product), kW/K
h	enthalpy, kJ/kg
NTU	number of transfer units, NTU = UA/$\dot{C}$
$\dot{Q}$	rate of heat transfer, kW
T	temperature, K
$\dot{S}$	entropy transport rate, kW/K
UA	heat exchanger conductance, kW/K

$\dot{W}$ power, kW

x ammonia mole or mass fraction

ε heat exchanger effectiveness

Subscripts

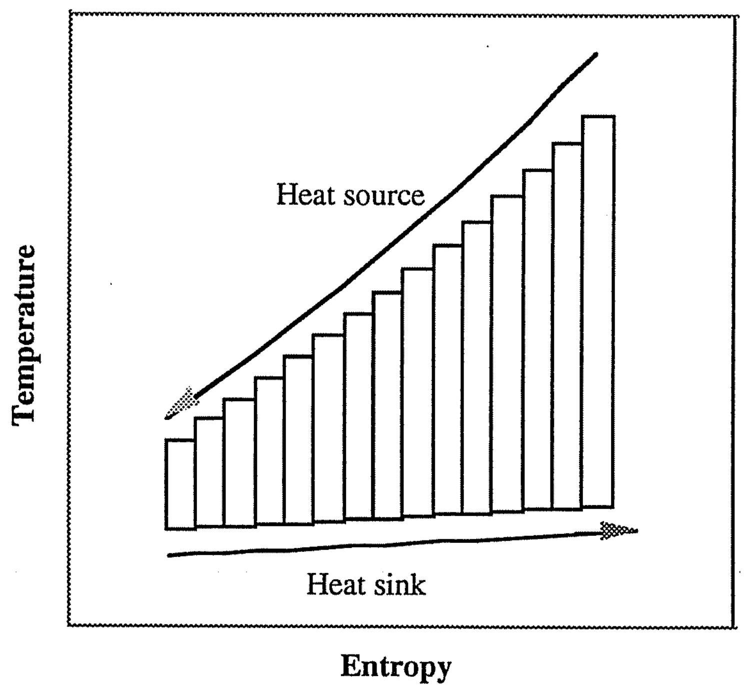
H heating fluid; heat source

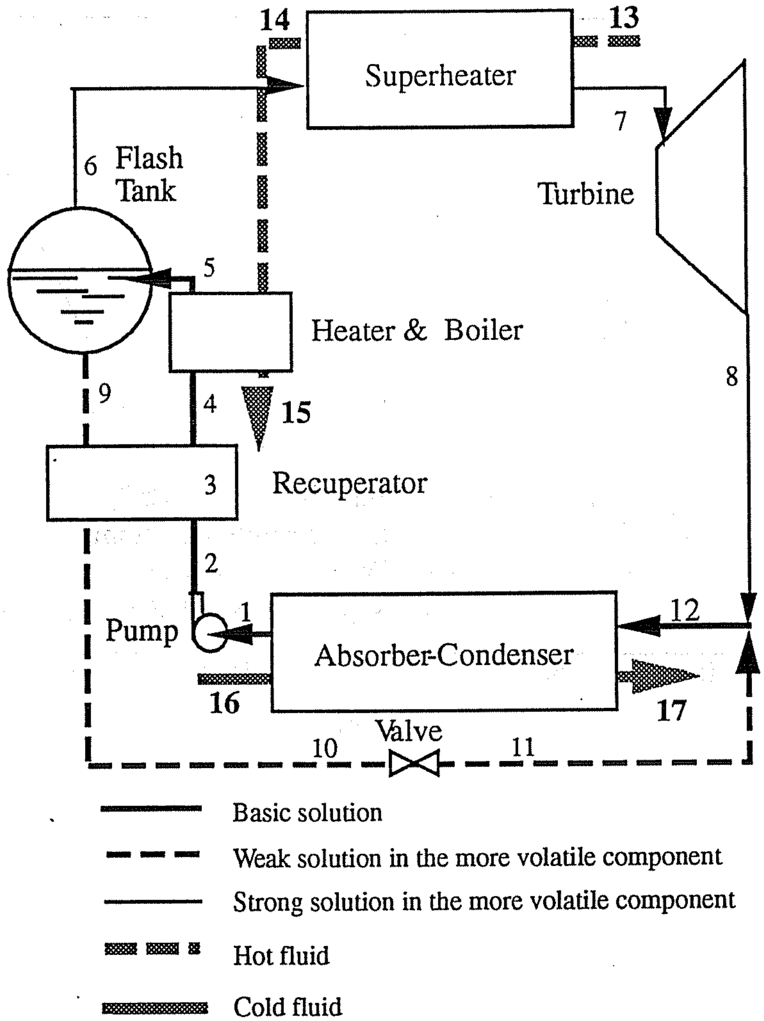
h High

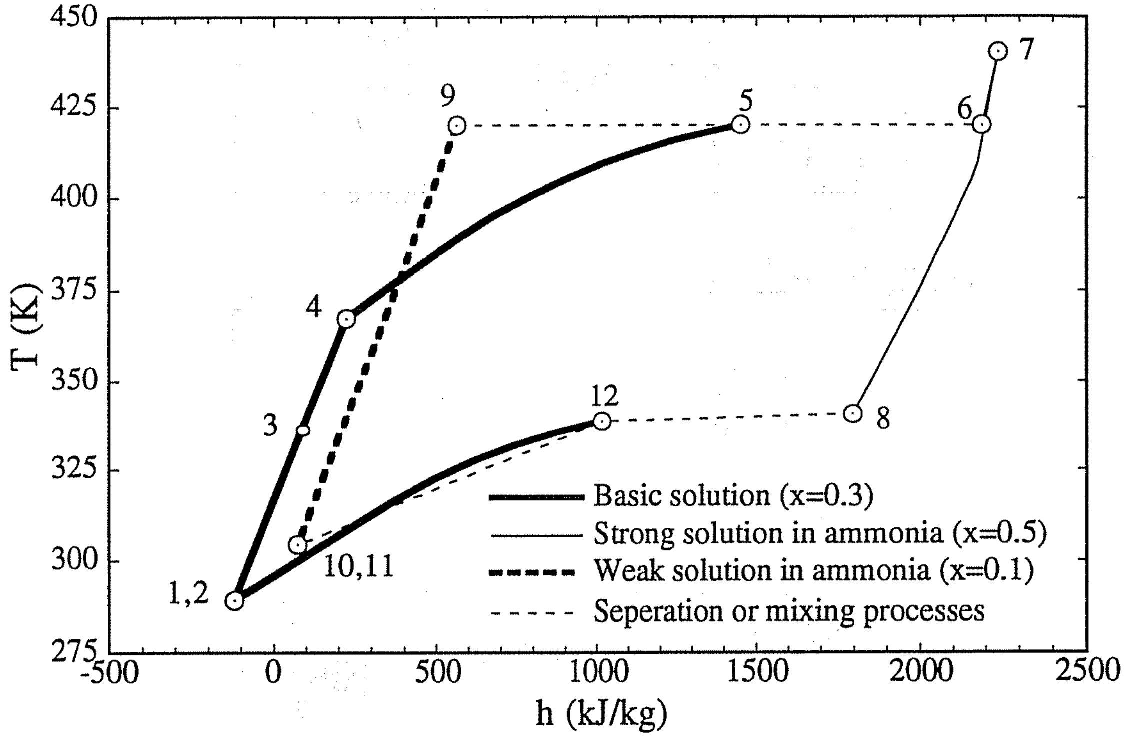
in in, inlet

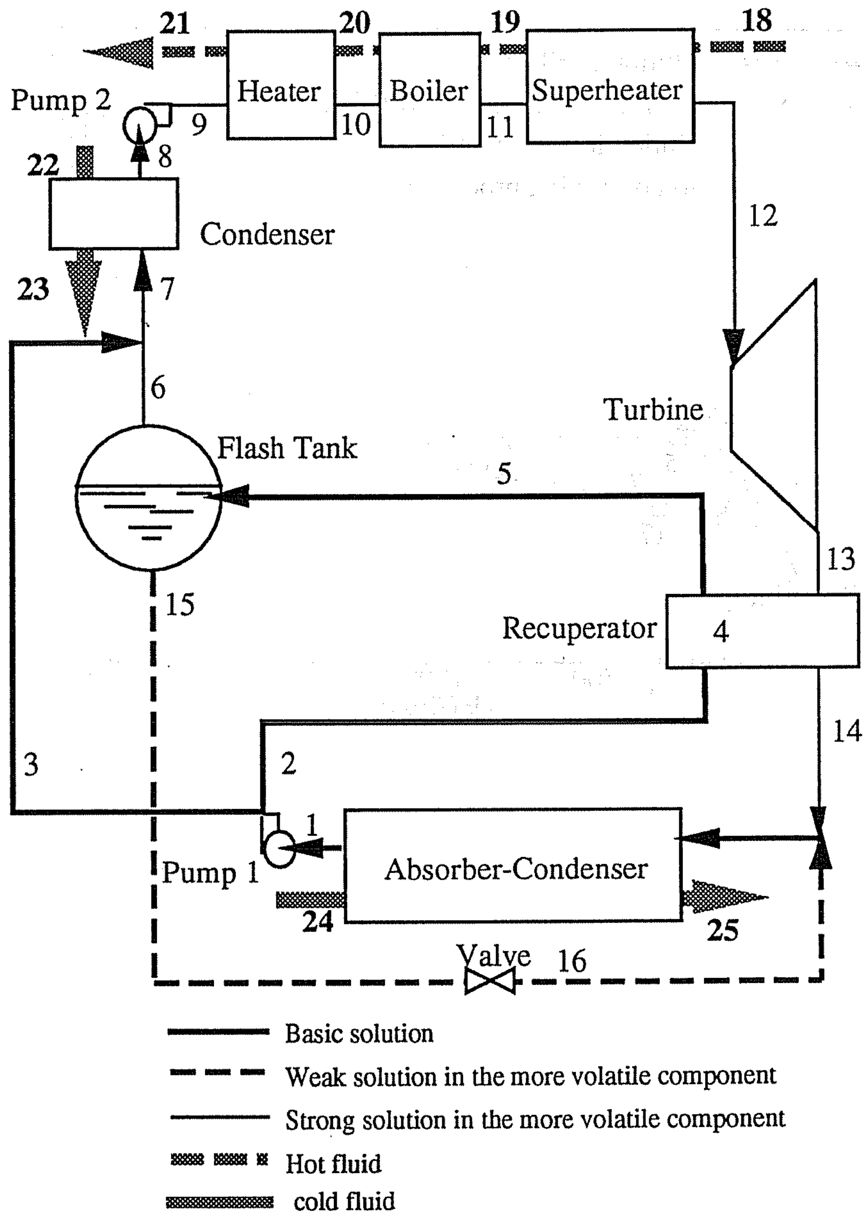
L cooling fluid; heat sink

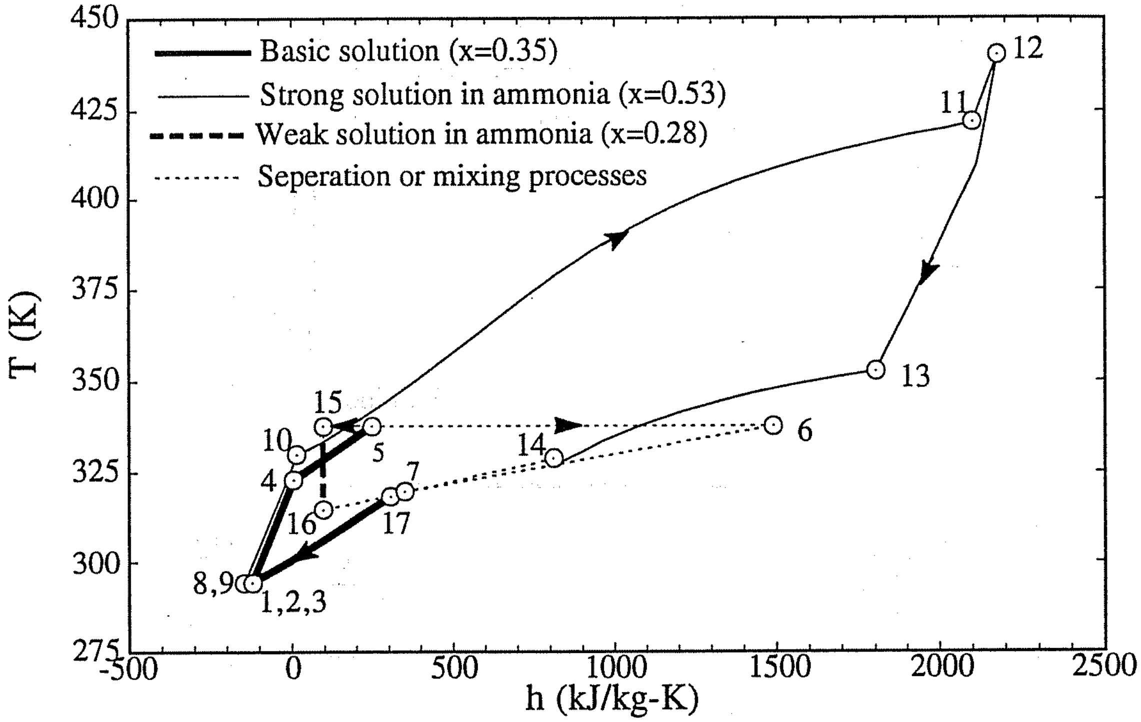
l low

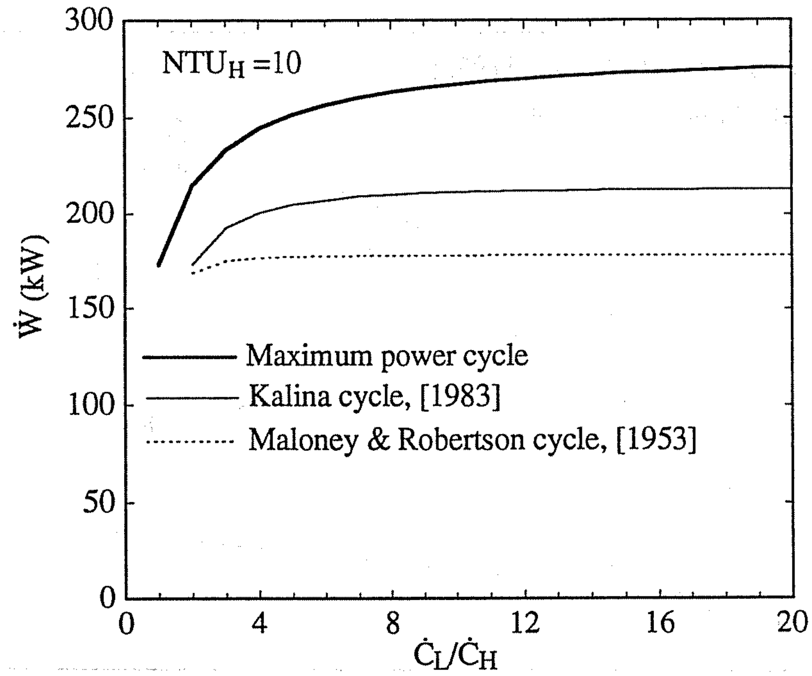

Heat source
Heat sink
Temperature
Entropy


14
13
Superheater
7
6
Flash
Tank
Turbine
5
Heater & Boiler
8
9
4
15
3
Recuperator
2
Pump
1
12
Absorber-Condenser
16
17
Valve
10
11
Basic solution
Weak solution in the more volatile component
Strong solution in the more volatile component
Hot fluid
Cold fluid

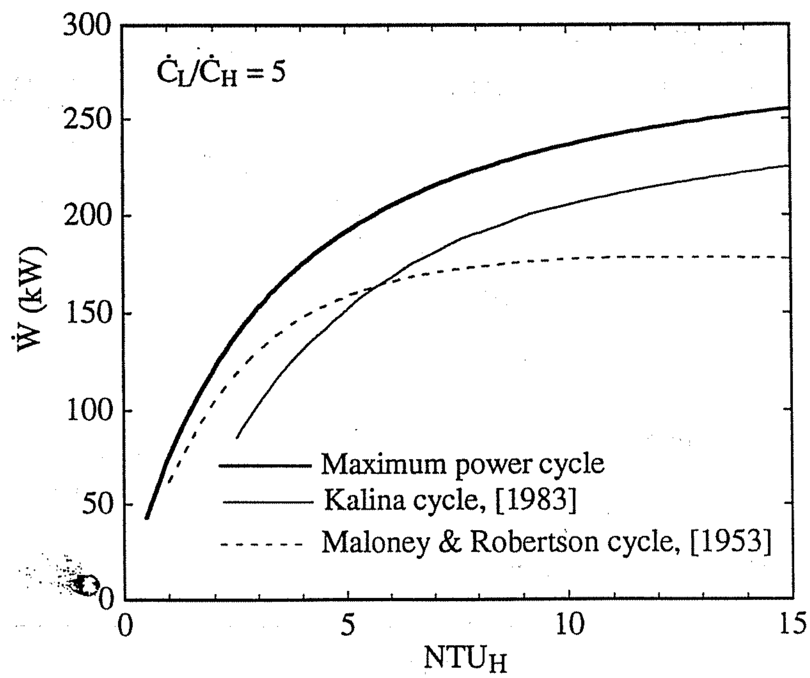

450
425
400
375
350
325
300
275
T (K)
-500
0
500
1000
1500
2000
2500
h (kJ/kg)
1,2
3
4
5
6
7
8
9
10,11
12
Basic solution (x=0.3)
Strong solution in ammonia (x=0.5)
Weak solution in ammonia (x=0.1)
Seperation or mixing processes


21
20
19
18
Pump 2
Heater
Boiler
Superheater
9
10
11
8
22
Condenser
12
23
7
6
Turbine
Flash Tank
5
13
15
Recuperator
4
14
3
2
1
Pump 1
Absorber-Condenser
24
25
Valve
16
Basic solution
Weak solution in the more volatile component
Strong solution in the more volatile component
Hot fluid
cold fluid


Basic solution (x=0.35)
Strong solution in ammonia (x=0.53)
Weak solution in ammonia (x=0.28)
Seperation or mixing processes
T (K)
h (kJ/kg-K)
450
425
400
375
350
325
300
275
-500
0
500
1000
1500
2000
2500
12
11
13
6
14
15
10
5
4
7
16
17
8,9
1,2,3



300
250
200
150
100
50
0
0
4
8
12
16
20
$NTU_H = 10$
$\dot{W}$ (kW)
$\dot{C}_L/\dot{C}_H$
Maximum power cycle
Kalina cycle, [1983]
Maloney & Robertson cycle, [1953]



300
250
200
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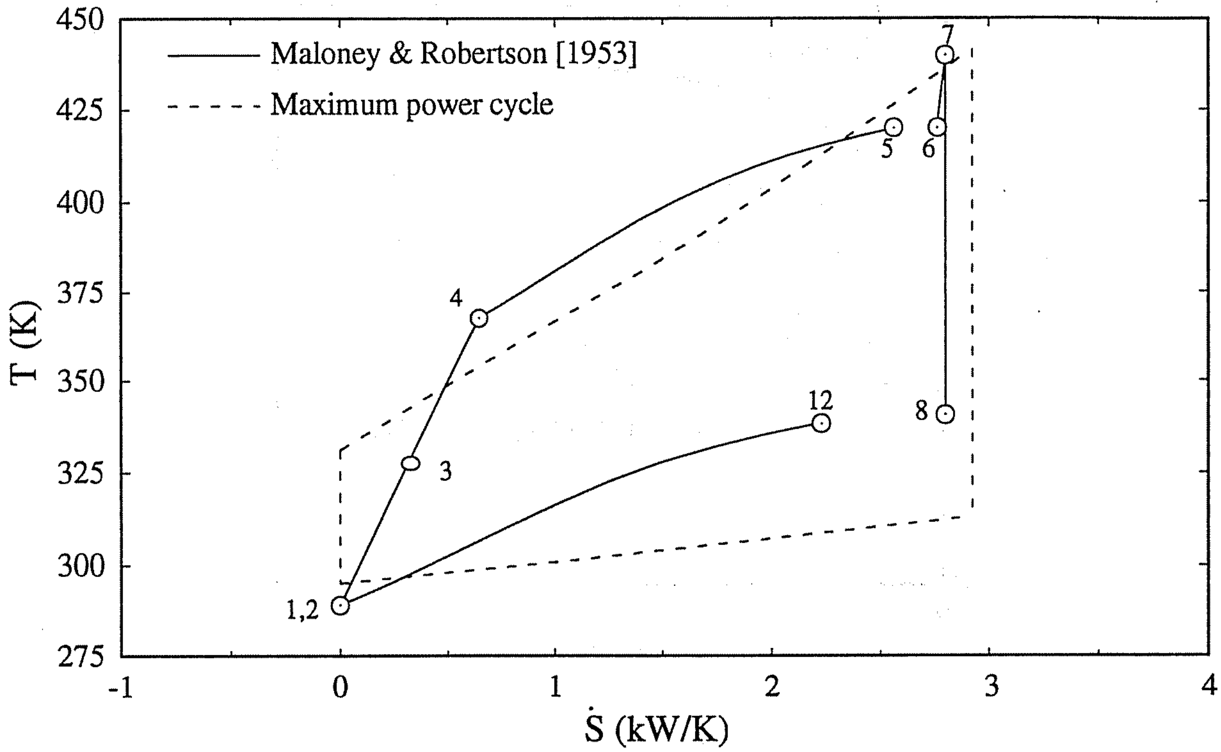
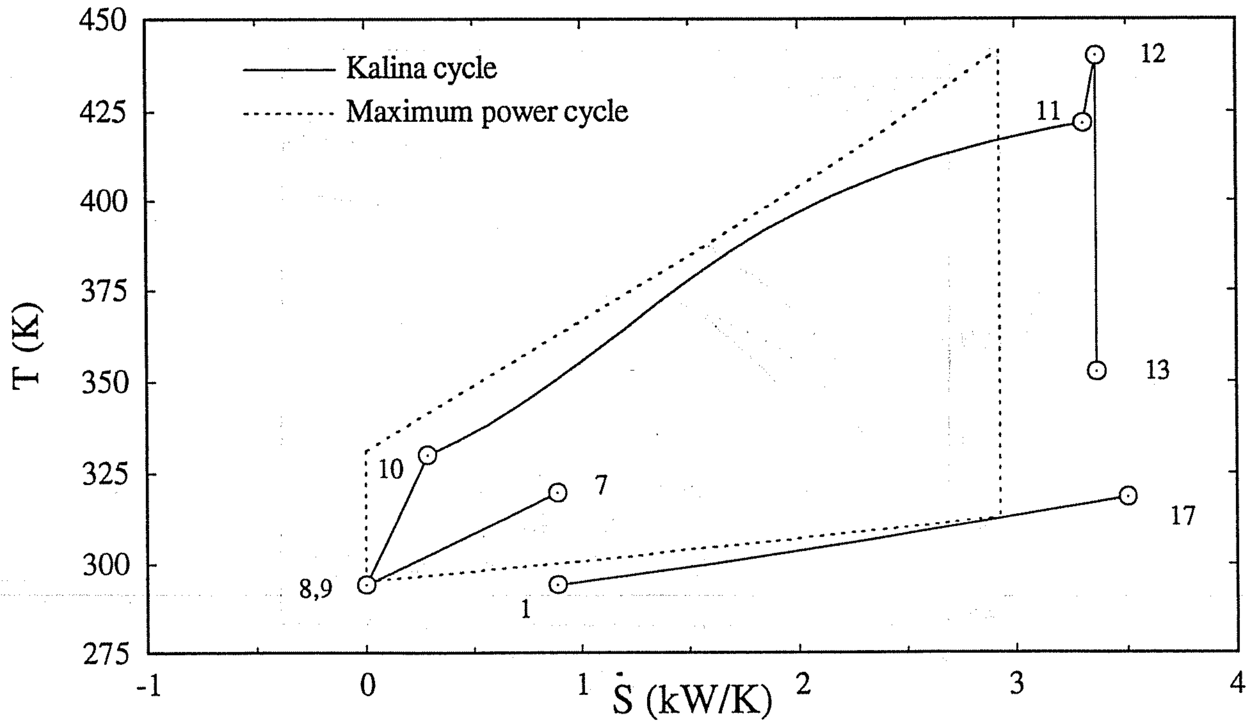
100
50
0
0
5
10
15
$\dot{C}_L/\dot{C}_H = 5$
$\dot{W}$ (kW)
NTU_H
Maximum power cycle
Kalina cycle, [1983]
Maloney & Robertson cycle, [1953]



450
425
400
375
350
325
300
275
T (K)
-1
0
1
2
3
4
Ṡ (kW/K)
Maloney & Robertson [1953]
Maximum power cycle
1,2
3
4
5
6
7
8
12



450
425
400
375
350
325
300
275
T (K)
-1
0
1
2
3
4
Ṡ (kW/K)
Kalina cycle
Maximum power cycle
1
7
8,9
10
11
12
13
17