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Exergy and Pinch Analysis of Diesel Engine Bottoming Cycles with Ammonia-Water Mixtures as Working Fluid*

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Abstract

In the present study, exergy and pinch analyses have been performed on ammonia-water and Rankine bottoming cycles for gas diesel engines. Gas diesel engines produce four waste heat streams at different temperature levels that can be used as heat sources by a bottoming cycle. The ammonia-water cycle has the potential to use these heat sources more efficiently than a Rankine cycle. The second law efficiency of the best ammonia-water cycle configuration is 43-48 % higher than the efficiency of a single-pressure Rankine cycle and 20-25 % higher than the efficiency of a dual-pressure Rankine cycle.

Key words: Kalina cycle, bottoming cycle, diesel engine, ammonia-water mixtures, exergy analysis, pinch analysis

1. Introduction

The ammonia-water cycle, as first presented by Dr. Alexander Kalina (1983), can generate more power than a steam Rankine cycle in a number of applications. The ammonia-water cycle has been show to produce more power than the Rankine cycle as a bottoming cycle to gas engines (Jonsson et al., 1999), diesel engines (Kalina, 1983), and gas turbines (Kalina, 1983; Bjorge et al., 1997; El Sayed and Tribus, 1985). Also as a gas turbine bottoming cycle for combined heat and power production (Olsson et al., 1991), as a geothermal power cycle (Lazzeri et al., 1995), and for power production from industrial waste heat (Olsson et al., 1994), the ammonia-water cycle can produce more power than a Rankine cycle. The use of a mixture as a working fluid gives some thermodynamic advantages over the conventional steam Rankine cycle which employs a pure substance, that is water, as the working fluid. When the heat source is in the form of sensible heat and has a large temperature drop, these advantages are most pronounced. The theory of the ammonia-water cycle has been validated by a demonstration plant in Canoga Park, California, USA.

In a previous study by the authors, several ammonia-water bottoming cycle configurations with gas diesel engines as prime movers were designed and compared to one-pressure and dual-pressure Rankine cycles (Jonsson and Yan, 2000). The ammonia-water bottoming cycles were further developed from three single-pressure cycle configurations found in the literature (Kalina, 1983; El-Sayed and Tribus, 1985). Gas diesel engines are fueled by natural gas and work like conventional diesel engines. The natural gas is compressed to 350 bar before it is supplied to the cylinders and the ignition of the gas is aided by a small amount of liquid pilot fuel. The waste heat from the gas diesel engines that a bottoming cycle can use is in the form of exhaust gas, charge air, jacket water, and lubricating oil. In the calculations, two gas diesel engines of different sizes were used. The size of the power plant simulated was chosen as three gas diesel engine modules together with one

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bottoming cycle module. The power output of three gas diesel engines, all of one size, was 19.5 MW and 47.2 MW, respectively. All bottoming cycles were optimized for the highest possible power output. The calculated power outputs of the ammonia-water bottoming cycle configurations ranged from 1.9 MW to 3.5 MW for the small engine, and from 4.3 MW to 7.4 MW for the big engine. The power outputs of the Rankine bottoming cycles were 1.8-2.8 MW for the small engine and 4.1-6.2 MW for the big engine. The best ammonia-water cycle configurations simulated in the previous study generated 43-47 % more power than a single-pressure Rankine cycle and 20-24 % more power than a dual-pressure Rankine cycle.

In the present study, exergy and pinch analyses have been performed on the ammonia-water and Rankine bottoming cycles previously designed, to allocate and evaluate the thermodynamic losses of different cycle components. The analyses can reveal why the ammonia-water cycles have an advantage over the Rankine cycles in the application investigated.

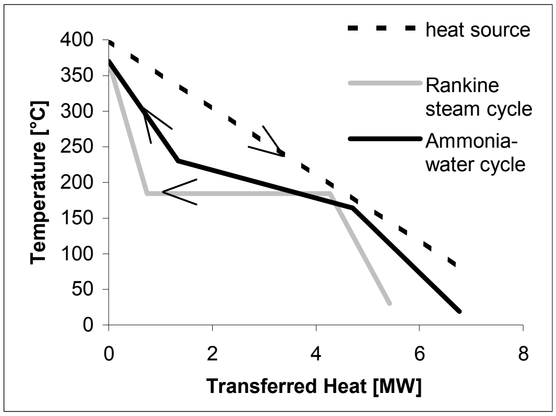
2. The Ammonia-Water Cycle

The phase change of a non-azeotropic mixture, i.e. ammonia and water, is non-isothermal. At a constant pressure, the mixture boils at increasing temperature and condenses at decreasing temperature. A pure substance like water, on the other hand, boils and condenses at a constant temperature, if the pressure is constant. This is illustrated in *Figure 1a*, which shows the temperature profiles of a bottoming cycle boiler. The exergy loss of a heat transfer process depends on the temperature difference between the hot stream and the cold stream. A large temperature difference implies a large driving force for the heat transfer process, but also a large exergy loss. The non-isothermal phase change of the ammonia-water mixture makes it possible to achieve a closer matching of the temperature profiles in the heat exchangers in the ammonia-water cycle than in the Rankine cycle.

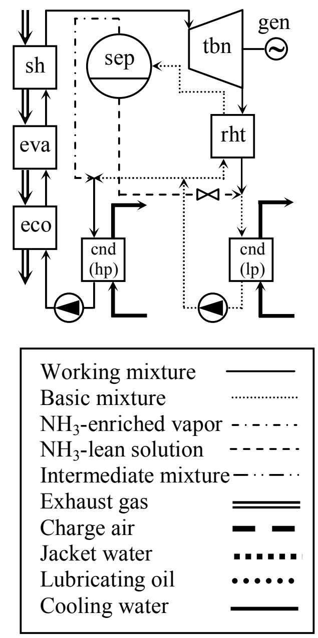
Another explanation for the high efficiency of the ammonia-water cycle compared to a Rankine cycle is that it is normally possible for the ammonia-water cycle to be highly recuperative, compared to the steam Rankine cycle. At atmospheric pressure, the boiling temperature of pure ammonia is –33°C while the boiling temperature of water is 100°C. Therefore, an ammonia-water mixture starts to boil at a lower temperature than water and the ammonia-water cycle can use heat sources of lower temperature than the Rankine cycle can use.

Figure 1b shows the simplest possible ammonia-water cycle, as presented by Kalina (1983). The ammonia concentration of the working mixture, which is used in the boiler (eco, eva, and sh) and the turbine (tbn) part of the cycle, is rather high. The stream exiting the turbine is used in the reheater (rht) to heat the stream supplied to the separator (sep). After the reheater, the working mixture stream is mixed with an ammonia-lean liquid stream from the separator, and the resulting stream is called the basic mixture. The basic mixture is condensed in the low-pressure condenser (cnd lp). The ammonia concentration of the working fluid must be decreased if normal cooling water is to be used for the condensation. When the ammonia concentration is lowered at a fixed pressure, the condensation temperature of the ammonia-water mixture is increased. The pressure of the saturated liquid is increased after the condenser and the stream is split into two streams. One of them flows through the reheater into the separator. There it is separated into one stream of ammonia-enriched vapor and one stream of ammonia-lean liquid. The ammonia-enriched vapor is mixed with the other stream from the split of the basic mixture stream, and the working mixture ammonia concentration is restored. Thereafter, this stream is condensed in the high-pressure condenser (cnd hp) and the pressure of the working fluid is increased to the maximum pressure of the cycle before it enters the boiler.

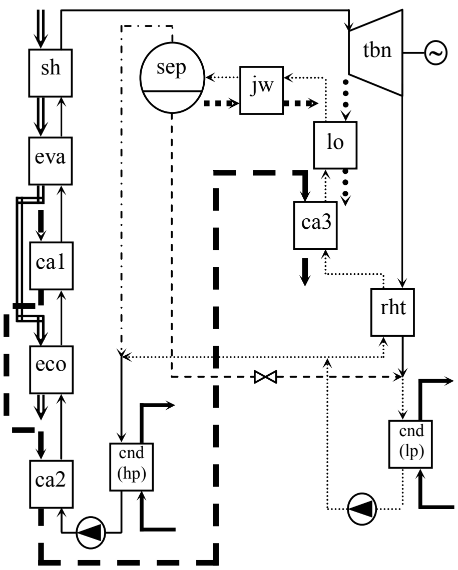
The possibility to vary the ammonia concentration of the working fluid introduces an additional degree of freedom into a single-pressure ammonia-water cycle, compared to a single-pressure steam Rankine cycle. When designing an ammonia-water cycle, the ammonia concentration of the working fluid is varied in order to achieve a close matching of the temperature profiles in all heat exchangers. If the ammonia concentration is kept constant, as in a mixture Rankine cycle with an ammonia-water mixture as the working fluid, the advantage of the ammonia-water cycle over the steam Rankine cycle would be smaller than in the case of the ammonia-water cycle type investigated in this study.



(a)



(b) cIa



(c) cIb 18V32GD

Figure 1. (a) Temperature profile of a bottoming cycle boiler; (b) Ammonia-water cycle configuration Ia and a legend for Figures 1-3; (c) Ammonia-water cycle configuration cIb for the engine model 18V32GD.

3. Calculations

The calculations on the ammonia-water and Rankine cycles were carried out in the process simulation program IPSEpro from SimTech Simulation Technology (IPSEpro). The thermodynamic properties of the ammonia-water mixtures were calculated by a library of subroutines developed by Stecco and Desideri (1989). Three gas diesel engine modules together with one bottoming cycle module are the basis for all calculations.

3.1 Assumptions and Fixed Parameters

In the calculations, two gas diesel engine models from Wärtsilä NSD Finland Oy were used. Values for the assumptions and fixed parameters, supplied by the manufacturer, used for the gas diesel engines can be found in TABLE I. The temperature of the exhaust gas in the bottoming cycle boiler is restricted due to the risk of corrosion. The main fuel is natural gas, but a small amount of a liquid pilot fuel is needed to ignite the gas. This pilot fuel can be crude oil, heavy fuel oil, or diesel oil. These oils contain sulfur, which may cause corrosion in the bottoming cycle boiler and the stack. The charge air is the combustion air, which is compressed to increase the specific power output of the gas diesel engine. The temperature of the air is increased by compression. Therefore, to increase

TABLE I. VALUES FOR THE ASSUMPTIONS AND FIXED PARAMETERS USED FOR THE GAS DIESEL ENGINES (MAUNU, 1999) AND FOR THE SIMULATIONS OF THE AMMONIA-WATER AND RANKINE BOTTOMING CYCLES. REFERENCE STATES USED FOR THE EXERGY CALCULATIONS.

Gas diesel engine	**18V32**	**18V46**	**Gas diesel engine**	**18V32GD**	**18V46GD**
$P_{engine, net}$ [MW]	19.53	47.16	$c_{p, ca}$ [kJ/kg,K]	1.01	1.01
Q_{fuel} [MW]	48.04	106.76	Q_{ca} [MW]	6.58	13.80
$\eta_{el, engine}$ [%]	40.65	44.18	m_{jw} [kg/s]	180	333
m_{eg} [kg/s]	44.43	93.30	$t_{jw, in}^{a}$ [°C]	91	91
$t_{eg, in}^{a}$ [°C]	321	330	p_{jw} [bar]	4	4
$t_{eg, minimum}$ [°C]	130	130	Q_{jw} [MW]	4.43	6.45
$c_{p, eg}$ [kJ/kg,K]	1.07	1.07	m_{lo} [kg/s]	84	225
Q_{eg} [MW]	9.08	19.97	$t_{lo, out}^{a}$ [°C]	63	63
m_{ca} [kg/s]	43.44	91.08	$c_{p, lo}$ [kJ/kg,K]	2.1	2.1
$t_{ca, in}^{a}$ [°C]	200	200	Q_{lo} [MW]	2.27	5.19
$t_{ca, out}^{a}$ [°C]	50	50			
Bottoming cycle					
η_{mech}	0.98	$\Delta p_{boiler, NH3-H20}$ [bar]	5	$\Delta t_{ca/liquid}$ [°C]	10
$\eta_{gen, 18V32GD}$	0.965	$\Delta p_{tbn-cnd\ lp, NH3-H20}$ [bar]	$0.01 \cdot p_{in}$	$t_{cooling\ water}$ [°C]	15/25
$\eta_{gen, 18V46GD}$	0.97	$p_{max, NH3-H20}$ [bar]	115	$t_{dea, R}$ [°C]	105
$\eta_{is, tbn}$	0.80	$\Delta t_{eg/vapor}$ [°C]	30	$\Delta t_{app, dea, R}$ [°C]	2
$\eta_{is, pump}$	0.80	$\Delta t_{eg/liquid}$ [°C]	15	$\Delta t_{app, drum, R}$ [°C]	0
$\Delta p_{boiler, R}$ [bar]	$0.04 \cdot p_{max}$	$\Delta t_{liquid/liquid}$ [°C]	5	$a_{tbn, min}$	0.90
Reference state for ammonia-water streams			**Reference state for all other streams**		
t_0 [°C]	15		t_0 [°C]	15	
p_0 [bar]	1.013		p_0 [bar]	1.013	
y_0 [kg NH3/kg tot]	0.99				

[a] into the bottoming cycle or out from the bottoming cycle, respectively

the power output of the engine even more, the compressed charge air is cooled before it is supplied to the cylinders. The jacket water is the engine cooling water. The lubricating oil lubricates different components in the engines, and it may coke if it is not cooled.

The ammonia-water and Rankine bottoming cycle configurations in the present study were designed in a previous study by the authors (Jonsson and Yan, 2000). The assumptions and fixed parameters in TABLE I were used. The maximum pressure in the ammonia-water cycles is 115 bar, as the database used to calculate the thermodynamic properties of the ammonia-water mixture is unreliable at pressures above this value.

The reference states used in the exergy calculations can be found in TABLE I. Only the physical, or thermomechanical, exergy has been calculated. The physical exergy is the exergy content of a stream due to the differences in pressure and temperature between the stream and the reference state. The chemical exergy, on the other hand, is the exergy content of a stream due to the difference in chemical potential between the stream and the reference state. The chemical exergy can be ignored if a closed system is studied. The chemical exergy of the streams external to the closed system, that is the exhaust gas, the charge air, the jacket water, the lubricating oil, and the cooling water, has not been calculated either. The components of these streams do not take part in any chemical reactions in the bottoming cycle and therefore their concentrations and chemical exergy contents are not changed. In the exergy and pinch calculations, it is assumed that the exhaust gas and the charge air are ideal gases. The lubricating oil is assumed to be an incompressible liquid. Therefore, the heat capacities of these streams are assumed to be dependent only on the temperature of the stream.

3.2 Equations

The thermomechanical exergy of a stream is calculated as in Eq. (1). This equation is used to calculate the thermomechanical exergy of ammonia-water and water streams.

$$E = m\left[(h - h_0) - T_0(s - s_0)\right] \qquad (1)$$

The exhaust gas and the charge air are modeled as ideal gases, and the thermomechanical exergy of these streams is calculated as in Eq. (2). When a difference in exergy for a stream is to be calculated, the effect of stream pressure may not be necessary to consider, as the last term in Eq. (2) cancels out. However, this cancellation is only valid if the pressure drop of the stream can be ignored.

$$E = m\left[c_p(T - T_0) - T_0 c_p \ln\left(\frac{T}{T_0}\right) + T_0 R \ln\left(\frac{p}{p_0}\right)\right] \quad (2)$$

The lubricating oil is modeled as an incompressible substance. The thermomechanical exergy of an incompressible substance is calculated as in Eq. (3). The second term in Eq. (3) cancels out when calculating an exergy difference for a stream, if the pressure drop of the stream is not considered. This simplification results in the same exergy equation as for an ideal gas.

$$E = m\left[c_p(T - T_0) + v(p - p_0) - T_0 c_p \ln\left(\frac{T}{T_0}\right)\right] \quad (3)$$

To calculate the exergy loss of a component, the exergy balance is used, as defined in Eq. (4). The exergy of the stream or streams entering the component and the work input to the component must equal the exergy of the stream or streams exiting the component, the work done by the component, and the exergy loss of the component. The shaft work is defined as the work output of the turbine or the work input to the pumps, without considering the mechanical efficiency. Stray heat losses, kinetic energy effects, and potential energy effects are ignored.

$$\sum_{\text{streams,in}} E + W_{\text{shaft,in}} = \sum_{\text{streams,out}} E + W_{\text{shaft,out}} + I \quad (4)$$

The first law efficiency of a bottoming cycle in this study is calculated as in Eq. (5).

$$\eta_I = \frac{\left(W_{\text{shaft}}^{\text{tbn}} - \sum_{\text{pumps}} W_{\text{shaft}}^{\text{pump}}\right)}{\left(Q_{eg} + Q_{ca} + Q_{jw} + Q_{lo}\right)} \quad (5)$$

The second law efficiency of a bottoming cycle is calculated as in Eq. (6). The shaft work is defined as in Eq. (4).

$$\eta_{II} = \frac{\left(W_{\text{shaft}}^{\text{tbn}} - \sum_{\text{pumps}} W_{\text{shaft}}^{\text{pump}}\right)}{\left(E_{eg} + E_{ca} + E_{jw} + E_{lo}\right)} \quad (6)$$

The exergetic temperature used in the pinch analyses is the Carnot factor, as defined in Eq. (7).

$$T_{ex} = \left(\frac{T - T_0}{T}\right) \quad (7)$$

TABLE II. SUMMARY OF THE CONFIGURATIONS ANALYZED IN THIS STUDY.

		Engine model	
		18V32GD	*18V46GD*
Ammonia-water cycles (c)	*Only exhaust gas (a)*	Confs.: cIa, cIIa	Confs.: cIa, cIIa
	All waste heat streams (b)	Confs.: cIb, cIIb, cIII	Confs.: cIb, cIIb, cIII
Rankine cycles (R)	*Only exhaust gas (a)*	One-pressure: R1a Dual-pressure: R2a	One-pressure: R1a Dual-pressure: R2a
	All waste heat streams (b)	One-pressure: R1b Dual-pressure: R2b	One-pressure: R1b Dual-pressure: R2b

4. Configurations

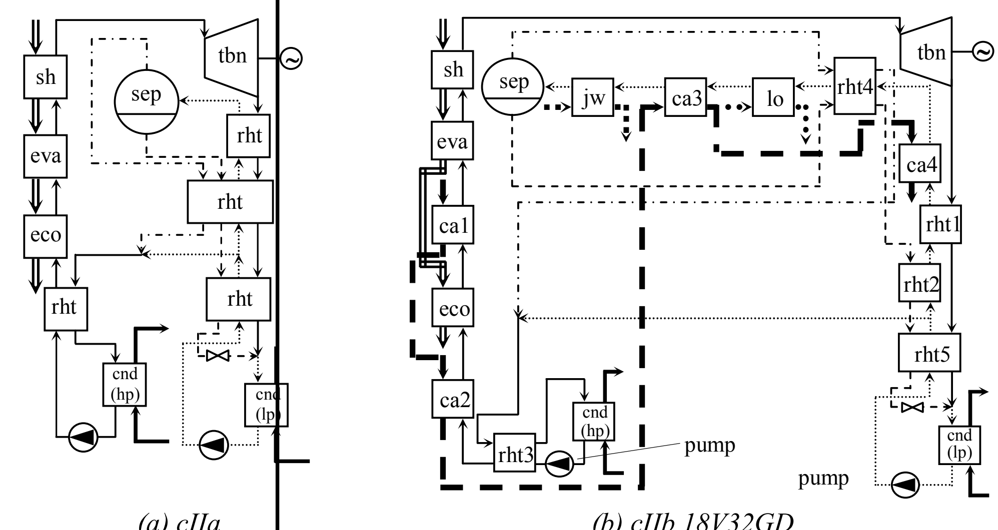
A summary of all ammonia-water and Rankine bottoming cycles investigated in this study can be found in TABLE II. The configurations called “a” use only the exhaust gas from the engines as a heat source. The configurations called “b” use, or try to use, all available heat sources produced by the engines. The available heat sources include the exhaust gas, the charge air, the jacket water, and the lubricating oil. As the specifications of the waste heat streams from the two engine models differ, the ammonia-water cycle layouts for maximal power production differ for the two engine models. The configurations presented in this paper are therefore, in many cases, not the same for the engine models 18V32GD and 18V46GD.

The simplest ammonia-water cycle configuration investigated in the current study is configuration Ia. It was first presented by Kalina (1983). In configuration Ia, the only heat source is the exhaust gas from the gas diesel engines, as shown in *Figure 1b*. The temperature of the inlet cooling water is 15 °C. Therefore, the lowest possible temperature of the condensate after a condenser is 20 °C, due to the minimal allowed temperature difference of 5 °C to the inlet cooling water. If the temperature of the condensate after the high-pressure condenser is increased to 22 °C for the engine model 18V32GD bottoming cycle and to 29 °C for the engine model 18V46GD bottoming cycle, the power output is increased compared to the case where the condensate temperature is 20 °C. The condensate temperature cannot be increased further, as this will cause the temperature difference in the reheater to fall below its minimal allowed value. The non-optimal temperature profile of the high-pressure condenser is probably more than compensated by closer matched temperature profiles in the boiler, thus resulting in a higher power output than in the case of an optimal condenser temperature profile. Configuration Ib is a modification of configuration Ia, in which all the available heat sources are used. Configuration Ib for the engine model 18V32GD can be seen in *Figure 1c*. Configuration Ib for the engine model 18V46GD is the same as for the engine model 18V32GD, except that the lubricating oil heat exchanger (lo) and the charge air heat exchanger number three (ca 3) have changed places. With a temperature of 23 °C of the high-pressure condensate for the engine model 18V32GD bottoming cycle more power is generated than if the minimal condensate temperature is used. For the engine model 18V46GD bottoming cycle, a minimal condensate temperature gives the highest power output.

Configuration IIa was first presented by El-Sayed and Tribus (1985). The only heat source is the exhaust gas, as shown in *Figure 2a*. This configuration has a higher degree of internal heat exchange compared to configuration I. Configuration IIb is a further development of configuration IIa, where heat exchangers have been inserted to make use of all the available heat sources from the engines. Configuration IIb for the engine model 18V32GD is shown in *Figure 2b* and configuration IIb for the engine model 18V46GD is shown in *Figure 3a*.

Configuration III was first presented by Kalina (1983). The two separators create a third main ammonia concentration level in the cycle, which is intermediate in ammonia concentration to the basic mixture and the working mixture. Configuration III is shown in *Figure 3c*. The configuration layout is the same for the two gas diesel engine models.

To have a basis for comparison for the ammonia-water cycles, steam Rankine cycles were also simulated in the previous study. For the Rankine cycles, the bottoming cycle layouts are always the same for both gas diesel engine models. In configuration R1a, only the exhaust gas from the engines is used as a heat source for the bottoming cycle. In configuration R1b, shown in *Figure 3b*, all available heat sources are used, if possible. The layout of configuration R1a is the same as for configuration R1b, except that there are no charge air heat exchangers and there is an exhaust gas feed preheater before the deaerator instead of the charge air heat exchanger named ca 2. A dual-pressure Rankine cycle has the same number of degrees of freedom as a single-pressure ammonia-water cycle. A dual-pressure Rankine cycle with only exhaust gas as a heat source, configuration R2a, has been simulated. All available heat sources have been used, if possible, in the dual-pressure configuration R2b, shown in *Figure 3d*. Configu-ration R2a is the same as configuration R2b, except that there are no charge air heat ex-changers and there is an exhaust gas economizer directly after the first boiler feed pump instead of the charge air heat exchanger number three (ca 3).



(a) cIIa (b) cIIb 18V32GD

Figure 2. (a) Ammonia-water cycle configuration IIa; and (b) Ammonia-water cycle configuration IIb for the engine model 18V32GD. A legend can be found in Figure 1b.

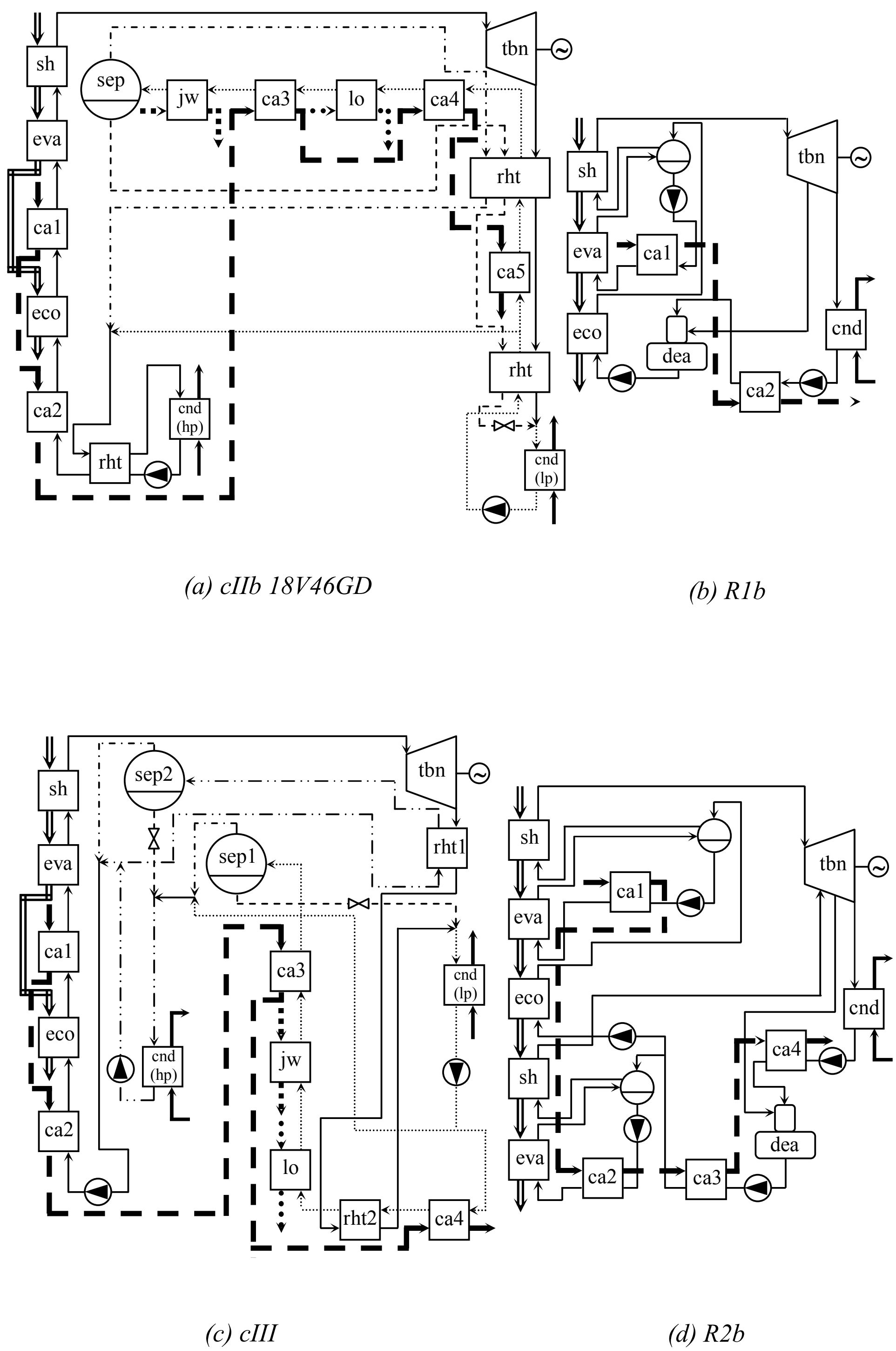


Figure 3. (a) Ammonia-water cycle configuration IIb for the engine model 18V46GD (b); Rankine cycle configuration R1b (c); Ammonia-water cycle configuration III and (d); Rankine cycle configuration R2b. A legend can be found in Figure 1b.

5. Results and Discussion

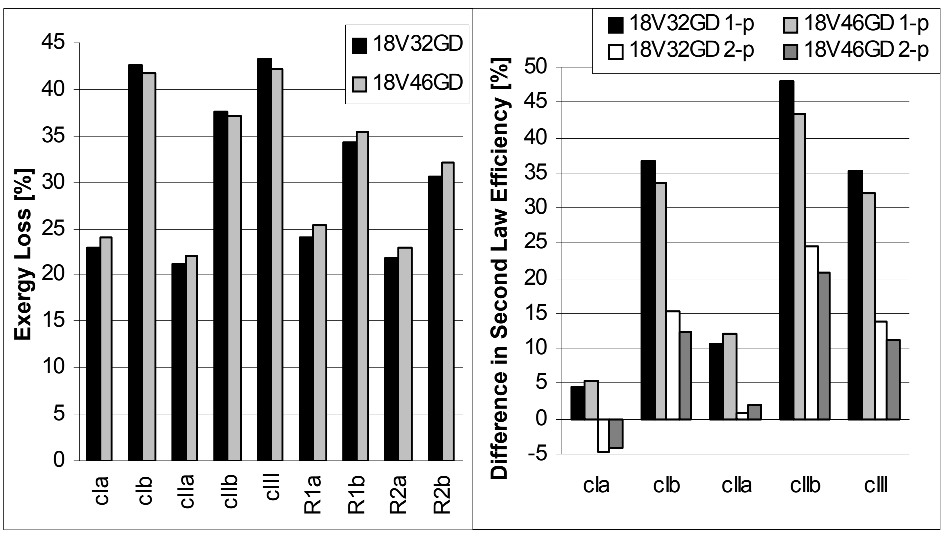
5.1 Combined exergy and pinch analysis

Exergy analyses have been performed on all bottoming cycles, to evaluate the exergy losses of different components in the cycles. In order to optimize the heat exchanger networks in the different configurations, pinch analyses have been performed on some of the cycle configurations. To construct the temperature profiles of a bottoming cycle, only the heat transfers in the heat exchangers were considered. The heat transferred in each temperature range was calculated and composite temperature profiles were constructed for the hot and cold streams, respectively. The temperature can also be expressed as an exergetic temperature, as defined in Eq. (7). The area below the exergetic temperature profile corresponds to the exergy of the stream. The area between the two composite curves represents the exergy losses of the heat transfer process.

The available energy and exergy of the waste heat streams from the two gas diesel engine models are shown in TABLE III. The useful part of the energy, the exergy, is, of course, smaller for the heat sources of low temperature, that is the jacket water and the lubricating oil, than for the heat sources of high temperature.

TABLE III. HEAT AND EXERGY RATES OF THE WASTE HEAT STREAMS.

Heat	*18V32GD*		*18V46GD*	
	Q [MW]	**E [MW]**	**Q [MW]**	**E [MW]**
eg	9.08	3.77	19.97	8.38
ca	6.58	1.76	13.80	3.69
jw	4.43	0.89	6.45	1.31
lo	2.27	0.36	5.19	0.81
sum	22.35	6.78	45.40	14.19



(a) *(b)*

Figure 4. (a) Total component exergy losses as a percentage of the total exergy available to the bottoming cycle; (b) Second law efficiencies for the ammonia-water cycle configurations compared to the Rankine cycles. Ammonia-water cycle configurations called "a" have been compared to Rankine cycles called "a". The same relation applies for the configurations called "b". When compared to a one-pressure Rankine cycle, the ammonia-water cycle is labeled "1-p". If compared to a dual-pressure Rankine cycle, the label is "2-p".

TABLE IV. FIRST LAW AND SECOND LAW EFFICIENCIES FOR ALL CONFIGURATIONS

	18V32GD		**18V46GD**			**18V32GD**		**18V46GD**	
Conf.	η_I [%]	η_{II} [%]	η_I [%]	η_{II} [%]	**Conf.**	η_I [%]	η_{II} [%]	η_I [%]	η_{II} [%]
cIa	9.28	30.60	10.28	32.90	**R1a**	8.88	29.28	9.76	31.23
cIb	15.62	51.50	16.44	52.61	**R1b**	11.42	37.66	12.32	39.43
cIIa	9.82	32.36	10.93	34.96	**R2a**	9.73	32.09	10.71	34.27
cIIb	16.90	55.71	17.67	56.55	**R2b**	13.56	44.70	14.62	46.79
cIII	15.44	50.88	16.26	52.03					

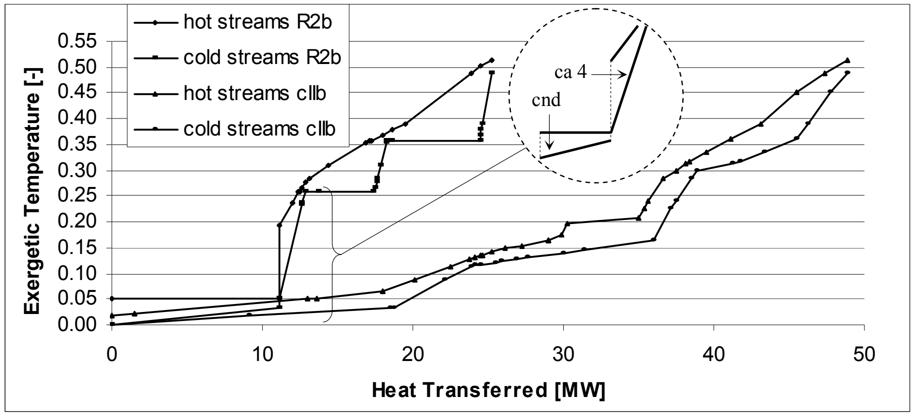


Figure 5. Exergetic temperature profiles for the configurations IIb and R2b for the engine model 18V32GD

The total exergy losses of all components of the bottoming cycles are shown in *Figure 4a*. The component exergy loss for each cycle is expressed as a percentage of the total exergy available to the bottoming cycle in the waste heat streams and by the pump work. Even though a configuration does not use all the available heat sources, the total exergy available from all heat sources has been used in the comparison to make it possible to compare all the configurations. The configurations with a high number of components and to which the most heat is supplied, i.e. configurations Ib, IIb, and III, exhibit large exergy losses. These configurations use more of the available waste heat and they can produce more power than the other configurations, as can be seen in *Figure 4b* and in TABLE IV. In TABLE IV, the first and second law efficiencies of all bottoming cycle configurations presented in this study are shown. The second law efficiencies of the ammonia-water cycles have been compared to the efficiencies of the Rankine cycles, as shown in *Figure 4b*. The results show that the ammonia-water cycles generate more power than the Rankine cycles, except configuration Ia when compared to the dual-pressure Rankine cycle configuration R2a.

The exergetic temperature profile of the ammonia-water cycle that generated the most extra power compared to a Rankine cycle, configuration IIb for the engine model 18V32GD, is shown in *Figure 5*. In the figure, the ammonia-water cycle is compared to the Rankine cycle, for the same engine model, with the highest power output of all simulated Rankine cycles, that is configuration R2b. Configuration IIb can also be compared to configuration R2b by comparing TABLE V, which presents the exergy losses for the Rankine configuration R2b, with TABLE VI, which shows the exergy losses for configuration IIb. In *Figure 5*, the hot and cold exergetic temperature profiles for configuration R2b intersect where the temperature profiles for the condenser and for the charge air heat exchanger number four (ca 4) meet. A dashed circle in *Figure 5* shows the intersection area. In the condenser, the hot stream is the steam-water mixture exiting the turbine and the cold stream is the cooling water. In the charge air heat exchanger, the hot stream is the charge air and the cold stream is the condensate from the condenser. This means that the hot stream from the condenser becomes the cold stream in the charge air heat exchanger. In general, the ammonia-water cycle has a higher potential for internal heat recovery than the Rankine cycle, and the ammonia-water cycle can use heat sources of low temperatures that the Rankine cycle cannot use. In *Figure 5*, the total energy transferred is 48.8 MW in the ammonia-water cycle and 25.2 MW in the Rankine cycle. An ammonia-water mixture starts to boil at a lower temperature than pure water, and in this study, many of the heat sources are at relatively low temperatures. The bottoming cycle configurations, both ammonia-water and Rankine, that only use exhaust gas as a heat source always use all of the available heat in the heat source, i.e. the bottoming cycles can lower the exhaust gas temperature to the minimal value allowed. The ammonia-water cycle configurations that can use all the heat sources use all the available heat in the waste heat streams, except configuration IIb that cannot decrease the charge air temperature to the minimal value of 50°C due to limitations of the cycle layout. The Rankine cycles that can use all four waste heat streams can only make use of the exhaust gas and the charge air, and the temperature of the charge air is not lowered to the minimal value. None of the Rankine cycles in this study can use the energy in the jacket water or the lubricating oil. Another feature that is to the advantage of the ammonia-water cycle is how the condensate temperature is determined in the ammonia-water cycle and in the Rankine cycle, respectively. The condensate temperature in the ammonia-water cycle is determined by the minimal allowed temperature difference of 5°C between the condensate and the inlet cooling water, that has a temperature of 15°C. In the Rankine case, the condensate temperature is determined by the minimal allowed temperature difference between the condensate and the outlet cooling water, that has a temperature of 25°C.

TABLE V. EXERGY LOSSES OF THE COMPONENTS IN RANKINE CONFIGURATION R2B, FOR THE ENGINE MODEL 18V32GD.

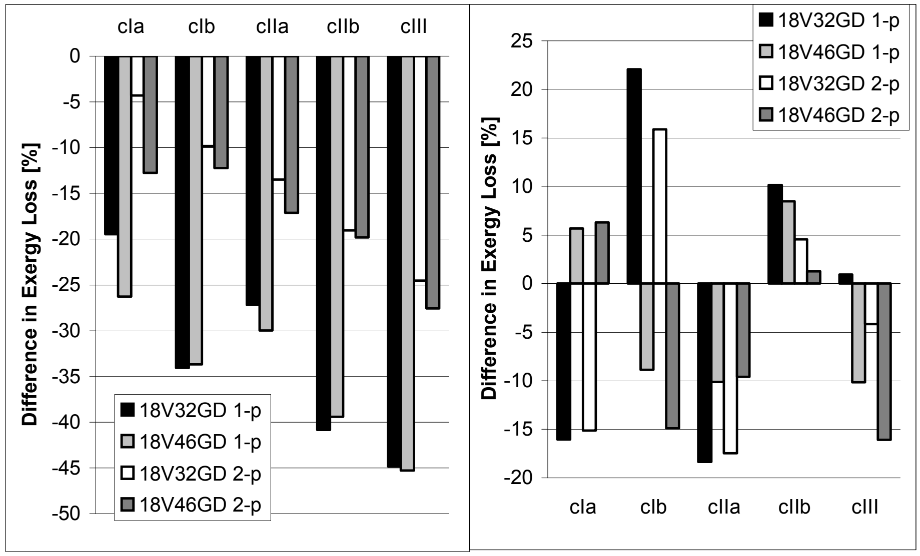
Comp.	I [kW]	I/E_{tot}[b] [%]	Comp.	I [kW]	I/E_{tot}[b] [%]
mix[a]	0.02	0.00	sh 1	8.95	0.13
splits[a]	0.00	0.00	eva 1	125.12	1.84
drums	0.05	0.00	ca 2	173.67	2.56
dea	0.09	0.00	ca 3	4.87	0.07
ca 4	128.96	1.90	cnd	362.24	5.34
sh 2	67.78	1.00	pumps	0.59	0.01
eva 2	482.19	7.11	tbn	660.50	9.73
ca 1	16.69	0.25	sum	2076.15	30.60
eco 2	44.42	0.65			
Efficiencies		η_I = 13.56 %		η_{II} = 44.70 %	

[a] These components are not shown in *Figure 3d.*
[b] E_{tot} = total available exergy

The exergetic temperature profiles in *Figure 5*, and also the TABLES V and VI, indicate that the main difference in exergy loss between the ammonia-water cycle and the Rankine cycle is in the boiler. In *Figure 6a*, the exergy losses from the boilers of the ammonia-water cycles have been compared to the exergy losses of the Rankine cycle boilers. The boiler has been defined as all heat exchangers between the boiler feed pump and the turbine. A negative value implies that the exergy loss of the boiler is smaller for the ammonia-water cycle than for the Rankine cycle. In the figure, it is shown that the exergy losses of the boiler are always smaller for the ammonia-water cycles than for the Rankine cycles. An ammonia-water mixture working fluid, with its non-isothermal boiling, generates much smaller exergy losses in the boiler than the water working fluid in the Rankine cycle. Smaller temperature differences between the hot stream and the cold stream in the boiler of the ammonia-water cycle mean smaller irreversibilities in the heat transfer process and a potential for a higher power production than in the Rankine cycle.

Regarding the exergy losses in the condensers, the picture is not that clear. The exergy losses from the condensers of the ammonia-water cycles have been compared to the condenser exergy losses of the Rankine cycles, see *Figure 6b*. A negative value means that the exergy loss of the ammonia-water cycle condenser is smaller than the exergy loss of the corresponding Rankine cycle condenser. The heat transfers in the ammonia-water cycle condensers are larger than in the Rankine cycle condensers in the cycle configurations where all heat sources are used, as more heat is transferred into the ammonia-water cycles than into the Rankine cycles. If the exhaust gas is the only heat source, then the heat transfers in the condensers are approximately the same for the ammonia-water and the Rankine cycles. A large heat transfer can imply a large exergy loss, even though the temperature profiles of the ammonia water cycle heat exchangers are often better matched than in the Rankine cycles. The difference between the boiler and the condenser heat transfer characteristics is the form of the temperature profiles of the heat source in the boiler and the heat sink in the condenser. In this study, the temperature drop of the main heat source, the exhaust gas, is large and, therefore, the ammonia-water cycle has an advantage over the Rankine cycle because of the non-isothermal boiling of the working fluid. However, the temperature increase of the heat sink, that is the cooling water, is small, thus the non-isothermal condensing of the ammonia-water mixture is of little advantage compared to the Rankine cycle.

The exergy losses of all components in two versions of configuration IIb for the gas diesel engine model 18V32GD are shown in TABLE VI. The exergy losses are expressed both as absolute values and as a percentage of the total exergy available from all heat sources and from the pump work supplied. The exergy losses for the optimized version of configuration IIb are compared to a version of the configuration where a reheater before the separator (rht 2) has been removed. The ammonia concentration of the basic composition in this version of configuration IIb has been increased to avoid too small temperature differences in the heat exchangers. The efficiencies and the exergy losses of the different components are almost the same for the two versions of configuration IIb, thus the removal of one small reheater had no major effect on the overall efficiency of the cycle. Also, the exergetic temperature profiles of the two versions are almost the same. This shows that a complex configuration with many heat exchangers, each with a small magnitude of the heat transfer, can be simplified without a large loss of power output. This is of importance when considering the cost of the ammonia-water bottoming cycle.



(a) boiler *(b) condenser*

Figure 6. Exergy losses in the ammonia-water cycles compared to the Rankine cycles. The comparison has been made in the same way as in Figure 1b.

TABLE VI. EXERGY LOSSES OF THE COMPONENTS IN CONFIGURATION IIB, FOR THE ENGINE MODEL 18V32GD, WITH AND WITHOUT RHT 2.

	cIIb with rht 2		*cIIb without rht 2*	
Comp.	**I [kW]**	**I/E_{tot}[a] [%]**	**I [kW]**	**I/E_{tot}[a] [%]**
mixs.	9.32	0.13	11.83	0.17
split	-0.01	-0.00	0.01	0.00
valve	15.52	0.22	13.38	0.19
sep	0.34	0.00	0.35	0.01
jw	214.72	3.08	202.08	2.90
ca 3	64.24	0.92	61.06	0.88
lo	56.00	0.80	54.03	0.78
rht 4	43.10	0.62	51.54	0.74
ca 4	9.73	0.14	12.09	0.17
rht 1	32.31	0.46	37.85	0.54
rht 2	4.65	0.07	-	-
rht 5	129.50	1.86	131.75	1.89
sh	203.36	2.92	203.36	2.92
eva	201.63	2.90	201.63	2.90
ca 1	76.05	1.09	76.05	1.09
eco	57.11	0.82	57.11	0.82
ca 2	197.10	2.83	199.69	2.87
rht 3	31.84	0.46	31.13	0.45
cnds.	388.61	5.58	416.78	5.99
pumps	35.10	0.50	34.47	0.50
tbn	847.67	12.18	841.81	12.10
sum	2617.89	37.61	2637.99	37.91
Efficiencies	η_I = 16.90 %		η_I = 16.83 %	
	η_{II} = 55.71 %		η_{II} = 55.46 %	

[a] E_{tot} = total available exergy

A simplified exergy flow diagram for the configuration IIb, for the engine model 18V32GD, is shown in *Figure 7*. In the figure, the boiler includes all heat exchangers between the boiler feed pump and the turbine, except the reheater named rht 3, which is included in the distillation-condensation subsystem (DCSS). The largest exergy losses are found in the turbine, the boiler, and the DCSS. The mechanical losses in the pumps are not shown in the diagram. The exergy loss due to mechanical losses is 3 kW for pump 1 and 0.6 kW for pump 2.

When optimizing the initial configuration III found in the literature (Kalina, 1983), it was found that the highest power output was reached when the condenser before the boiler, cnd 1, was removed. To investigate why the removal of this condenser could increase the power output of the cycle, calculations were made on a version of the configuration with the condenser inserted again. There are other differences between the two versions of configuration III apart from the extra condenser. The calculations on the cycle with the extra condenser did not converge and the temperature differences in the heat exchangers were too small if the same values of the parameters varied were used as for the cycle without the extra condenser. The exergy losses of the two versions of configuration III for the engine model 18V32GD are shown in TABLE VII, and in *Figure 8* the exergetic temperature profiles of the two versions are compared. The first and second law efficiencies of the cycle with an extra condenser are significantly lower than the efficiencies of the optimized configuration. The temperature difference between the hot composite curve and the cold composite curve are larger for the cycle with the extra condenser, indicating larger exergy losses in the heat exchangers than for the cycle without the extra condenser. Configuration III with an extra heat exchanger has not been optimized. The difference in electrical efficiency between configuration III with and without an extra heat exchanger may be smaller if both configurations are optimized with respect to the ammonia concentrations of the working and basic mixtures.

5.2 Optimization

The ammonia-water and Rankine bottoming cycles have been optimized to achieve the highest possible power outputs. The ammonia-water cycle configurations presented in this study have been further developed from cycle configurations found in the literature. These cycles had originally been designed as bottoming cycles to gas turbines and diesel engines. That meant that they were adapted for one or two heat sources. In the present study, there are four heat sources available to the bottoming cycle. By inserting or removing heat exchangers, the configurations were further developed to improve the cycle performance.

TABLE VII. EXERGY LOSSES OF THE COMPONENTS IN CONFIGURATION III WITHOUT AND WITH CND 1, FOR THE ENGINE MODEL 18V32GD.

Comp.	*cIII without cnd 1*		*cIII with cnd 1*	
	I [kW]	**I/E_{tot}[a] [%]**	**I [kW]**	**I/E_{tot}[a] [%]**
mixs.	80.06	1.15	131.16	1.89
splits	0.00	0.00	0.00	0.00
valves	37.79	0.54	2.67	0.04
seps.	0.05	0.00	0.06	0.00
rht 1	54.33	0.78	235.84	3.39
ca 3	150.95	2.16	24.55	0.35
jw	524.09	7.51	586.08	8.43
lo	199.92	2.87	259.93	3.74
rht 2	81.04	1.16	8.76	0.13
ca 4	21.49	0.31	27.25	0.39
sh	255.31	3.66	254.37	3.66
eva	182.28	2.61	182.33	2.62
ca 1	90.41	1.30	90.92	1.31
eco	46.14	0.66	46.06	0.66
ca 2	142.58	2.04	259.09	3.73
cnds.	356.85	5.12	391.77	5.63
pumps	34.68	0.50	33.40	0.48
tbn	754.81	10.82	689.10	9.91
sum	3012.78	43.19	3223.35	46.35
Efficiencies	η_I = 15.44 %		η_I = 14.48 %	
	η_{II} = 50.88 %		η_{II} = 47.73 %	

[a] E_{tot} = total available exergy

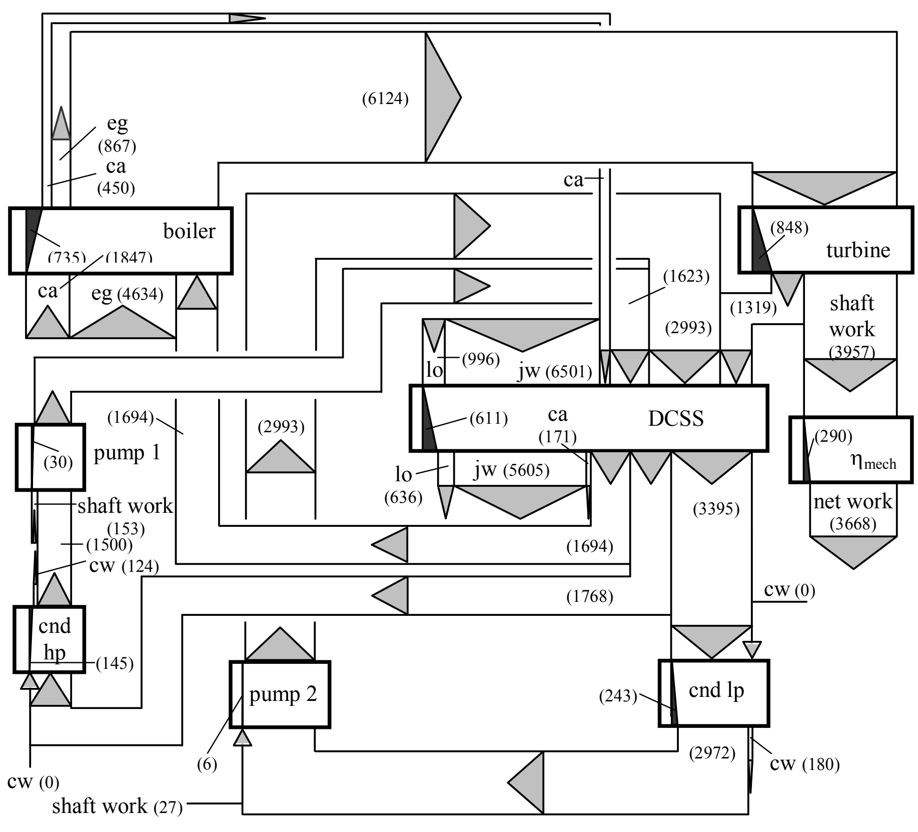


Figure 7. Exergy flow diagram for configuration IIb, engine model 18V32GD. The streams consist of ammonia and water if not otherwise stated. The numbers in brackets are the exergy flow rates, in kW, of the streams.

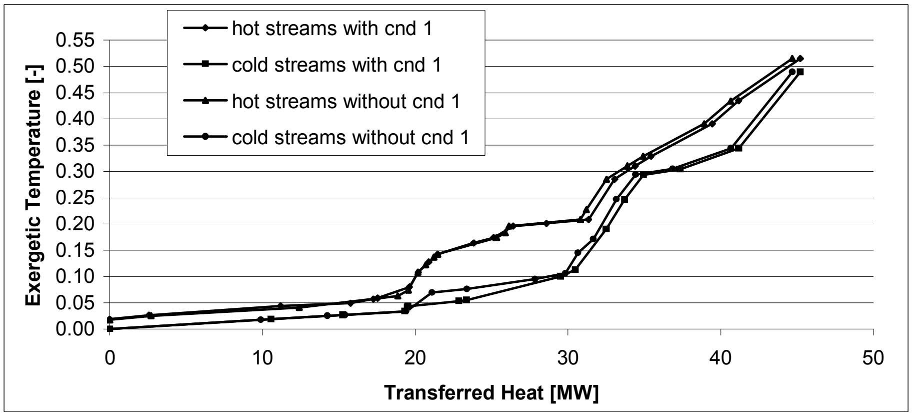


Figure 8. Exergetic temperature profiles for configuration III, for the engine model 18V32GD, with and without an extra condenser (cnd 1).

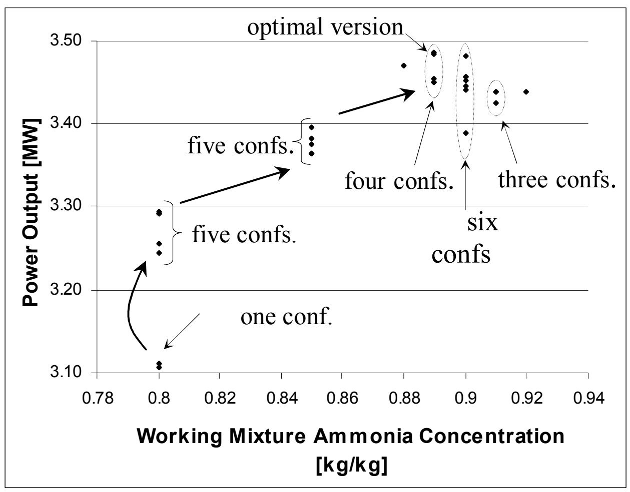


Figure 9. Optimization process for the configuration IIb, for the engine model 18V32GD

The process of optimization of the ammonia-water cycle differs from that of the Rankine cycle. The optimization of an ammonia-water cycle involves both parameter variations, that is variation of the ammonia concentrations of the working fluid, and configuration modifycations. To optimize an ammonia-water cycle with a fixed configuration, the ammonia concentration of the working mixture is kept constant while the ammonia concentration of the basic mixture is varied until the maximum power output is determined. This process is repeated for a number of values of the ammonia concentration of the working mixture, until the highest possible power output of the cycle is found. In this study, the cycle configurations were not fixed as new cycle layouts were to be designed for a new application of the ammonia-water cycle. Therefore, the process of optimization was more complicated than in the case of a fixed configuration. The optimization process started from an initial cycle configuration and a certain ammonia concentration of the working mixture. The ammonia concentration of the basic mixture was then varied. After some initial simulations, possibilities for improved heat exchange in the cycle were often observed. Heat exchangers could be removed or new heat exchangers could be inserted, resulting in an increased power output of the cycle. When the cycle layout was changed, the ammonia concentration of the basic mixture was varied again to find the optimal power output of the new version of the cycle configuration and the ammonia concentration of the working mixture specified. This resulted in a step-by-step optimization process as shown in *Figure 9*.

To optimize the single-pressure Rankine cycles, the maximum pressure of the cycle was varied. To optimize the dual-pressure Rankine cycles, the pressure of the high-pressure components was kept constant while the low-pressure value was varied. This variation of the low-pressure was repeated for a number of high-pressure values until the optimal power output was found. Adaptation of a Rankine cycle to a new application may require modifications of the cycle layout, but not to the same extent as for an ammonia-water cycle.

5.3 Method

The ammonia-water cycle configurations designed in this study have been compared to steam Rankine cycles, to evaluate the potential of a higher power production from gas diesel engine waste heat from ammonia-water cycles. The single-pressure ammonia-water cycles have been compared to both single-pressure and dual-pressure Rankine cycles. To use a single-pressure Rankine cycle as a comparison is rational if the number of pressure levels in the boiler is taken as a measure of the complexity of the cycle. If, instead, the number of degrees of freedom available for the optimization of the cycle is the important parameter for comparison, a dual-pressure Rankine cycle should be used.

To calculate the first and second law efficiencies of a bottoming cycle requires definitions of the energy and exergy inputs to the cycle. In this study there are both configurations that only use the exhaust gas as a heat source and configurations that use all the available heat sources. A cycle configuration may, in some cases, not use all of the available heat in the waste heat streams. The energy or exergy of the heat sources are thermodynamically available to the bottoming cycle, but not technically available due to limitations of the bottoming cycle layout and the properties of the working fluid. For a fair comparison between the different bottoming cycles, the magnitudes of the energy and exergy inputs used in the calculations must be the same for all bottoming cycles. In this study, the total available energy and exergy of all the heat sources have been used in the calculations. There are many possible definitions of the first and second law efficiencies of the bottoming cycles. In this study, four different definitions are possible. First, the shaft work or the net work of the turbines and the pumps can be used in the calculations of the efficiency. Second, the pump work can either be subtracted from the turbine work in the numerator of the efficiency equation or it can be added to the energy or exergy input in the denominator of the equation. Efficiencies calculated by different equations can differ by as much as 11 %.

6. Conclusions

The first and second law efficiencies of all the ammonia-water bottoming cycles investigated in this study, except one, are higher than the efficiencies of the Rankine cycles. The second law efficiency of the best ammonia-water cycle configuration, configuration IIb, is 43-48 % higher than the efficiency of a single-pressure Rankine cycle and 20-25 % higher than the efficiency of a dual-pressure Rankine cycle. This ammonia-water cycle configuration can be simplified by removing one heat exchanger, or possibly a few heat exchangers, without lowering the efficiency of the cycle much.

Exergy and pinch analyses have been performed on the bottoming cycle configurations, and the analyses show that the boiler is the cycle component where the exergy losses between the ammonia-water cycle and the Rankine cycle differ most. The exergy losses of the ammonia-water cycle boilers are smaller than in the Rankine cycle boilers as the non-isothermal boiling behavior of an ammonia-water mixture working fluid generates smaller exergy losses in the boilers. In the condenser, the advantage of a non-isothermal condensing behavior is not that pronounced. The total heat transferred to the ammonia-water cycles is often larger than the heat transferred to the Rankine cycles. This is because the ammonia-water cycles can use heat sources of low temperatures that the Rankine cycles cannot use, due to the characteristics of the working fluid. The total component exergy losses are larger for the most complex ammonia-water cycles than for the Rankine cycles, but they also produce more power than the Rankine cycles.

The process of optimizing an ammonia-water cycle differs from the optimization process of a steam Rankine cycle. To develop an ammonia-water cycle configuration for a new application, the optimization process includes both parameter variations, that is variation of the ammonia concentration levels, and configuration modifications. The layout of a Rankine cycle may, of course, change if the cycle is adapted as a diesel engine bottoming cycle, but not to the same extent as for an ammonia-water cycle. The method used for the exergy analysis influences the results of the analysis. The method includes the choices of reference state, definition of the exergy supplied to the system, and definitions of the first and second law efficiencies of the system.

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Nomenclature

a	vapor quality [kg dry vapor/kg tot]
c_p	specific heat capacity [kJ/kg,K]
E	exergy flow rate [MW]
h	specific enthalpy [kJ/kg]
I	exergy loss [kW], [MW]
m	mass flow rate [kg/s]
P	power output [MW]
p	pressure [bar]
Q	available heat rate of heat source [MW]
Q_{fuel}	fuel input rate (LHV) [MW]
R	universal gas constant [kJ/kgmol,K]
s	specific entropy [kJ/kg,K]
T	temperature [K]
t	temperature [°C]
v	specific volume [m^3/kg]
W	work [MW]
y	ammonia mass fraction [kg NH_3/kg tot]
η	efficiency [-],[%]

Subscripts and Abbreviations

app	approach
c	ammonia-water cycle configuration
ca	charge air
cnd lp	low-pressure condenser
cnd hp	high-pressure condenser
cw	cooling water
DCSS	distillation-condensation subsystem
dea	deaerator
eco	economizer
eg	exhaust gas
eva	evaporator
ex	exergetic
is	isentropic
jw	jacket water
lo	lubricating oil
R	Rankine cycle
rht	reheater
sep	separator
sh	superheater
tbn	turbine
0	reference state
I	first law
II	second law

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