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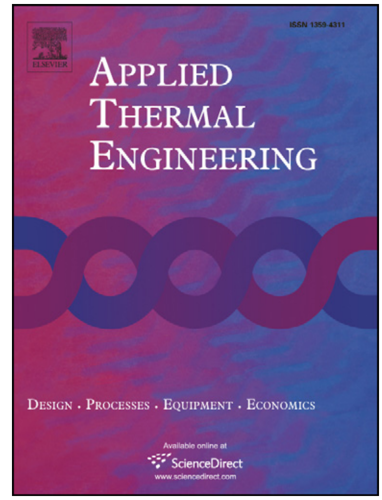

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Abstract

This paper presents the integration of the Kalina cycle process in a combined heat and power plant for improvement of efficiency. In combined heat and power plants, the heat of flue gases is often available at low temperatures. This low-grade waste heat cannot be used for steam production and therefore power generation by a conventional steam cycle. Moreover, the steam supply for the purpose of heating is mostly exhausted, and therefore the waste heat at a low-grade temperature is not usable for heating. If other measures to increase the efficiency of a power plant process, like feed-water heating or combustion air heating, have been exhausted, alternative ways to generate electricity like the Kalina cycle process offer an interesting option. This process maximizes the generated electricity with recovery of heat and without demand of additional fuels by integration in existing plants. The calculations show that the net efficiency of an integrated Kalina plant is between 12.3 and 17.1 % depending on the cooling water temperature and the ammonia content in the basic solution. The gross electricity power is between 320 and 440 kW for 2.3 MW of heat input to the process. The gross efficiency is between 13.5 and 18.8 %.

Keywords

Kalina, Ammonia-water, Power cycle, Combined heat and power plant, coal, cogeneration

1. Introduction

The existing combined heat and power plant at the Industriepark Hoechst has a dry flue gas treatment system with an inlet temperature of 130 °C. Before the flue gas enters the system, it is cooled down from 150 °C to 130 °C with water injection. The first aim of the project is to improve the efficiency of this cooling process. The second aim of the project is to look for a technology to enable the use of the waste heat for power generation, i.e. waste-heat-to-electricity conversion. The Kalina cycle process, which is comparable to that of a Rankine cycle, shows a possibility of using low-grade temperature waste heat for electricity generation.

Different articles deal with the use of waste heat recovering of power plants [1], [2], [3] but every planned project has different parameters which influence the process efficiency and the costs. This article describes thermodynamic models of the Kalina cycle process with different parameters.

2. Combined heat and power plant at Industriepark Hoechst

At the Industriepark Hoechst site in Frankfurt am Main (Germany), a combined heat and power (CHP) plant for cogeneration is installed that is controlled on the basis of the heat demand. This CHP plant supplies the chemical, pharmaceutical and related process industries with process and heat steam.

The plant consists of four boilers with a steam capacity of 830 t/h together. The two gas fired boilers are exclusively for natural gas burning and the other two boilers burn hard coal, heating oil and natural gas. Heating oil is only a backup-fuel. One of the gas fired boilers is connected with the oxygen-rich exhaust gas of a gas turbine. The live steam parameters are 121 bar and 515 °C. Both coal boilers make up the base load and basic power production sources. The two gas fired boilers are used to meet the intermediate and peak load requirements.

The live steam is reduced in four topping-back-pressure turbines from 121 bar to 16 bar in which an electric power of 100 MW is produced. Part of the 16 bar steam is used for feed-water-pre-heating and for the steam distribution system, for the production site. In the second expansion part, steam is expanded from 16 bar to 4,2 bar in back-pressure turbines. Moreover, two steam powered air compressors and one feed-water pump expand steam from 16 to 4.2 bar. The 4.2 bar steam and 16 bar steam are used as an auxiliary supply for feed-water pre-heating. The rest of the 4.2 bar steam goes to the production site and as heating steam to the buildings. Two more turbines reduce steam from 16 bar to 1,2 bar, one is used as feed pump and the other as either feed pump or back-pressure turbine. The 1,2 bar steam goes, finally, into the feed-water pre-heating. All turbines together have an electricity generation capacity of 160 MW. Some of the generated power is spent as auxiliary supply for pumps, fans and mills in the CHP plant. In addition to the electricity produced within, the Industriepark Hoechst is also supplied with external electricity.

3. The Kalina cycle process

In the Kalina cycle process, heat at a low temperature is transferred indirectly to a circulating fluid. Figure 1 shows a schematic diagram of the Kalina cycle process. The working fluid is a mixture of ammonia and water. The ammonia-water mixture has a varying boiling and condensing temperature. During evaporation the mixing ratio of the binary working fluid changes because of the lower boiling temperature of ammonia which evaporates predominantly.

In Figure 2, the different curves present the variable boiling temperatures of different ammonia-water mixtures against the isothermal evaporation of pure water at a pressure of 30 bar. The mixture of ammonia and water boils at a variable temperature depending on its composition. The evaporation and condensation processes are not isothermal. The higher the fraction of ammonia in the mixture, the lower is its boiling temperature. With the increasing ammonia concentration, the specific enthalpy of steam decreases.

Before the turbine, the ammonia-rich steam is separated from the liquid phase in a separator. Afterwards, the ammonia-rich steam passes through the turbine. The generator, coupled to the turbine, produces electricity. The molecular weight of the ammonia (17 kg/kmol) is close to that of the water (18 kg/kmol) and therefore it is possible to use normal back-pressure turbines. The turbine needs no special materials for the ammonia-water mixture.

After the turbine, the steam and liquid phases are merged together and condensed in the condenser. Because of the change in the mixture ratio, the evaporation temperature increases continuously in the wet-steam region while it decreases during condensation. The low temperature (LT) and high temperature (HT) recuperators use the internal residual heat within the cycle. The efficiency is improved with these recuperators.

Worldwide, there are only a few plants made on the basis of the Kalina principle. The most common are given in Table 1. The Kalina cycle process uses as heat sources geothermal heat, waste heat or heat from exhaust gases.

4. Thermodynamic Simulation

The thermodynamic simulation of the Kalina cycle, that is integrated in a combined heat and power plant, was performed with the software EBSILON®Professional, Version 7.01 Beta-Release, from Evonik Energy Services GmbH. This program has a graphical user interface where it is possible to simulate different cycles. Further information can be found in the related handbook [7].

5. Development of base model

On the basis of literature references, a base model was developed [8], [9] using the Kalina power plant in Husavik, Iceland. Parameters used in the thermodynamic simulation are shown in Table 2. The Kalina power plant of Husavik uses geothermal heat. The water from the well has a temperature of about 124 °C and is cooled down to a temperature suitable to the district heating system. The cycle has LT and HT recuperators for pre-heating of ammonia-water mixture. The condenser is feeded with water of 5 °C.

Figure 3 shows the model developed in EBSILON®Professional. The generated electricity is calculated with 2.2 MW. For the Husavik power plant, the same value of the electricity power output is also specified in the literature [8].

This thermodynamical model based on the Husavik plant in Iceland was used for further calculations. Only the energy supply to the evaporator was adapted respectively. In the Husavik plant, the heat source is geothermal heat. On the other hand, the following investigations use low-grade waste heat from flue gas of the combined heat and power plant at the Industriepark Hoechst site.

6. Integration of the Kalina cycle process in the existing CHP plant

The flue gas from both coal combustion systems in the CHP plant leaves the boilers after the heat recovery system with a temperature of about 150 °C. Then, the flue gas is desulfurized in a dry flue gas cleaning system with activated carbon. This process was developed by Bergbau-Forschung and Uhde GmbH, both located in Germany. Today the process is owned by J-Power Entech, Japan. Required NOx emissions are reached only by primary measures with the used and advanced HERENOX-K process [10]. The dry flue gas cleaning process with activated carbon is operated at temperatures of between 90 and 170 °C [11].

The lower boundary of cooling is defined from the sulfuric acid dew point temperature T_S (in Kelvin). This temperature can be determined with empiric equation (1) [12]. In this equation A, B, C and D are constants, p_{H2O} is the partial pressure of water and p_{H2SO4} is the partial pressure of sulfuric acid (both in Pa).

$$T_S = \left[A - B \cdot \ln(p_{H_2O}) - C \cdot \ln(p_{H_2SO_4}) + D \cdot \ln(p_{H_2O}) \cdot \ln(p_{H_2SO_4})\right]^{-1} \quad (1)$$

with A = 0,0029880

B = 0,0000597

C = 0,0001161

D = 0,0000062

The sulfuric acid dew point temperature is calculated with 123 °C for the condition at the CHP plant.

Before the flue gas cleaning system the flue gas is cooled down to 130 °C through direct water injection. It is possible to use this available heat potential between 150 and 130 °C in a Kalina cycle process to generate electricity power.

The rate of conversion in investigated Kalina cycle processes are limited by the Carnot efficiency η_C. The Carnot efficiency can be calculated with equation (2). The cold temperature is T_C and the hot temperature is T_H (both in Kelvin).

$$\eta_C = 1 - \frac{T_C}{T_H} \qquad (2)$$

The ambient temperature is 298.15 K and the average logarithmic temperature difference for flue gas cooling between 150 and 130 °C is 413.07 K. As a result the Carnot efficiency is 27.8 %. In winter time the Carnot efficiency can rise to nearly 32.7 %.

Table 3 shows the conditions of the flue gas from both coal boilers.

7. Results of simulation

Five cases were investigated. Table 4 presents the parameters for simulation of different effects to find the optimized points of efficiency. The content of ammonia, cooling water temperature and, with it, condenser pressure are varied for each case. Case 1 is based on the Husavik plant and Case 2 has a variation in the NH_3 content. The Industriepark Hoechst uses cooling water from the river Main. A cooling water temperature of 5 °C like in Husavik is not achievable. The average water temperature in summer is about 20 °C. Thus, in Cases 3 to 5 the cooling water temperature is 20 °C and the NH_3 content decreases from Case 3 to 5 to investigate the influence of Ammonia content and therefore condenser pressure.

With increasing pressure before the evaporator, the ammonia content of the steam increases at the same temperature of evaporation. Furthermore, with increasing ammonia content in the ammonia-water mixture and with increasing pressure the specific enthalpy of the live steam and exhaust steam, after the turbine, decreases, see Figure 2 and 4. Moreover, the enthalpy of feed-water increases very slowly (Figure 4). The boiling curves and the dew point curves at different ammonia contents and at different pressures of the binary mixture are displayed in Figure 5.

Figure 7 shows the net electrical efficiency against the turbine inlet pressure. The five cases present a variation of different ammonia rates in the ammonia-water mixture and different cooling water temperatures at the condenser. Low cooling temperatures, and with it low condensing pressures, lead to a higher electric power generation (Cases 1 and 2). While keeping the outlet pressure constant, the higher the inlet pressure to the turbine, the higher is the generation of electricity. The enthalpy of the live steam decreases more slowly than the enthalpy of the exhaust steam (Figure 6). Therefore, the turbine provides a higher enthalpy difference. This condition corresponds to the normal power plant cycle that works with a pure substance, like water, as a working fluid. Furthermore, there is an optimum in the Kalina cycle between generated electricity (Figure 7) and power consumption for the feed pump. With the increasing pressure of the binary mixture, the enthalpy of the live steam decreases. On the other hand, the enthalpy of the feed-water changes marginally. The pre-heating of the basic solution in the high temperature recuperator can be improved at a certain mass flow of the liquid stream coming from the separator. Thus, the enthalpy of the pre-heated binary fluid increases before the evaporator. Consequently, the enthalpy difference is lower because of the constant evaporation temperature. Hence, the mass flow of the circulating basic solution rises exponentially (Figure 8). An increase in mass flow leads to an increase in power consumption of the feed pump (Figure 9).

Figure 7 shows also that at low turbine inlet pressures (< 36 bar), the generated electricity is higher with the low ammonia content in Case 5 than with the higher ammonia contents in Cases 3 and 4. Otherwise, the power consumption of the feed pump is higher because a higher circulating mass flow is needed in Case 5, owing to the lower ammonia content. Hence, the specific ammonia consumption is higher (Case 5) as compared with a case with a higher ammonia content in the binary basic solution (e.g. Case 3). As a result, the operating costs are also higher.

The power of a turbine is the product of the enthalpy difference and the mass flow. At lower ammonia contents, the ammonia steam mass flow is also lower compared to that of the higher ammonia proportions.

Also, the condenser pressure has an influence on the generated electricity. The condenser pressure can be dropped down on the one hand with low cooling water temperatures and on the other hand with decreasing ammonia concentrations in the basic solution (see Table 4). The efficiency of the plant can be improved with optimization of the cold end of the process. An absorption heat pump could be used also with waste heat from the power plant.

A high ammonia content in the basic solution and a high turbine inlet pressure result in a higher liquid amount at the end of the turbine. The moisture content should be kept less than 10 % [13]. This fact has to be considered for the design of the process to avoid turbine damages.

8. Capital cost

It is difficult to determine capital costs of Kalina plants from literature. References expose capital costs of between 2000 and 3000 EUR/kW for small plants with an electric power capacity of less than 500 kW [5]. The capital costs for Kalina plants with more than 6 MW should be comparable to that of conventional power plants [1].

The capital costs between 2000 and 3000 EUR/kW show that the payback period is between 7.9 and 18.5 years for a Kalina plant with an electric power output of 400 kW. If the capital costs decrease to about 1000 EUR/kW the payback period could reach 3 years. This calculations assume 8000 operating hours per year, 35 kW of electric power demand and an electricity price of 60 EUR/MWh.

9. Conclusion

The Kalina cycle process is a modified Clausius Rankine process. The basic solution in a Kalina plant is a binary fluid with a certain ratio of water and ammonia. Using an ammonia water mixture has many advantages:

- The use of a fluid (ammonia) with lower boiling point than water, allows the efficient use of the waste heat that cannot be used for steam production.
- For working fluids with lower boiling points the turbine inlet pressure can be higher and the circulating mass flow is lower (minimization of operating costs).
- The operation of conventional steam turbines is possible.
- Ammonia is easily available and inexpensive.
- The handling and use of ammonia in industrial processes is proven.

For the design of a Kalina cycle, an optimum between heat exchanger surface and the generated electricity has to be found. This optimum is influenced by different parameters such as the temperature level at the condenser (turbine outlet pressure), ammonia content in the binary fluid, turbine inlet pressure, pre-heating or the temperature of heat source.

The temperature of cooling water changes during the year. It is not practical to change the ammonia-water mixture for different cooling water temperatures. For the design of such a Kalina plant a certain ratio should be used as the starting point. During the operation of the plant, the mixture ratio can be used in a specific range for optimization.

The calculations show that the net efficiency of an integrated Kalina plant is between 12.3 and 17.1 %.

The gross electricity power is between 320 and 440 kW for 2.3 MW of heat input to the process. The gross efficiency is between 13.5 and 18.8 %.

Often, in power plants, there are low-temperature heat sources that are not used efficiently. With more detailed investigations, and perhaps public sponsorship, the Kalina cycle process could become an interesting option for power plants to use heat at low temperatures for waste-heat-to-electricity conversion. In addition, this process contributes to increasing efficiency and protecting resources.

10. Acknowledgements

The author is very grateful to the staff of the company Evonik Energy Services GmbH in Bensheim, who provided the software EBSILON®Professional for modeling of circuit processes with ammonia-water mixtures as working fluid, particularly the Kalina cycle process.

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12. Figure Captions

Figure 1: Kalina cycle process – Schematic diagram

Figure 2: Comparison between boiling of pure water and different ammonia-water mixtures at 30 bar

Figure 3: Base model of Kalina cycle

Figure 4: Enthalpy of ammonia-water mixture at different pressures and temperatures

Figure 5: Ammonia-water phase diagram

Figure 6: Boiling point diagram of ammonia-water mixture

Figure 7: Electrical net efficiency against turbine inlet pressure

Figure 8: Mass flow of mixture against turbine inlet pressure

Figure 9: Auxiliary power of feed pump against turbine inlet pressure

13. Tables

Table 1: Kalina projects worldwide [2], [4], [5], [6]

Project name/ Location	Country	Heat source	Electrical Output	Start up
Canoga Park (Demo)	USA	515 °C exhaust gas of gas turbine, later Solar Centaur gas turbine	3 MW, later 6,5 MW	1992-1996
Fukuoka City	Japan	Waste heat from incineration plant	5 MW	1999
Kashima Steel Works	Japan	98 °C water, waste heat of production	3,1 MW	1999
Husavik	Iceland	geothermal brine at 124 °C	2 MW	2000
Unterhaching	Germany	geothermal	3,4 MW	2007

Table 2: Parameters of the base model

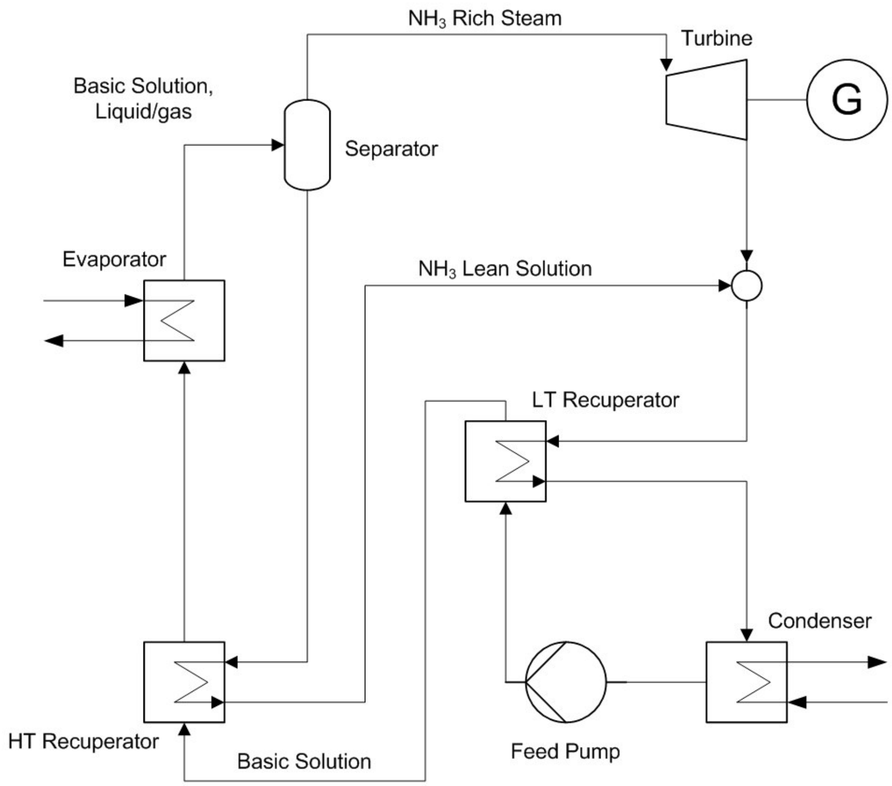
Temperature of the water from the well	124 °C
Temperature of cooled water	80 °C
Mass flow of ammonia-water mixture to evaporator	16.8 kg/s
Ammonia content	82 %
Turbine inlet pressure	32.3 bar
Turbine outlet pressure	6.6 bar
Cooling water inlet temperature	5 °C
Turbine isentropic efficiency	87 %
Mechanical efficiency	98 %
Generator efficiency	96 %
Pump isentropic efficiency	98 %
Pressure losses	1 bar
Minimum temperature differences Evaporator Recuperator Condenser	 6 K 5 K 3 K

Table 3: Parameters of flue gas from the coal boilers

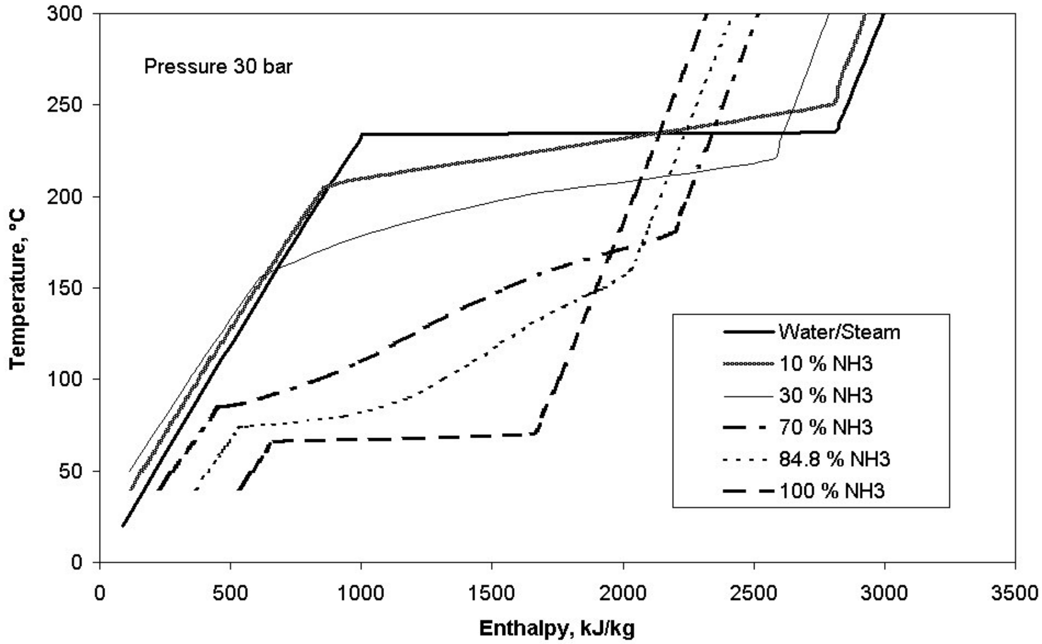
Flue gas mass flow	112 kg/s
Flue gas evaporator inlet temperature	150 °C
Flue gas evaporator outlet temperature	130 °C
Gas composition N_2 O_2 Ar H_2O CO_2 SO_2	 75.2 % vol 5.0 % vol 0.9 %vol 5.6 % vol 13.3 % vol 0.04 % vol

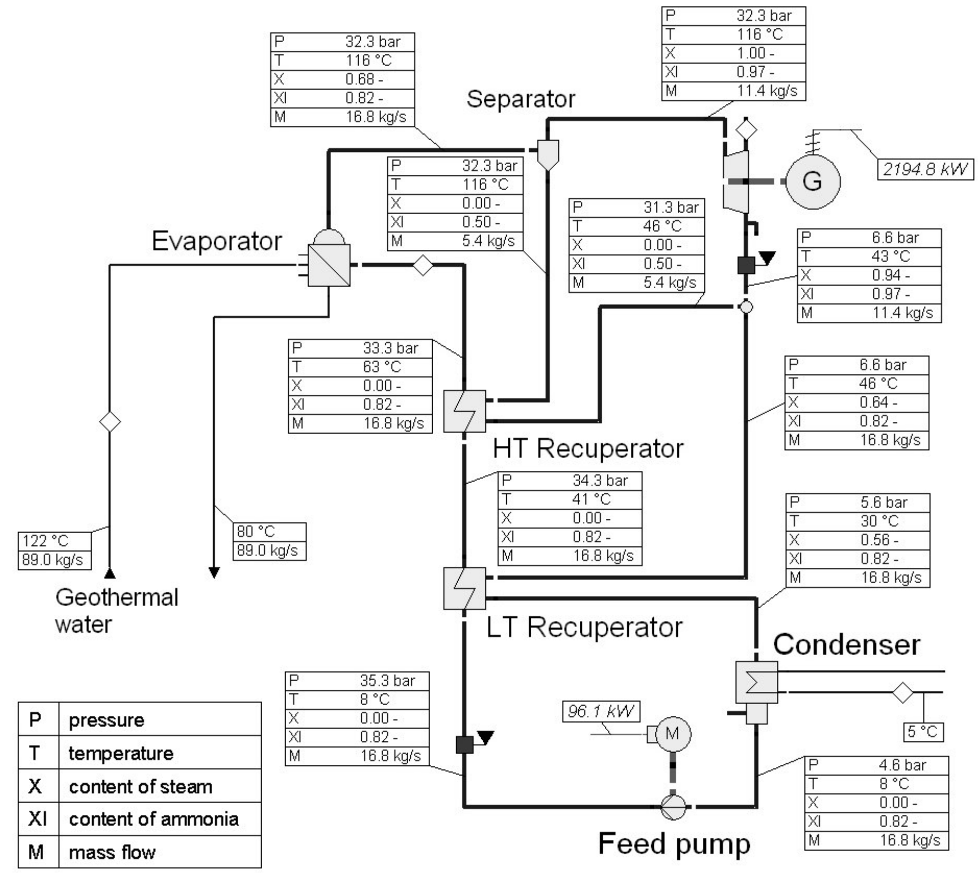
Table 4: Parameters of investigated cases

	Case 1	Case 2	Case 3	Case 4	Case 5
NH3 content, -	0.82	0.7	0.8	0.7	0.5
Cooling water temperature, °C	5	5	20	20	20
Pressure after turbine, bar	6.6	5.1	8.5	7.2	4.2

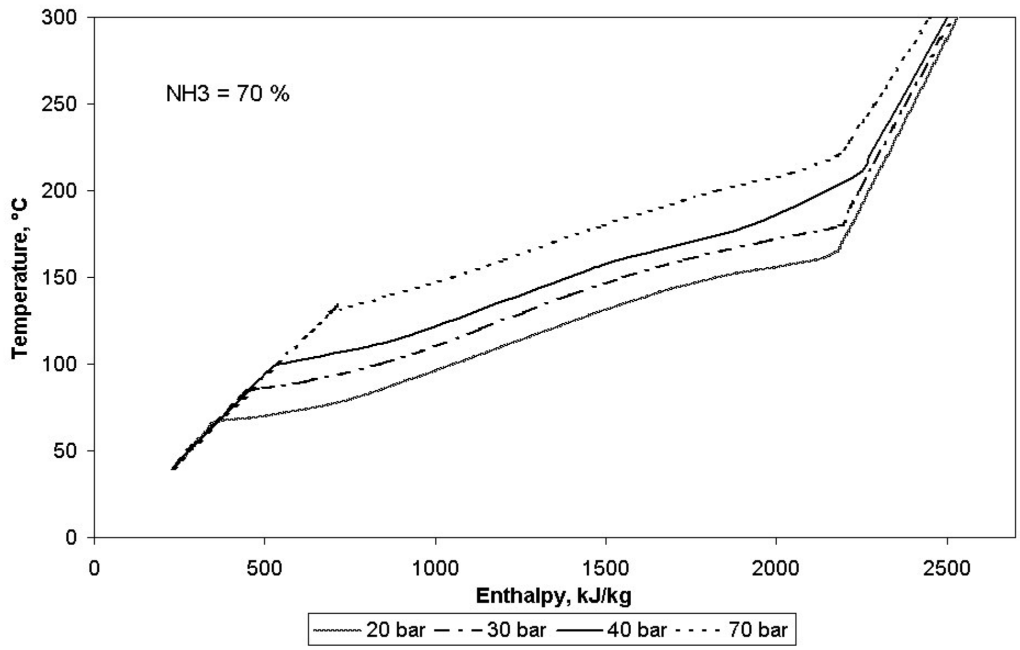


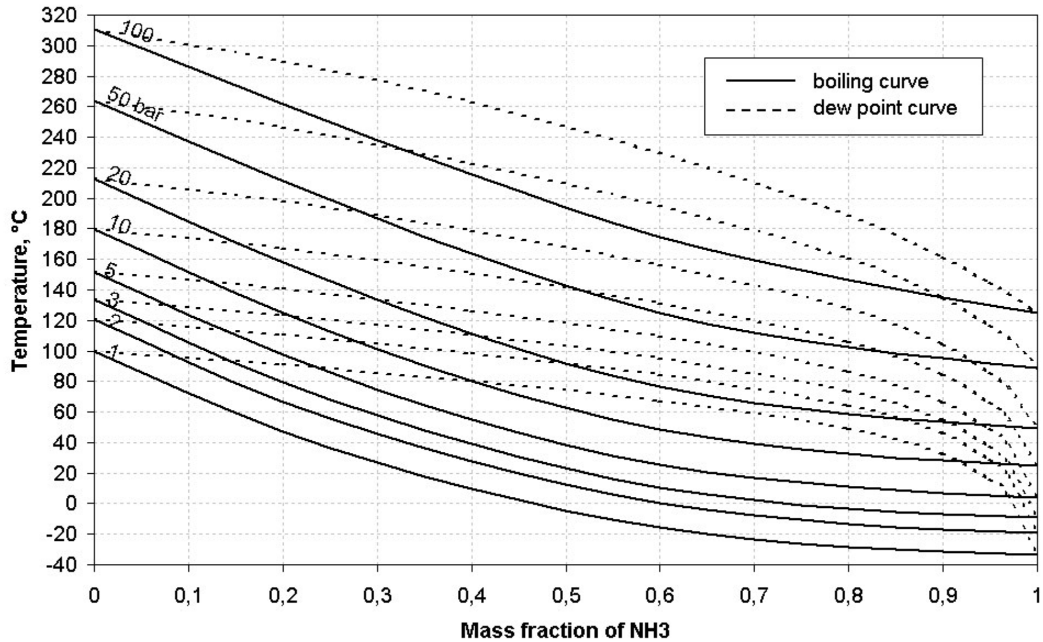
NH3 Rich Steam
Turbine
Basic Solution,
Liquid/gas
G
Separator
Evaporator
NH3 Lean Solution
LT Recuperator
Condenser
HT Recuperator
Basic Solution
Feed Pump

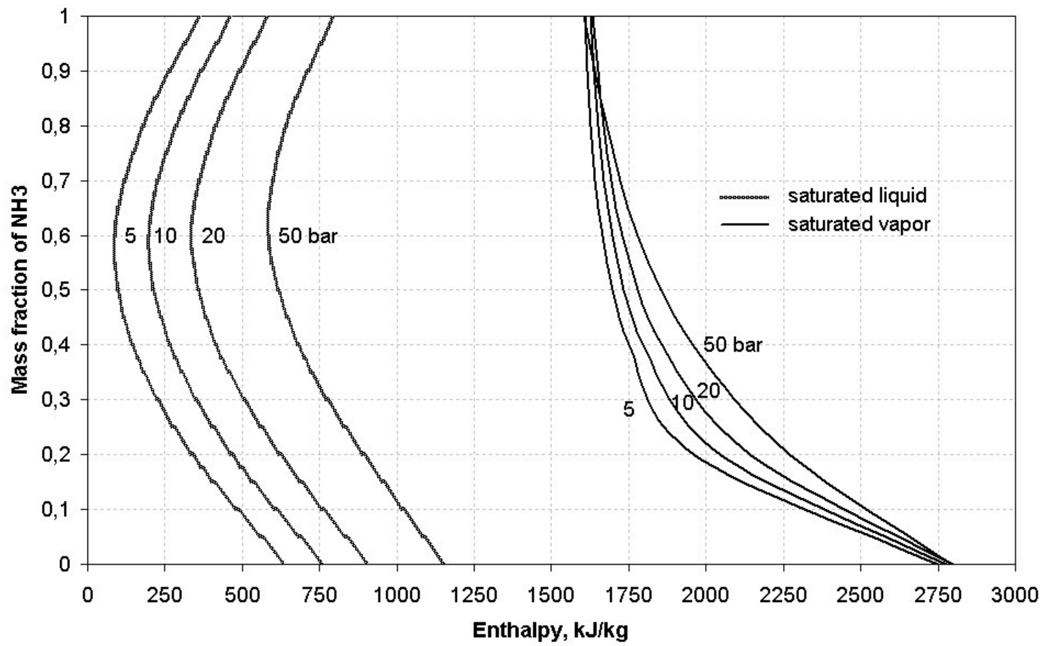

Pressure 30 bar
Temperature, °C
Enthalpy, kJ/kg
Water/Steam
10 % NH3
30 % NH3
70 % NH3
84.8 % NH3
100 % NH3
0
50
100
150
200
250
300
0
500
1000
1500
2000
2500
3000
3500



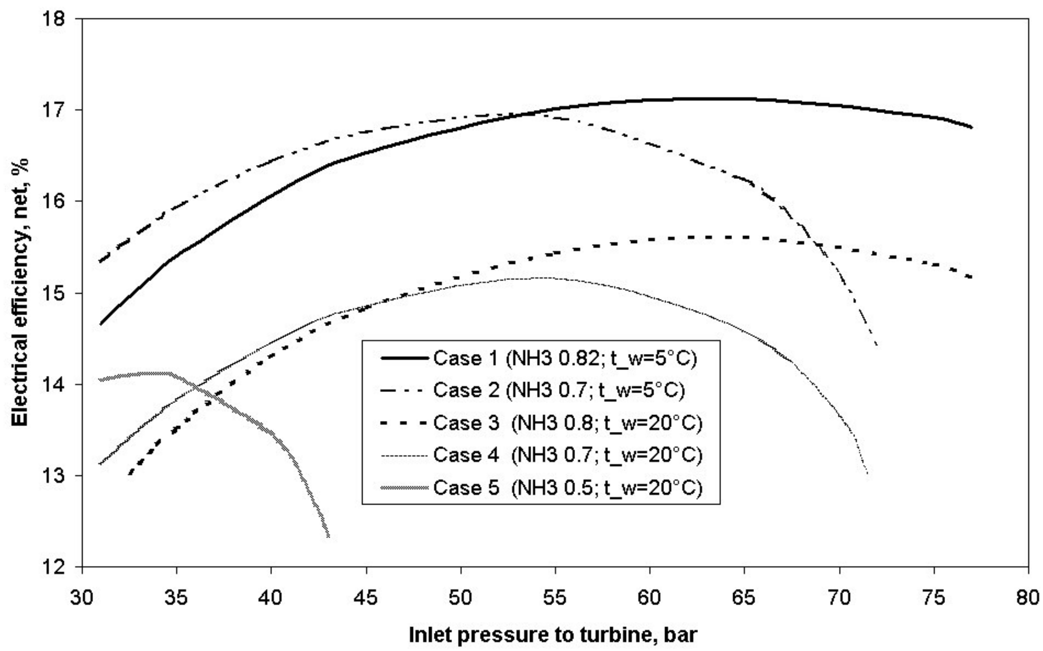
P 32.3 bar
T 116 °C
X 0.68 -
XI 0.82 -
M 16.8 kg/s
Separator
P 32.3 bar
T 116 °C
X 1.00 -
XI 0.97 -
M 11.4 kg/s
G
2194.8 kW
P 32.3 bar
T 116 °C
X 0.00 -
XI 0.50 -
M 5.4 kg/s
Evaporator
P 31.3 bar
T 46 °C
X 0.00 -
XI 0.50 -
M 5.4 kg/s
P 6.6 bar
T 43 °C
X 0.94 -
XI 0.97 -
M 11.4 kg/s
P 33.3 bar
T 63 °C
X 0.00 -
XI 0.82 -
M 16.8 kg/s
P 6.6 bar
T 46 °C
X 0.64 -
XI 0.82 -
M 16.8 kg/s
HT Recuperator
P 34.3 bar
T 41 °C
X 0.00 -
XI 0.82 -
M 16.8 kg/s
122 °C
89.0 kg/s
80 °C
89.0 kg/s
P 5.6 bar
T 30 °C
X 0.56 -
XI 0.82 -
M 16.8 kg/s
Geothermal water
LT Recuperator
Condenser
P 35.3 bar
T 8 °C
X 0.00 -
XI 0.82 -
M 16.8 kg/s
96.1 kW
M
5 °C
P pressure
T temperature
X content of steam
XI content of ammonia
M mass flow
P 4.6 bar
T 8 °C
X 0.00 -
XI 0.82 -
M 16.8 kg/s
Feed pump


NH3 = 70 %
Temperature, °C
Enthalpy, kJ/kg
0
50
100
150
200
250
300
0
500
1000
1500
2000
2500
20 bar
30 bar
40 bar
70 bar

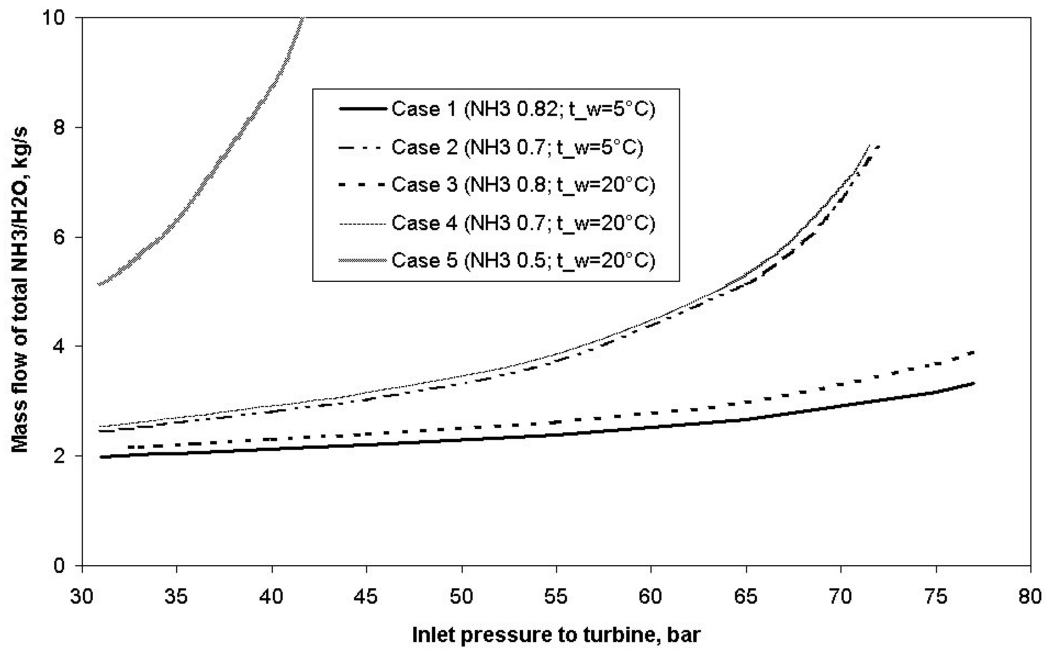

320
300
280
260
240
220
200
180
160
140
120
100
80
60
40
20
0
-20
-40
0
0,1
0,2
0,3
0,4
0,5
0,6
0,7
0,8
0,9
1
Temperature, °C
Mass fraction of NH3
boiling curve
dew point curve
100
50 bar
20
10
5
3
2
1



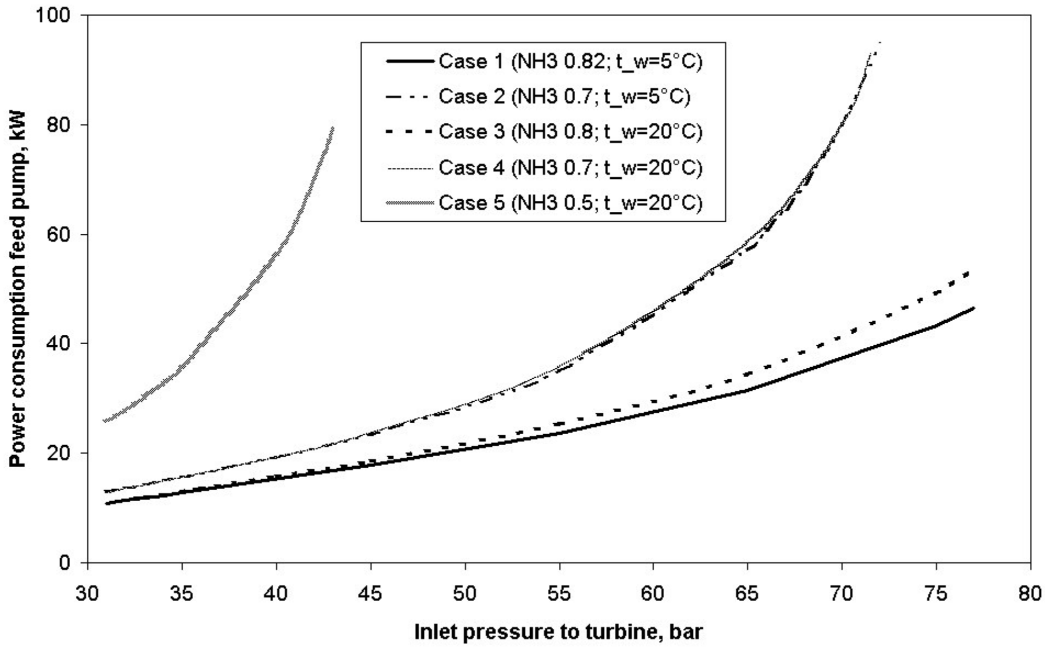
Mass fraction of NH3
Enthalpy, kJ/kg
saturated liquid
saturated vapor
5
10
20
50 bar
5
10
20
50 bar
0
0,1
0,2
0,3
0,4
0,5
0,6
0,7
0,8
0,9
1
0
250
500
750
1000
1250
1500
1750
2000
2250
2500
2750
3000



18
17
16
15
14
13
12
Electrical efficiency, net, %
30
35
40
45
50
55
60
65
70
75
80
Inlet pressure to turbine, bar
Case 1 (NH3 0.82; t_w=5°C)
Case 2 (NH3 0.7; t_w=5°C)
Case 3 (NH3 0.8; t_w=20°C)
Case 4 (NH3 0.7; t_w=20°C)
Case 5 (NH3 0.5; t_w=20°C)



10
8
6
4
2
0
Mass flow of total NH3/H2O, kg/s
30
35
40
45
50
55
60
65
70
75
80
Inlet pressure to turbine, bar
Case 1 (NH3 0.82; t_w=5°C)
Case 2 (NH3 0.7; t_w=5°C)
Case 3 (NH3 0.8; t_w=20°C)
Case 4 (NH3 0.7; t_w=20°C)
Case 5 (NH3 0.5; t_w=20°C)



Case 1 (NH3 0.82; t_w=5°C)
Case 2 (NH3 0.7; t_w=5°C)
Case 3 (NH3 0.8; t_w=20°C)
Case 4 (NH3 0.7; t_w=20°C)
Case 5 (NH3 0.5; t_w=20°C)
Power consumption feed pump, kW
Inlet pressure to turbine, bar
0
20
40
60
80
100
30
35
40
45
50
55
60
65
70
75
80