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Industrial trigeneration using ammonia–water absorption refrigeration systems (AAR)

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Abstract

In many industrial processes there is a simultaneous need for electric power and refrigeration at low temperatures. Examples are in the food and chemical industries. Nowadays the increase in fuel prices and the ecological implications are giving an impulse to energy technologies that better exploit the primary energy source and integrated production of utilities should be considered when designing a new production plant. The number of so-called trigeneration systems installations (electric generator and absorption refrigeration plant) is increasing. If low temperature refrigeration is needed (from 0 to −40 °C), ammonia–water absorption refrigeration plants can be coupled to internal combustion engines or turbogenerators. A thermodynamic system study of trigeneration configurations using a commercial software integrated with specifically designed modules is presented. The study analyzes and compares heat recovery from the primary mover at different temperature levels. In the last section a simplified economic assessment that takes into account disparate prices in European countries compares conventional electric energy supply from the grid and optimized trigeneration plants in one test case (10 MW electric power, 7000 h/year).

Keywords: Trigeneration; Absorption refrigeration; Cogeneration; Energy systems modelling; Ammonia Water

1. Introduction

In many industrial processes there is a simultaneous need for electric power and refrigeration at low temperatures. Examples are found in the food industry [1], cold storage and ice production

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Nomenclature

AAR	ammonia–water absorption refrigeration
COP	coefficient of performance
HRSG	heat recovery steam generator
ICE	internal combustion engine
LHV	lower heating value
NG	natural gas
OGT	optimal generator temperature (°C)
P	pressure (bar)
T	temperature (°C)
R	reflux ratio of the AAR distillation column
ΔP	pressure drop (bar)
ΔT	temperature difference (°C)
$\dot{Q}$	heat flow (MW)
$\dot{W}_{e1}$	electric power produced by the cogenerator (MW)
$\dot{W}_{e2}$	electric power consumption of a compression refrigerator (MW)

Greek letters

ϕ_m	mass flow rate (kg/s)
η	efficiency (%)
$\eta_{\mathrm{el.equiv.}}$	electric equivalent efficiency (%)
ξ	ammonia mass fraction

Subscripts

$1, 2, \ldots, 45$	pipe numbers in process flow diagrams of Figs. 2,3,5 and 7
e	electrical
comp.ref	compression refrigerator
in	input
max	maximum
rec.	recovered
ref.	refrigerating
th	thermal

industry [2], chemical and petrochemical industry, pharmaceutical industry [3–6]. In these cases a trigeneration system, i.e. a cogeneration plant that produces electric energy and uses rejected heat to power an ammonia–water absorption refrigeration (AAR) plant is a viable and, under certain circumstances, economical option.

While water–lithium bromide absorption chillers are becoming more and more widespread and therefore their production is standardized, AAR plants are usually customized to a particular need. AAR is the oldest refrigeration technology, but until recent times it was applied mainly in large scale process plants, mostly in the petrochemical industry. New developments in AAR technology in the smaller range appeared in the literature in the last few years and few installations are known [1,2].

Recovered heat from an internal combustion engine or from a turbogenerator is a suitable source to power an AAR system. Stating that in most cases the plant electric capacity that can fulfill the needs of the mentioned industrial processes is in the range of few to few tens of MW_e and that this is also the best compromise between the initial investment and payback, the thermodynamic optimization of such a system must take into account the different electric efficiencies of the generators and the different refrigeration efficiencies due to different temperature levels at which heat can be recovered.

In this work, only internal combustion (i.c.) engines and turbogenerators are compared, even if fuel cells could be an option in the future. In the following, a specific example is illustrated in order to show thermodynamic implications and give approximate figures regarding operating parameters. A 10 MW_e natural gas (NG) fuelled cogeneration plant is considered: a comparison is made between a modular configuration based on three internal combustion engines, and a single gas turbine.

The somewhat arbitrary choice of comparing a modular plant formed by i.c. engines and a single turbogas genset is due to the fact that NG cogenerative i.c. engines in the 1–3 MW_e range are far more common than bigger units, and have approximately the same efficiency and cost per kW. Another important reason is the availability in this power range of models featuring a pressurized water cooling system as further detailed later. The advantage of a modular plant is that engine maintenance can be scheduled so that plant operation is not interrupted and that even if working at partial load, the overall efficiency does not change. On the other hand small gas turbines (1–3 MW_e) tend to be much less efficient than bigger units, if still less efficient than i.c. engines, therefore the advantages mentioned in the case of i.c. engines are more than offset by the electric efficiency decrement. Despite the above practical assumptions, thermodynamic results emerging from this study can be extended to other configurations, like, for instance, single i.c. engine vs. single turbogas.

As mentioned, in the case of i.c. engines, only units adopting a pressurized water cooling system are considered: the coefficient of performance (COP) of an AAR cycle decreases very rapidly with decreasing generator temperature, therefore a generator temperature of 80–85 °C, corresponding to a cooling water temperature of 90–95 °C leads to an unacceptably low COP (0.2–0.3 for normal ambient conditions, see e.g. [15,16]).

As pointed out for example in [1,3,7], high temperature heat from either gas turbine or engine exhaust is often recovered through a steam generator. The choice of indirect heating for the absorption system allows for a more flexible heat usage: steam can be used for both process purposes and as driving energy for the absorption cycle. Furthermore, decoupling of the AAR working fluid from exhaust gases avoids the potential problem of ammonia leaking in the flue gas. A heat recovery exchanger implies higher initial investments: technical and economical implications are well explained in [1]. As it is shown in Fig. 1, within the above mentioned hypothesis, several configurations are possible:

(A) Internal combustion engines reject heat to a heat recovery exchanger from both the water cooling jackets (the oil cooler temperature is too low) and the stack. The produced superheated water is the heat source for the AAR plant. The AAR cycle receives heat at a temperature which is quite lower than its optimal generator temperature (OGT).

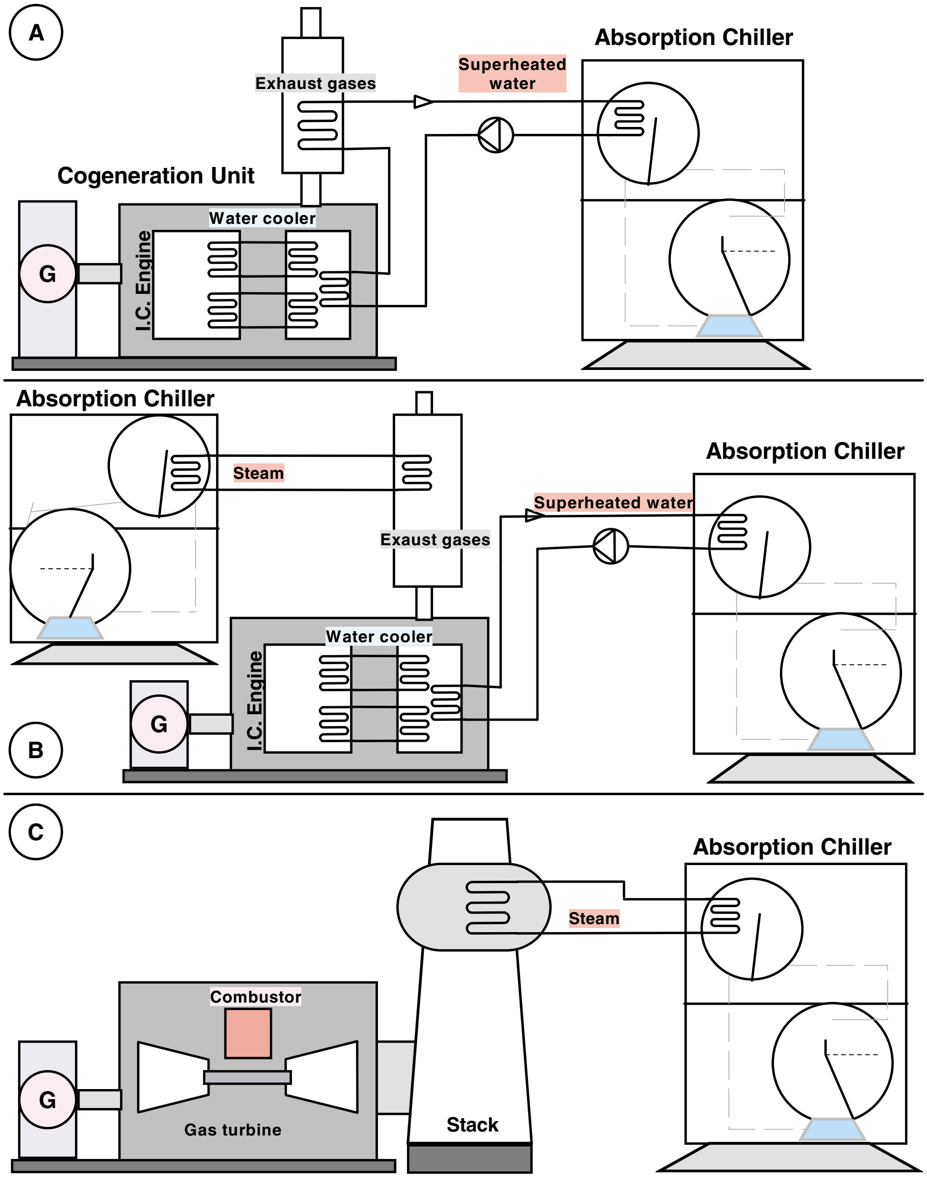

Fig. 1. Possible configurations of a trigeneration system based on an AAR plant.

(B) Internal combustion engines exchange heat at two different temperature levels: steam is produced in a heat recovery steam generator (HRSG) from exhaust gases and superheated water is recovered in a heat recovery exchanger. Two AAR plants are driven by both the HRSG and the pressurized water heat exchanger. In this way the AAR plant that receives heat at high temperature can work close to its OGT.

(C) A turbogenerator rejects heat to a HRSG that power the AAR plant: the AAR plant can work in optimal conditions but the electric efficiency of a turbogenerator in the considered sizes is lower than that of an i.c. engine.

The present work deals mainly with the thermodynamic optimization of the system and addresses the economical aspects using a simplified procedure. A more detailed approach should include other factors like reliability, availability, maintenance, and many others.

In order to compute all energy and mass balances used in the thermodynamic and economic analysis of the considered systems, the simulation software *Cycle-Tempo* [8] was used. The program allows the user to graphically assemble the components of the systems, input the necessary data, solve the system of equations formed by mass and energy balances with a robust algorithm [9] and present calculated results either in graphical or tabular form.

The software comes with a library of performance data for current gas turbines by various manufacturers. The absorption refrigeration system is simulated by first assembling the components (heat exchangers, pumps, valves, etc.) and properly connecting them in the graphical input form. Input operating parameters can be subsequently entered, checking that the underlying system of equations is not over or under-determined. The working fluid model for the ammonia–water mixture is the widely adopted one by Ziegler and Trepp [10]. The necessary code to compute energy and mass balances related to internal combustion engines was added as a custom feature to the software by developing FORTRAN routines that can be linked to the main program [11]. The custom module is interfaced to a database containing performance data for many models of several different manufacturers.

2. AAR plant modelling

The simulation tool was validated by comparing several key values computed by the *Cycle-Tempo* model of an AAR cycle (Fig. 2) with those published in [12]. In the cited reference an extensive comparison with published AAR operating parameters is also included. In the mentioned work design data and cycle configuration were taken from [13] and concern an ammonia manufacturing plant for which 12.6 MW of refrigeration is required at −10 °C. The comparison is presented in Tables 1–3.

If trigeneration is considered, a higher COP of the absorption cycle directly affects the overall optimization: for this reason the configuration in Fig. 3 with the beneficial refrigerant pre-cooling is considered for the single stage absorption cycle included in the analyzed trigenerating systems. Given the recoverable heat, a greater COP implies an increase in refrigeration production which can be compared to the value of the same refrigerating effect produced by a conventional compression refrigeration plant.

The AAR cycle design parameters are listed in Table 4. Values of ammonia concentration in the distilled vapor and the reflux ratio are design parameters for the distillation column and determine the number of theoretical plates: they are taken from common practice in current distillation technology [3,7,14] and take into account the capacity of the plant. Also values of temperature differences in heat exchangers and pump efficiency approximately reflect common

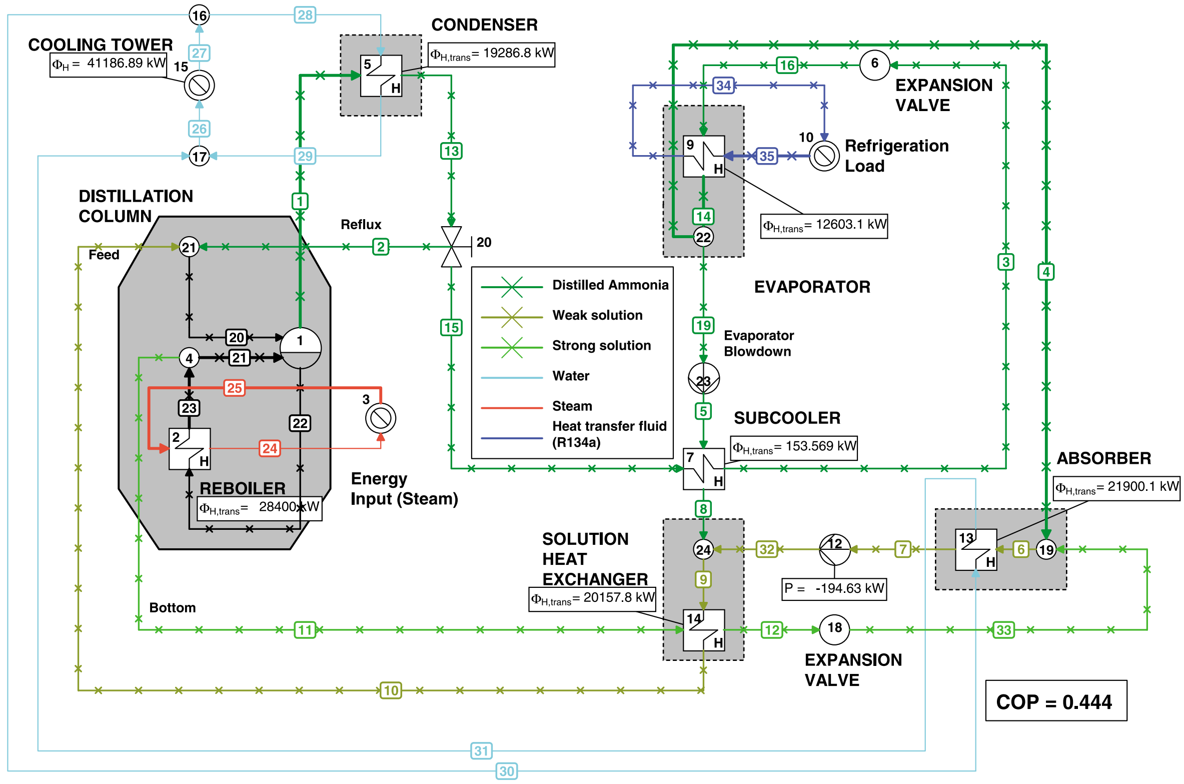

Fig. 2. Process flow diagram of an ammonia water refrigeration cycle: evaporator blowdown is dumped to the absorber section. Displayed results are calculated with *Cycle-Tempo* [8] and are used for validation against calculations published in [12]. Design data and configuration in [12] are taken from [13] and concern an ammonia manufacturing plant.

Table 1
Main thermodynamic operating parameters regarding the absorption system in Fig. 2 for comparison between values computed with *Cycle-Tempo* and published data [12]

	Stream no.											
	1	2	3	4	5	6	7	8	9	10	11	12
Mole flow (kmol/s)	0.986	0.236	0.750	0.711	0.039	3.331	3.331	0.039	3.370	3.370	2.620	2.620
NH_3 conc. (mol%)	99.69	99.69	99.69	100.00	93.95	35.50	35.50	93.95	36.15	36.15	18.76	18.76
Temperature (°C)	63.59	44.00	41.60	−10.00	−9.77	66.71	43.88	40.00	45.11	114.34	152.6	56.6
Pressure (bar)	17.28	17.28	17.28	2.748	17.28	2.748	2.628	17.280	17.280	17.280	17.280	17.280
Vapor quality %	100	0	0	100	0	16.06	0	0	0	1.39	0	0

techno-economic design optimization for the considered components. All pressure drops across components are concentrated in the absorber for simplicity. The cycle maximum pressure ($P_1 = 17.28$ bar) is set so that the constraint of the minimum temperature difference in the condenser is respected. The cycle minimum pressure ($P_{18} = 2.83$ bar) is determined by the mixture saturation condition at the evaporator setpoint temperature.

Table 2
Comparison between *Cycle-Tempo* streams computed values and data reported in [12] for the system of Fig. 2

Stream no.	Mole flow %Difference	Pressure %Difference	Temperature %Difference	NH_3 conc. %Difference
1	0.07	−1.05	0.02	0.01
2	0.40	−1.05	0.00	0.01
3	−0.04	−1.05	0.00	0.01
4	−0.02	−1.33	0.00	0.00
5	−0.29	−1.05	−3.94	0.37
6	0.34	−1.33	1.02	3.32
7	0.34	−1.39	0.27	3.32
8	−0.29	−1.05	0.00	0.37
9	0.33	−1.05	−0.02	3.29
10	0.33	−1.05	−0.39	3.29
11	0.44	−1.05	0.00	4.29
12	0.44	−1.05	4.87	4.29

Table 3
Comparison between *Cycle-Tempo* computed values for heat flows in heat exchangers and data reported in [12] for the system of Fig. 2

	Reboiler duty (MW)	Condenser duty (MW)	Refrigeration duty (MW)	Absorber duty (MW)	Absorber pump (MW)
Cycle-Tempo	28.4	19.286	12.60	21.9	0.194
Ref. [12]	28.4	19.3	12.6	21.8	0.2

3. Modelling of trigeneration systems and thermodynamic optimization

The thermodynamic optimization in 'on-design' conditions of the trigeneration systems under consideration implies the search for the maximum refrigerating effect, which in turn depends from the recoverable heat from the cogenerator and from the COP of the AAR cycle, i.e.

$$\dot{Q}_{\text{ref.}} = \dot{Q}_{\text{rec.Heat}} \cdot \text{COP}_{\text{AAR}}.$$

As known, the COP of a single-stage AAR cycle depends on the generating temperature (T_{23} in Fig. 3) and has an optimum (see for examples [15,16]). For the given operating parameters the OGT is approximately 135 °C (see Fig. 4). The generating temperature however affects also the amount of recoverable heat. Furthermore the OGT cannot be attained by recovering heat from the cooling jackets of an internal combustion engine, therefore the optimization of the 3 systems depicted in Fig. 1 is complex and is illustrated in the following sections. The optimization constraints are listed in Table 5 and are obtained from information regarding current engineering practice, taking into consideration the size of the HRSG.

3.1. Gas turbine and AAR plant

The process flow diagram of the system is reported in Fig. 5 which also displays the main operating parameters resulting from the thermodynamic optimization. Note that also the gas

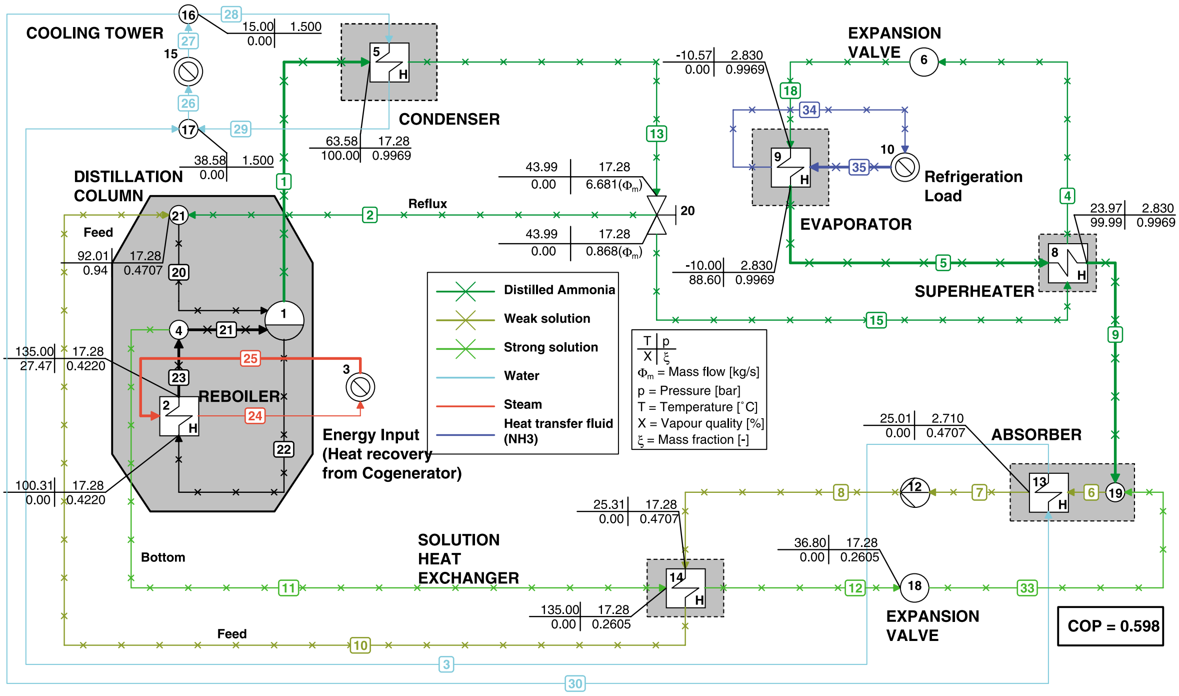

Fig. 3. Process flow diagram of the single stage AAR cycle considered for the coupling with cogenerators. The configuration includes refrigerant pre-cooling made using spent vapor from the evaporator.

Table 4
Design parameters of the considered absorption cycle (components and connections numbering refers to the system in Fig. 3)

Operating parameter	Symbol	Value
Set temperature for the refrigerated environment	T_{34}	−5 °C
Cooling water temperature	T_{27}	15 °C
Ammonia mass fraction in the distillate	ξ_1	0.997
Reflux ratio	$R = \phi_2^m/\phi_{13}^m$	0.13
Min. temperature difference in the condenser (Comp. no. 5)	ΔT_5^{min}	5 °C
Min. temperature difference in the superheater (Comp. no. 8)	ΔT_8^{min}	5 °C
Min. temperature difference in the absorber (Comp. no. 13)	ΔT_{13}^{min}	10 °C
Min. temperature difference in the solution HX (Comp. no. 14)	ΔT_{14}^{min}	10 °C
Min. temperature difference in the evaporator (Comp. no. 9)	ΔT_9^{min}	5 °C
Min. temperature difference in the reboiler (Comp. no. 2)	ΔT_2^{min}	$\geqslant$ 10 °C
Weak solution pump isentropic efficiency (Comp. no. 12)	η_{12}	0.65
Concentrated pressure losses (absorber)	ΔP_{13}	0.12 bar

turbine operating parameters in ISO conditions are indicated and are consistent with a manufacturer data sheet: they represent average performance data for gas turbines in the 10 MW power range for cogeneration applications. No superheater is considered for the HRSG because high

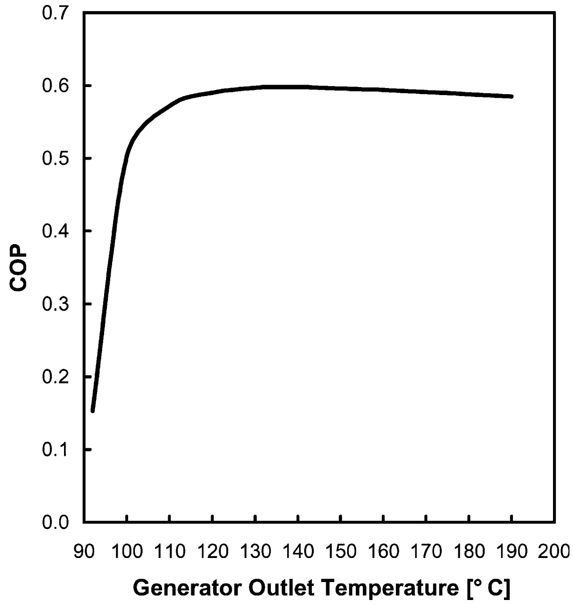

Fig. 4. Effect of the generator (reboiler) temperature on COP for the single stage AAR cycle of Fig. 3. Operating parameters are listed in Table 4.

Table 5
Constraints for the trigenerating systems thermodynamic optimization

Design constraint	Symbol	Value (°C)
HRSG pinch point	$\mathrm{Min}(\Delta T_{25}^{\mathrm{min}}, \Delta T_{26}^{\mathrm{min}})$	15
Reboiler pinch point	$\Delta T_{2}^{\mathrm{min}}$	10
Minimum feedwater temperature (HRSG preheater)	T_{36}	110

temperatures cannot be effectively exploited by the AAR cycle. For simplicity no pressure losses are considered in the HRSG and no consequential decrease in gas turbine performance due to exhaust overpressure.

Optimization variables are the steam evaporator pressure (P_{45}) and the generating temperature (T_{23}) which are simultaneously varied in order to respect design constraints for the HRSG and the reboiler.

As it can be seen in Fig. 6, a generator temperature lower than the OGT, which implies decreasing evaporator pressure, increases the amount of heat flow that can be transmitted to the AAR cycle, therefore the generating temperature which maximize the refrigerating heat flow is 15 °C lower than the OGT (120 °C). This corresponds to an HRSG evaporator pressure of 2.7 bar. In this condition the trigeneration system produces 10.14 MW_e, 25.8 t/h of steam (16.2 MW_{th}) from which 9.57 MW_{ref} of refrigerating effect can be generated. The energy flow entering the system is 32.84 MW_{th} (LHV).

It is also noted that the generating temperature can be varied over quite a wide range (110–140 °C) without considerably reducing the performance of the absorption system.

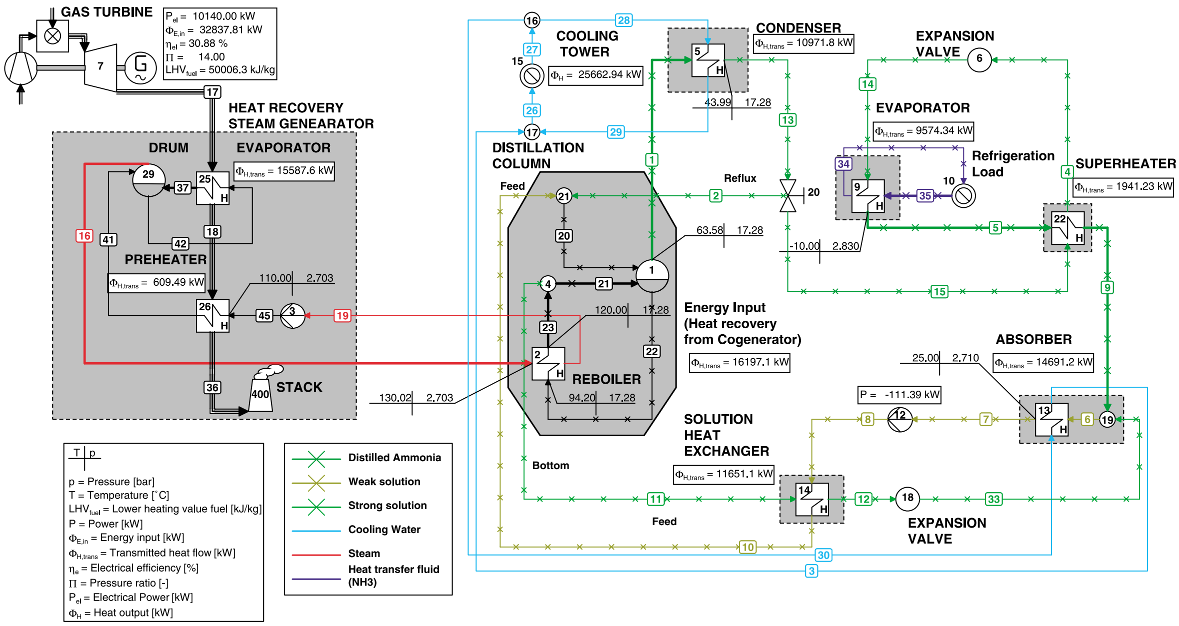

Fig. 5. Process flow diagram of the trigeneration system composed by a gas turbine, an HRSG and an ammonia water absorption plant. The highlighted operating parameters are the result of the thermodynamic optimization.

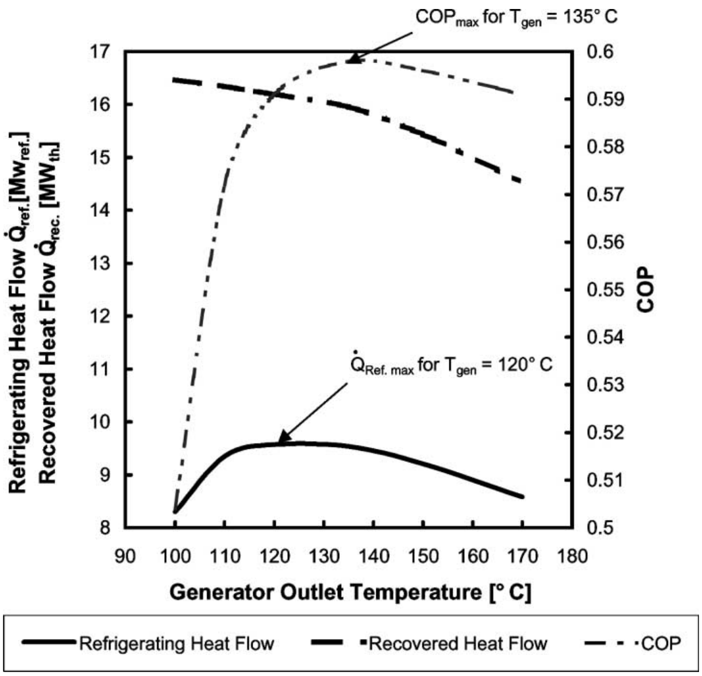

Fig. 6. Refrigerating heat flow and COP of the AAR cycle together with the recovered heat flow from the gas turbine as a function of the generating temperature, which is also bound to the HRSG pressure.

3.2. Internal combustion engines and AAR cycles

In case of an internal combustion engine as the prime mover of the trigenerating system, two configurations are considered: (a) two AAR systems are separately coupled with the flue gas and with the cooling jacket loop; (b) the whole recoverable heat from the i.c. engines system is transferred to the AAR cycle using a single heat exchanger with pressurized liquid water at 120 °C.

To the knowledge of the authors only one manufacturer provide natural gas internal combustion engine for cogenerative applications with the characteristic of the pressurized water cooling, the main reason being that the NG models are derived from diesel engines: for this reason the electric efficiency is lower if compared to some of the same size engines of other manufacturers but higher quality heat is available for cogeneration.

Design parameters for the HRSG of configuration (a) are the same as listed in Table 5, while for the pressurized water heat exchanger the minimum water temperature is set to 105 °C as dictated by engine cooling requirements and the pinch point is set to 10 °C. The generating temperature is set to the highest possible for the cycle exploiting heat from the pressurized water cooling loop (110 °C) and to the optimizing generator temperature (120 °C) for the cycle recovering heat from the exhaust gases.

Results of system calculations are summarized in Table 6. It can be noted that the thermodynamic advantage of recovering part of the heat at optimal conditions for the AAR system does not significantly improve the overall performance of the system, therefore the increase in the investment cost due to a second AAR plant is not justified.

Fig. 7 shows the process flow diagram and main operating parameters of the system formed by 3 internal combustion engines and the AAR refrigeration plant. The trigeneration system in design condition (full load) would produce 10.15 MW_e, 641.2 t/h of superheated water at 120 °C corresponding to 11.18 MW_{th} and 6.40 MW_{ref} refrigerating heat flux. The energy flow entering the system is 25.71 MW_{th}.

3.3. Comparison of trigeneration plant configurations

One possible index of thermodynamic performance is defined by relating utility production to the energy input in purely electric terms: the refrigeration produced by the trigeneration system is equivalent to the electric power necessary to generate the same cooling effect by means of a conventional compression refrigerator. An electric equivalent efficiency is therefore defined as

$$\eta_{\mathrm{el.equiv.}} = \frac{\dot{W}_{e1} + \dot{W}_{e2}}{\dot{Q}_{in}}$$

where $\dot{W}_{e1}$ is the electric power produced by the cogenerator and $\dot{W}_{e2}$ is the electric power consumption of a conventional compression refrigerator for comparison. It is also noted that

$$\dot{W}_{e2} = \frac{\dot{Q}_{ref}}{\mathrm{COP}_{\mathrm{comp.ref.}}}.$$

Table 6
Comparison between (a) i.c. engines coupled with a single AAR cycle recovering heat from the water cooling loop at 120 °C and (b) i.c. engines coupled with 2 AAR cycles, one recovering heat from the cooling loop at 120 °C and the other from the exhaust stack

i.c. Engines system characteristic	
Electrical efficiency	39.5%
Total electric power	10.155 MW_e
(a) 3×3385 *MW_e i.c. engines + AAR*	
Cooling water outlet temperature	120 °C
Cooling water mass flow	178.12 kg/s
Generator temperature	110 °C
COP	0.572
Recovered heat flow	11.18 MW_{th}
Refrigerating heat flow	6.40 MW_{ref}
(b) 3×3385 *MW_e i.c. engines + $2 \times$ AAR*	
Cooling water outlet temperature (1)	120 °C
Cooling water mass flow (1)	84.85 kg/s
Generator temperature (1)	110 °C
COP (1)	0.572
Recovered heat flow (1)	5.711 MW_{th}
Refrigerating heat flow (1)	3.38 MW_{ref}
Evaporator pressure (2)	2.703 bar
Steam mass flow (2)	3.379 kg/s
Generator temperature (2)	120 °C
COP (2)	0.591
Recovered heat flow (2)	5.328 MW_{th}
Refrigerating heat flow (2)	3.05 MW_{ref}
Total refrigerating heat flow (1 + 2)	6.42 MW_{ref}

If it is stated that $COP_{comp.ref.} = 3.0$ is representative of the performance of compression refrigeration plants for the same capacity and evaporator temperature level, the optimized trigeneration system $\eta_{el.equiv.}$ is 40.60% for the turbogas-based trigeneration plant and 47.79% for the i.c. engine-based. This number can be compared to the electric efficiency of modern combined cycle power plants which can be regarded as *benchmark* electric efficiency for NG energy conversion. Being the margins of electricity sale quite low, the comparison can be meaningful as a first instance economical index, in case all cogeneration incentives are neglected.

4. Simple economic assessment

In order to give a first rough estimate of the economic feasibility of the proposed trigeneration systems given present economic boundaries for the European market, the trigeneration systems are compared to conventional energy supply in two exemplifying situations regarding two countries with wide differences in energy prices and policies with respect to industrial cogeneration.

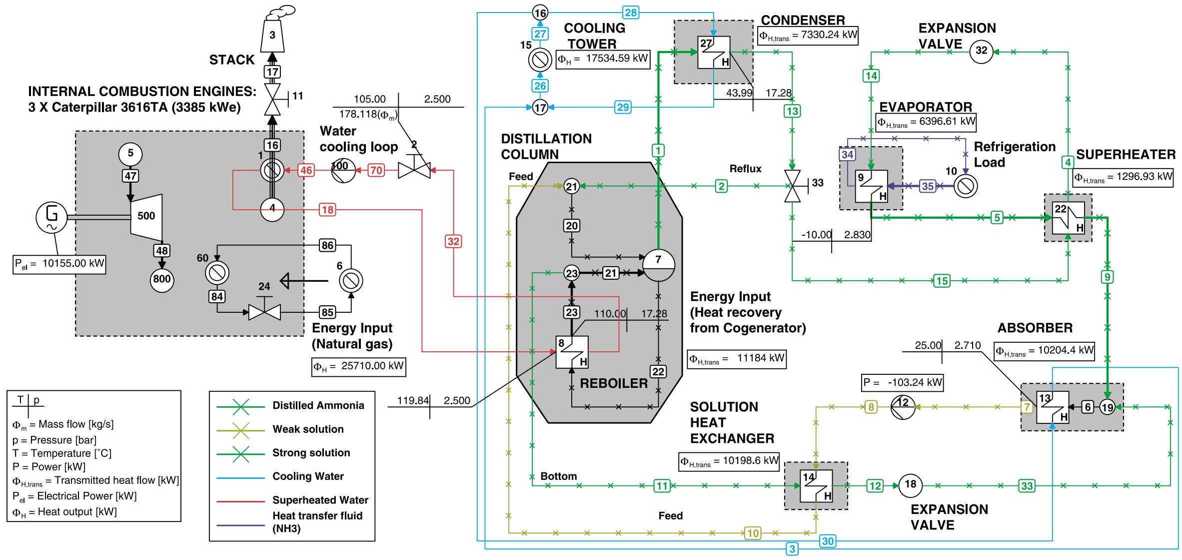

Fig. 7. Process flow diagram of the trigeneration system composed by three natural gas internal combustion engines connected in parallel and an ammonia–water absorption plant.

Table 7
Simplified economic assessments of trigeneration configurations

	GT + AAR	ICE + AAR
Yearly natural gas consumption (GJ/year)	827.492	647.892
Electric energy production (MW h_e/year)	70.980	71.085
Refrigeration production (MW h_{ref}/year)	66.990	44.772
Specific investment cost for prime mover + heat recovery equipment (€/kW_e)	725	837
Specific investment cost for AAR plant (€/kW_{ref})	558	558
Total investment cost (M€)	12.690	12.063
Yearly electric energy consumption with conventional supply (MW h_e/year)	93.310	86.009
	Italy	Netherlands
Natural gas specific cost (€/MJ)	0.0057	0.0045
Electric energy specific cost (€/kW h)	0.0794	0.0476
Simple payback period for GT + AAR (years)	4.8	12.0
Simple payback period for ICE + AAR (years)	3.9	7.9

Energy cost estimates refer to 2001. GT = gas turbine; ICE = internal combustion engine; AAR = ammonia–water absorption refrigeration plant. The yearly electric energy consumption with conventional supply includes the energy needed by a compression plant with COP = 3 to produce the same amount of refrigeration as the AAR plant.

The proposed trigeneration systems are compared to a situation in which the electric energy and the refrigeration are obtained conventionally, i.e. by acquiring the amount of electric energy equivalent to the generator production augmented by the amount needed by compression refrigerators to produce the same cooling flow as the absorption plant of the trigeneration system.

It is supposed that the plant is operated 7000 h/year at full load and that all the produced electric power is utilized on-site. Maintenance and capital repayment costs are neglected for simplicity. In the case of conventional energy supply it is supposed that the needed refrigeration is provided by a compression refrigeration system having COP = 3.0, which is an average value for plants of the considered capacity and temperature levels.

Table 7 reports main results of the simple payback period calculation. Average energy market prices for year 2001 are taken from reports published by Eurostat with regards to Italy, while for the Netherlands the natural gas price estimate is obtained from the major producer and the electric energy price is computed from the average market price and includes average transport costs.

Fiscal aspects related to buying natural gas for cogeneration are taken into account: in the Netherlands there is presently a 0.05 €/kW$_e$ tax rebate if electricity is produced by cogeneration and with an electric efficiency greater than 30%. No fiscal benefits on the investment cost are considered while they are available in the Netherlands under some circumstances but are difficult to take into account. Data for specific investment costs of turbogas and gas engines cogeneration systems are obtained from specialized engineering companies and compared to usual assumptions in the field. AAR specific investment cost for the installed plant are taken from the literature [4] and compared with other source of information received from engineering companies involved in the sector.

The scope of the presented simplified economic analysis is to provide a frame for the positioning of the proposed high efficiency cogenerating technology. It can be noted that the calculated simple payback period is comparable with that of cogenerating plants in the same power range, even if the initial investment is considerably higher. As it is the case with cogeneration, the payback period is strongly influenced by the electricity price and it is seen that the completely liberalized market in the Netherlands caused a considerable decrease of the kW h price, thus adversely affecting cogeneration.

5. Concluding remarks

This paper presents a system study on trigeneration plants composed by gas turbines or gas internal combustion engines which drive an ammonia–water absorption refrigeration plant through a heat recovery exchanger producing either steam and–or pressurized water in the case of the i.c. engines. Indirect coupling with the absorption plant has the advantage of giving the possibility of other process uses for the recovered heat thus possibly increasing the utilization factor. Another advantage is safer operation.

The system modelling is accomplished with a software for thermodynamic systems calculations, appropriately customized in order to simulate the whole absorption cycle and the overall energy and mass balances of the prime movers. The tool allows for easily changing the plant configuration and input variables.

The system study focuses on the comparison of plant configurations for a 10 MW$_e$ trigeneration system for industrial applications. The evaporator temperature of the absorption cycle is set to −10 °C. The mentioned design parameters are believed to be representative of many possible applications in the food, pharmaceutical and ice production industries to name a few. The considered plant configurations comprise a gas turbine coupled to the AAR plant through an heat recovery steam generator, three i.c. engines in parallel producing pressurized water from the cooling loop and steam from the exhaust, both driving separate AAR cycles at different temperature levels and finally the same i.c. engines system in which the whole heat is recovered through a pressurized water cooling heat exchanger driving a single absorption plant.

It is showed that the most complex plant configuration with heat recovery at two temperature levels has no practical advantage while it is noted that for the turbogas system the optimizing heat recovery temperature for the trigeneration system is not correspondent to the optimal generating temperature of the absorption cycle. As expected, the higher electric efficiency of i.c. engines in the considered power range is compensated by a higher heat recovery from the gas turbine which in turns allows for a higher refrigeration production. If the refrigeration is evaluated in terms of the electric power needed to produce it with a conventional compression plant, the trigeneration configuration based on the internal combustion engines is better if compared to the one based on gas turbine. The balance between electric and refrigerating capacity together with considerations regarding maintenance and modularity are the most likely driving factors in a possible techno-economic assessment.

A simple economic evaluation positions the studied trigeneration plant in the same payback frame as more traditional cogeneration plants even if the initial investment is considerably higher. Two different tariff systems applicable in European Communion countries are considered as an example: the two countries have very different energy prices and regulations and these factors reflect in wide differences in the calculated payback period. The simple payback period of the most favorable plant configuration (engine based trigeneration) is 3.9 years in Italy and 7.9 years in the Netherlands.

As it is the case for all heat recovery equipment when the heat exchange temperature is low, the heat exchanging surfaces are the predominant factor for the overall investment. If the trigeneration system is based on a turbogas generator, heat could be recovered at a higher temperature than that suitable for an AAR systems: if advancements on absorption refrigeration technology will make available other working fluids and therefore an increase in COP will be attainable at higher temperatures, this could have a beneficial effect on the initial investment for an industrial trigeneration plant.

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