Exergoeconomic Analysis of an Absorption Refrigeration and Natural Gas-Fueled Diesel Power Generator Cogeneration System

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ABSTRACT

This work presents a thermoeconomic analysis of a cogeneration system using the exhaust gas from a natural gas-fueled diesel power generator as heat source for an ammonia-water absorption refrigeration system. The purpose of the analysis is to obtain both unit exergetic and exergoeconomic costs of the cogeneration system at different load conditions and replacement rates of diesel oil by natural gas. A thermodynamic model of the absorption chiller was developed using the Engineering Equation Solver (EES) software to simulate the exergetic and exergoeconomic cogeneration costs. The data entry for the simulation model included available experimental data from a dual-fuel diesel power generator operating with replacement rates of diesel oil by natural gas of 25%, 50% and 75%, and varying engine load from 10 kW to 30 kW. Other required data was calculations using the GateCycle software, from the available experimental data. The results show that, in general, the cogeneration cold unit exergetic and exergoeconomic costs increases with increasing engine load and decreases with increasing replacement rate of diesel oil by natural gas under the conditions investigated. Operating with 3/4 of the rated engine power and replacing 50% of diesel oil by natural gas, the exergoeconomic cost of the produced power is increased by 75%, and the exergoeconomic cost of the produced cold is decreased by 17%. The electric power unit exergetic and exergoeconomic costs indicate that the replacement of diesel oil by natural gas is feasible in the present considerations for engine operation at medium and high loads.

Keywords: natural gas; cogeneration; absorption refrigeration; power generation; exergoeconomic analysis.

1. INTRODUCTION

Absorption cycles have emerged as promising alternatives for cooling and refrigeration applications in terms of emissions (zero ozone depletion fluids and zero global warming fluids) and low electric energy consumption [1]. Absorption refrigeration systems are capable of using different energy sources such as fossil fuels, renewable energies and waste heat from other thermal systems, such as engine exhaust gas. Diesel engines deliver high amounts of easily recovered waste heat energy, but requires single-effect absorption cycles to operate with low activation temperatures once the exhaust gas temperature is low [1].

Several authors [2-5] studied cogeneration plants with reciprocating engines. An absorption refrigeration system using waste heat from a 55-passenger bus engine could completely meet the coach cooling demand of 30 kW when the vehicle operated over 100 km/h [6]. A simulation analysis of an absorption refrigeration unit operating with the exhaust gas from a diesel engine showed that the overall system performance could be improved with precooling of the engine intake air charge to increase the pressure ratio, while maintaining low cycle temperature ratio [7]. A combined effect Lithium-Bromide (LiBr) absorption chiller was shown to have higher coefficient of performance (COP) and cooling capacity than a single effect absorption chiller, both using waste heat from the exhaust gas of an engine as energy source [8].

The generator is the component of an absorption refrigeration system with the highest exergy destruction, followed by the absorber, condenser and evaporator [9]. It was reported that the generator, evaporator, condenser and absorber temperatures, and the solution concentration affect the absorption refrigeration system COP [10]. In another work, it was found that the highest performance of an ammonia-water absorption refrigeration cycle integrated with a marine diesel engine was obtained at high generator and evaporator temperatures, and low condenser and absorber temperatures [11].

An experimental investigation of a solar thermal powered ammonia-water absorption refrigeration system indicated a chiller COP of 0.69 and cooling capacity of 10.1 kW, with generator inlet temperature of 114°C, condenser/absorber inlet temperature of 23°C, and evaporator outlet temperature of -2°C [12]. A hybrid absorption-compression refrigeration powered by mid-temperature waste heat reached a COP of 0.71, which is about 42% higher than that of a conventional ammonia-water absorption refrigeration system [13]. An energetic and exergetic study of a 10 RT (35.17 kW), single effect, indirect heated LiBr absorption chiller coupled to a 30 kW microturbine, cooling tower and a heat exchanger, using the Engineering Equation Solver (EES) software to evaluate the influence of the system parameters, reports a COP around 0.7 for microturbine operation between 80% and 100% of the rated load [14]. The COP of a double effect LiBr absorption chiller, of 1.411, was higher than that of a single effect chiller, of 0.809, both operating with waste heat recovery from a boiler flue gas [15]. The exergetic efficiency of the absorption systems decreased with increasing flue gas temperature due to the rise of irreversibility in the low pressure generator.

A thermoeconomic evaluation is important to improve absorption refrigeration systems, as they are less efficient than vapor compression systems [16,17]. An exergoeconomic analysis was performed for three classes of double-effect, lithium bromide-water absorption refrigeration systems, showing that lower investment costs are attained when the temperatures of the high-pressure generator and the evaporator are high, the condenser temperature is low [16]. The exergoeconomic analysis of series flow double effect and combined ejector-double effect lithium bromide-water absorption refrigeration systems pointed out that, with similar operating conditions, the overall system investment cost and the product cost flow rate are lower for the combined cycle [17]. In another work, an exergoeconomic analysis of a 5 kW ammonia-water refrigeration cycle with hybrid storage system, with the solution properties determined by the EES software, showed that the system overall exergetic efficiency tends to a constant at temperatures higher than 120ºC, and decreases with evaporator temperature lower than -15°C [18]. A thermoeconomic analysis performed for an absorption refrigeration system using the exhaust gas of a hydrogen-fueled diesel engine as energy source showed that engine combustion is the process with the highest exergy destruction, and that it is feasible to operate the system at intermediate and high engine loads [19].

This work presents a thermoeconomic analysis of a cogeneration system consisted by a direct heated, single effect, ammonia-water absorption refrigeration system using as heat source the exhaust gas from a diesel power generator fueled by diesel oil and natural gas. The exergetic and exergoeconomic analysis uses a similar approach as that applied by [19]. The main aim is to study the performance parameters of the cogeneration system and to get both exergetic and exergoeconomic costs of power and cold production at different engine load conditions and replacement rates of diesel oil by natural gas. The absorption refrigeration system was modeled in the EES software, using as input data the experimental data available from a production, stationary diesel engine operating in dual fuel mode with replacement rates of diesel oil by natural gas of 25%, 50% and 75%, under variable load [20]. The experimental data available was also used by the GateCycle software to calculate unmeasured exhaust gas properties required by the absorption chiller simulation model.

Natural gas has clean burn features and produces lower levels of most pollutant emissions components, compared with gasoline and diesel oil [21-27]. In dual fuel operation with diesel oil, natural gas combustion increases heat release by about 27-30%, compared to operation with diesel oil as a single fuel [28]. This results in reduced specific fuel consumption, especially at high engine load and intake air temperature [21-23,29]. The use of different replacement rates of diesel fuel by natural gas affects combustion duration and exhaust gas temperature and, therefore, the energy available to be used by the absorption refrigeration system [20]. In this work, the replacement rates chosen allows for the analysis of a broad range of engine operation with equal increments of natural gas in the fuel. The investigation of a cogeneration system composed by an absorption refrigeration system and a diesel power generator operating with different replacement rates of diesel oil by natural gas finds no resemblance to previous works [5,9,11,14].

2. DESCRIPTION OF THE COGENERATION SYSTEM

A schematics of the absorption refrigeration system simulation coupled with the diesel power generator is shown in Fig. 1. The power generation unit features a four-stroke, four-cylinders, naturally aspirated diesel engine, with direct fuel injection and 44 kW rated power at 1800 rpm. The engine has a compression ratio of 17:1, 3.922 L total displacement, 120 mm bore and 120 mm stroke. The simulated absorption refrigeration system is direct heating, single effect, with ~17 kW (~ 4.8 TR) of capacity and COP ~

0.6. The refrigeration system has a generator containing a double rectifying column with a second heat exchanger and a binary mixture as a combination of refrigerant and absorbent. Ammonia is the refrigerant and water is the absorbent.

A strong liquid solution with a large concentration of ammonia refrigerant leaves the absorber at state 1 and is pumped to the condensing pressure, being preheated in the heat exchanger to reduce heating at state 3 (Fig. 1). The heated strong solution enters into the generator, which produces a weak liquid solution with low concentration of ammonia refrigerant at the bottom, at state 4, and nearly pure ammonia (99.98%) vapor at the top, at state 7. The weak solution enters the heat exchanger and flows through the pressure reducing valve to enter the absorber. The strong solution is sent to the condenser at state 7, then it condenses to sub-cooled liquid at state 8. The liquid enters the heat exchanger to cool at state 9 and, then, it enters the expansion valve. The ammonia leaving the expansion valve at state 10 enters the evaporator, where the liquid phase vaporizes to absorb the refrigerant load in the system. The refrigerant is further heated in the heat exchanger prior to being absorbed in the weak-liquid solution in the absorber at state 12, and, then, it returns to state 1, thus restarting the refrigeration cycle.

3. METHODOLOGY

Figure 2 presents the stages used in the methodology of the cogeneration system simulation: processing of the available data from experimental engine testing and calculation of exhaust gas related parameters by the GateCycle software, and simulation of the absorption refrigeration system and exergoeconomic analysis in the EES software. The experimental data and the results from the GateCycle software are used as input data for the EES software, and both softwares operate independently. The simulation does not aim to optimize the performance of the combined cogeneration system, but to produce the necessary information for an exergetic and exergoeconomic analysis of system operation with different replacement rates of diesel fuel by natural gas.

The experimental data was available from tests in a production, four-stroke, four-cylinder, stationary diesel engine, model MWM D229-4, of 44 kW rated power operating at 1800 rev/min, compression ratio 17:1 and direct diesel fuel injection (Tab. 1) [29]. For all tested operating conditions, the exhaust gas temperature at the outlet of the refrigeration system generator was 58°C ± 6°C lower than the inlet gas temperature. The engine was operated with varying load from 10 kW to 30 kW and with replacement rates of diesel oil by natural gas of 0%, 25%, 50% and 75% on energy basis. During the tests, the load power range was limited to 30 kW and the natural gas concentration was limited to 75% due to engine instability to operate with natural gas at higher loads and concentrations without major modifications. Additional details of the tests, including the uncertainties of the results, can be found in Ref [29].

The GateCycle software uses the experimental data from the engine tests varying the load applied and the replacement rate of diesel fuel by natural gas (Tab. 1) to calculate unmeasured parameters by bivariate interpolation. The motivation to use the GateCycle software was the possibility to use its internal libraries and adequately estimate the exhaust gas properties required by the simulation model of the absorption refrigeration system.

The compositions of natural gas and diesel oil are presented in Table 2. For calculation of the total exergy of air, exhaust gas, diesel oil and natural gas, it was considered steady state condition, negligible pressure drop and ambient at 30°C, 101.32 kPa [20]. The exergetic efficiency of the diesel power generator ( ) is calculated by [19]:

Where is the output power from the diesel power generator (kW), is the total exergy supplied with the fuel (kW), is the diesel oil mass flow rate (kg/s), is the natural gas mass flow rate (kg/s), is the diesel oil specific exergy (Table 2) (kJ/kg), and is the natural gas specific exergy (Table 2) (kJ/kg).

The simulation model of the absorption refrigeration system, developed in the EES software, was validated against experimental data available from a commercial, Consul CQG22D model ammonia-water absorption refrigerator used for domestic application, of 215 L internal volume [2,31]. The refrigerator COP was kept nearly constant, varying from 0.60 to 0.61, for all engine load range investigated. The thermodynamic simulation of each system component calculates mass, energy, entropy and exergy balances at steady state condition and neglecting pressure drop. The exergetic efficiencies of the ammonia-water absorption refrigeration system ( ) and the system generator ( ) are calculated as [19]:

Where and are the total exergies of the produced cold and the engine exhaust gas, respectively (kW), and is the power consumed by the solution pump (kW). , , are pure ammonia specific exergies at the state 7, evaporator inlet and evaporator outlet, respectively (kJ/kg), and is the exhaust gas specific exergy variation from the generator inlet to outlet (kJ/kg). and are the binary solution specific exergies at the states 3 and 4, respectively (kJ/kg). and are the binary solution specific enthalpies at the pump inlet and outlet, respectively, in kJ/kg. is the exhaust gas flow rate at the generator inlet (kg/s); and are pure ammonia flow rates at the evaporator and state 7, respectively (kg/s). and are the binary solution flow rates at states 3 and 4, respectively (kg/s).

Other results from the third stage of the simulation include component irreversibilities, generator efficiency, heat transfer in the condenser, evaporator, absorber and heat exchanger, pump power, COP, and the thermodynamic properties used in the exergoeconomic analysis (stage 5 in Fig. 2). The exergoeconomic analysis refers to the exergetic costs of system operation according to the physical structure of the cogeneration system (Fig. 1), using the streams thermodynamic properties and component parameters that were computed in the previous stages (Fig. 2). For the exergoeconomic analysis, the unit exergetic cost at the cogeneration system inlet was assumed as 1, the exergetic cost balance was applied for components and junctions, and the costs distribution in the bifurcations was performed proportionally to the exergy. Additionally, the negentropy was considered to be generated by dissipative equipment, such as the condenser and absorber, and the exhaust gas from the diesel power generator was taken as waste when assigning the costs.

Table 3 presents the fuel-product definition for each component of the cogeneration system, based on which the cogeneration plant productive structure was built (Fig. 3). Figure 3 shows that the negentropy ( ) related to heat dissipation in the condenser is located in the generator, heat exchanger, evaporator and expansion valve (streams 39 to 42), and related to heat dissipation in the absorber is located in the pressure reducing valve, generator, solution heat exchanger and solution pump (streams 35 to 38). The negentropy distribution adopted was based on the criteria that some components work with nearly pure ammonia (heat exchanger, expansion valve and evaporator) while others use ammonia-water solution (solution heat exchanger, pressure reducing valve and solution pump) or both (generator). For the generator, two negentropy streams were located (36 and 39) because it works with two fluid types: nearly pure ammonia (flow 7 in Fig. 1) and ammonia-water solution (flows 3 and 4 in Fig. 1).

The diesel engine negentropy is due to dissipation of the chemical exergy of the exhaust gas flow to the ambient (ambient product in Tab. 3 and stream 47 in Fig. 3). From the 50 streams presented in Fig. 3 and the assumptions mentioned before, 50 equations were written in the EES software to compute the unit exergetic cost for each stream, with the aim to calculate the unit exergetic cost (, in dimensionless form) and the specific exergoeconomic cost ( ) of each stream in the productive structure. The main calculated costs were the net electrical power (, in US$/kW.h) and cold produced (, in US$/RT.h, 1 RT = 3.517 kW) by the system at the different loads and fuel replacement rates simulated. The specific costs are calculated by the following equations [19]:

Where is the total stream exergetic cost (kW), is the total stream exergy (kW), is the exergetic efficiency (dimensionless), and is the stream exergoeconomic cost (US$/h).

The exergoeconomic costs for each stream in Fig. 3 are calculated from the exergetic unit costs. The exergoeconomic costs are due to fuel prices, taking into account the initial investment, maintenance and external valorization. The diesel oil and natural gas prices considered in the calculations were 0.2 US$/L and 0.5465 US$/m^3, respectively. These are commercialization prices for thermal power generation established by the Brazilian Ministry of Finance [32]. The calculation of the external valorization was based on Ref. [33], and it includes an investment cost of US$ 13,950.00 for the diesel power generator and US$ 13,167.00 for the absorption refrigeration system. Further details on the exergoeconomic analysis are available in Ref. [20].

4. RESULTS AND DISCUSSION

Figure 4 shows that the engine exergetic efficiency increased with increasing load power. This trend is explained because, at low loads, a high fraction of the power produced is used to overcome friction losses. At partial load, fluid flows, mixing, heat transfer and combustion processes increase the specific entropy generation, thus reducing the exergetic efficiency. It is also observed that, with increasing diesel oil replacement by natural gas at any load, the engine exergetic efficiency is enhanced due to improved combustion. The increase of natural gas fraction in the fuel also increases the pre-mixed combustion phase, which is a process more efficient than diffusive combustion.

In Fig. 5, it is observed that the exergetic efficiency of the refrigeration system tends to decrease with increasing load, due to rise of heat transfer and irreversibility in the refrigeration system. When the engine load increases, the exhaust gas mass flowrate and temperature are also increased (Tab. 1). Thus, more heat is transferred to the refrigeration system, and the heat transfer process in the refrigeration system regenerator occurs with a higher temperature difference. For those reasons, both entropy generation and irreversibility are increased, causing a decrease of the exergetic efficiency of the absorption refrigeration system. Increasing the replacement rate of diesel oil by natural gas until 50% decreases the exhaust gas temperature (Tab. 1), which can improve the exergetic efficiency.

Figure 6 shows a tendency of reducing generator exergetic efficiency when the engine load is increased, similarly to what was observed for the absorption refrigeration system (Fig. 5). This means that the exergetic efficiency of the absorption refrigeration system is strongly influenced by the generator exergetic efficiency. The generator exergetic efficiency decreases with increasing engine load because of higher entropy generation (or irreversibility) caused by high heat transfer rate and temperature difference between the engine exhaust gas and the refrigeration system working fluid.

Figure 7 shows that the produced cold unit exergetic cost is increased with increasing engine load, and is decreased with increasing replacement rate of diesel oil by natural gas. This means that more exergy is necessary to supply the refrigeration system for each unit of produced cold when increasing engine load. In Fig. 8, it is observed that the produced power unit exergetic cost decreases for medium and high loads while, for low and partial loads, the cost is higher. This means that less exergy is necessary to supply the engine for each unit of the produced power when increasing engine load or, in other words, it is more interesting to operate the engine at high loads to reduce the power generation cost. Increasing the replacement rate of diesel oil by natural gas also decreases unit exergetic cost of power generation.

Figure 9 shows that the exergoeconomic cost of the cogenerated cold is increased with increasing load and decreased with increasing replacement rate of diesel oil by natural gas. The decrease of the exergetic efficiency of the absorption refrigeration system with increasing engine load (Fig. 5) increases the irreversibility and, thus, the final cost of the cogenerated cold. On the other hand, the exergetic efficiency of the absorption refrigeration system is increased with increasing replacement of diesel oil by natural gas (Fig. 5), having a positive effect on the exergoeconomic cost of the cogenerated cold (Fig. 9).

The variation of the exergoeconomic cost of electrical power production is shown by Fig. 10. Unlike cold cogeneration, in this case the trend of decreasing cost with increasing load is due to the increase of the engine exergetic efficiency (Fig. 4), which reduces the irreversibility of the power system. Increasing the replacement rate of diesel oil by natural gas increases the exergoeconomic cost of power production (Fig. 10). Considering the prices of residential rates with taxes, both the use of diesel oil as a single fuel or partially replacing it by natural gas can be competitive in the depicted scenario if the cost of electrical power is lower than the existing rate with taxes. When natural gas is used, the exergoeconomic cost of the produced power is below the existing rate with taxes only at intermediate and high loads. The gaseous fuel cost has a strong influence on the calculated results, playing a major role to make the cogeneration system economically viable.

From comparison of the results of the present work with those when hydrogen was used as fuel in similar conditions [19], the same trends were observed for the produced cold and power exergoeconomic costs (Figs. 9 and 10). Nevertheless, considering the replacement rate of 50%, the reduction of the produced cold exergoeconomic cost is of about 26% when hydrogen replaces diesel oil [19], while, using natural gas instead, the reduction is of around 17% (Fig. 9). When analyzing the produced power exergoeconomic cost, the use of hydrogen is more viable for a slightly larger range of load power [19]. However, natural gas allows for a larger replacement rate of diesel oil, up to 75% without major engine modification, while the maximum replacement rate of diesel oil by hydrogen was 50% [19].

5. CONCLUSIONS

From the results obtained, the following conclusions can be drawn:

– Increasing engine load reduces entropy generation and irreversibility in the engine and increases entropy generation and irreversibility in the absorption refrigeration system;

– Increasing the replacement rate of diesel oil by natural gas decreases entropy generation and irreversibility in both the engine and the absorption refrigeration system;

– The cogeneration cold unit exergetic cost and exergoeconomic cost increase with engine load due to an increase of exergy destruction in the absorption refrigeration system mainly by the reduction of the exergetic efficiency in the generator of the refrigeration system.

– The cogeneration cold unit exergetic cost and exergoeconomic cost decrease with increasing replacement rates of diesel oil by natural gas;

– The electric power unit exergetic cost and exergoeconomic cost decrease with increasing engine load and diesel oil replacement by natural gas rise, being viable in the economic scenario considered if the engine is operated at medium and high loads;

– In comparison with diesel fuel replacement by hydrogen, natural gas provides lower decrease of the exergoeconomic cost of cold production, but allows for a larger range of replacement rate without major engine modification.

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7. NOMENCLATURE

Specific exergoeconomic cost, US$/kW.h or US$/RT.h

Exergoeconomic cost, US$/h

CO Carbon monoxide

COP Coefficient of Performance

EES Engineering Equation Solver

Specific exergy, kJ/kg

Total exergy, kW

Total exergy cost, kW

h Binary solution specific enthalpy, kJ/kg

Unit exergetic cost, kW/kW

LiBr Lithium Bromide

Mass flowrate, kg/s

NG Natural Gas

NMHC Non-Methane unburned hydrocarbons

NOx Oxides of nitrogen

SPECO Specific Exergy Costing

Power, kW

Greek letters

Variation or difference

Efficiency

Subscripts

1, 2,… Flow number (Fig. 1) or stream number (Fig. 5)

Absorber

Absorption Refrigeration System

Cold

Chemical

Condenser

Diesel oil

Diesel Engine

Electric power

Expansion Valve

Evaporator

Diesel oil and natural gas blend

Exhaust gas

Generator

Heat Exchanger

Negentropy

Natural Gas

Pressure Reduction Valve

Refrigerant

Shaft power

Solution Heat Exchanger

Solution pump

Superscripts

Exergetic

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LIST OF TABLE CAPTIONS

Table 1 – Experimental data from diesel power generator operating with natural gas (NG) used in the simulation [20].

Table 2 – Natural gas and diesel data assumed for calculations.

Table 3 – Fuel – Product definition by component for the productive structure.

LIST OF FIGURE CAPTIONS

Figure 1 – Simplified schematics of the absorption refrigeration system coupled to the diesel power generator.

Figure 2 – Summary of the stages of the simulation model.

Figure 3 – Cogeneration plant productive structure.

Figure 4 – Variation of engine exergetic efficiency with load power and natural gas concentration in the fuel.

Figure 5 – Variation of absorption refrigeration system exergetic efficiency with engine load power and natural gas concentration in the fuel.

Figure 6 – Variation of generator exergetic efficiency with engine load power and natural gas concentration in the fuel.

Figure 7 – Variation of produced cold unit exergetic cost with engine load power and natural gas concentration in the fuel.

Figure 8 – Variation of produced power unit exergetic cost with engine load power and natural gas concentration in the fuel.

Figure 9 – Variation of produced cold exergoeconomic cost with engine load power and natural gas concentration in the fuel.

Figure 10 – Variation of produced power exergoeconomic cost with engine load power and natural gas concentration in the fuel.

Table 1 – Experimental data from diesel power generator operating with natural gas (NG) used in the simulation [20].

ENGINE LOAD (kW)	100% DIESEL OIL		75% DIESEL OIL + 25% NG			50% DIESEL OIL + 50% NG			25% DIESEL OIL + 75% NG		
	EXHAUST GAS TEMP (°C)	DIESEL OIL FLOW RATE (kg/h)	EXHAUST GAS TEMP (°C)	DIESEL OIL FLOW RATE (kg/h)	NG FLOW RATE (kg/h)	EXHAUST GAS TEMP (°C)	DIESEL OIL FLOW RATE (kg/h)	NG FLOW RATE (kg/h)	EXHAUST GAS TEMP (°C)	DIESEL OIL FLOW RATE (kg/h)	NG FLOW RATE (kg/h)
0	143.01	1.91	145.00	1.86	-	138.63	1.93	-	148.41	1.32	-
10	224.09	3.37	220.00	2.94	0.786	214.64	2.88	1.573	223.64	2.33	2.359
20	324.17	5.17	312.00	4.50	1.171	307.11	4.10	2.341	313.99	3.48	3.511
30	447.79	7.15	430.00	6.32	1.627	419.85	5.80	3.254	420.20	5.23	4.880

Table 2 – Natural gas and diesel data assumed for calculations.

Natural gas		Diesel	
Component	Molar fraction	Component	Mass fraction
Nitrogen	0.015	Carbon	0.8670
Carbon Dioxide	0.007	Hydrogen	0.1271
Methane	0.871	Oxygen	0.0032
Ethane	0.078	Nitrogen	0,0000
Propane	0.029	Sulfur	0.0020
Hexane	0.000	Wet	0.0005
Hydrogen	0.000	Ash	0.0002
Lower Heating Value, kJ/kg	47451	Lower Heating Value, kJ/kg	43000
Specific exergy, kJ/kg	49243	Specific exergy, kJ/kg	42145

Table 3 – Fuel – Product definition by component for the productive structure.

COMPONENT	FUEL	PRODUCT
Diesel engine		
Electric generator		
Ambient		
Generator		
Condenser		
Evaporator		
Absorber		
Solution pump		
Heat exchanger		
Solution heat exchanger		
Expansion valve		
Pressure reducing valve		

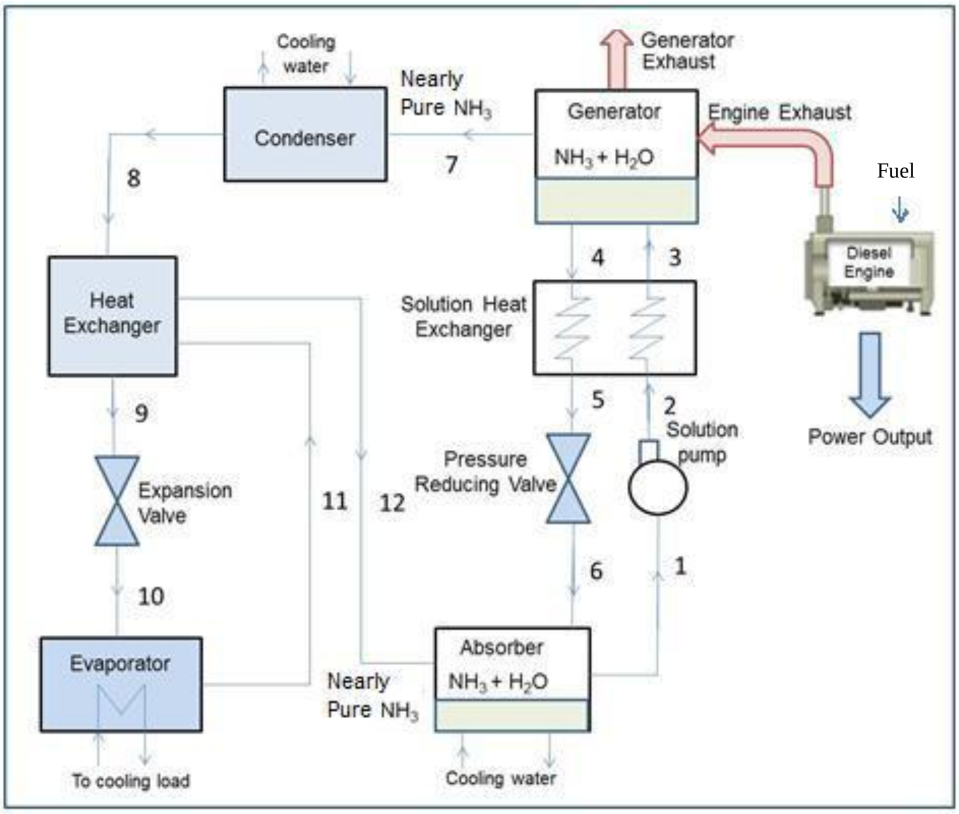

Figure 1 – Simplified schematics of the absorption refrigeration system coupled to the diesel power generator.

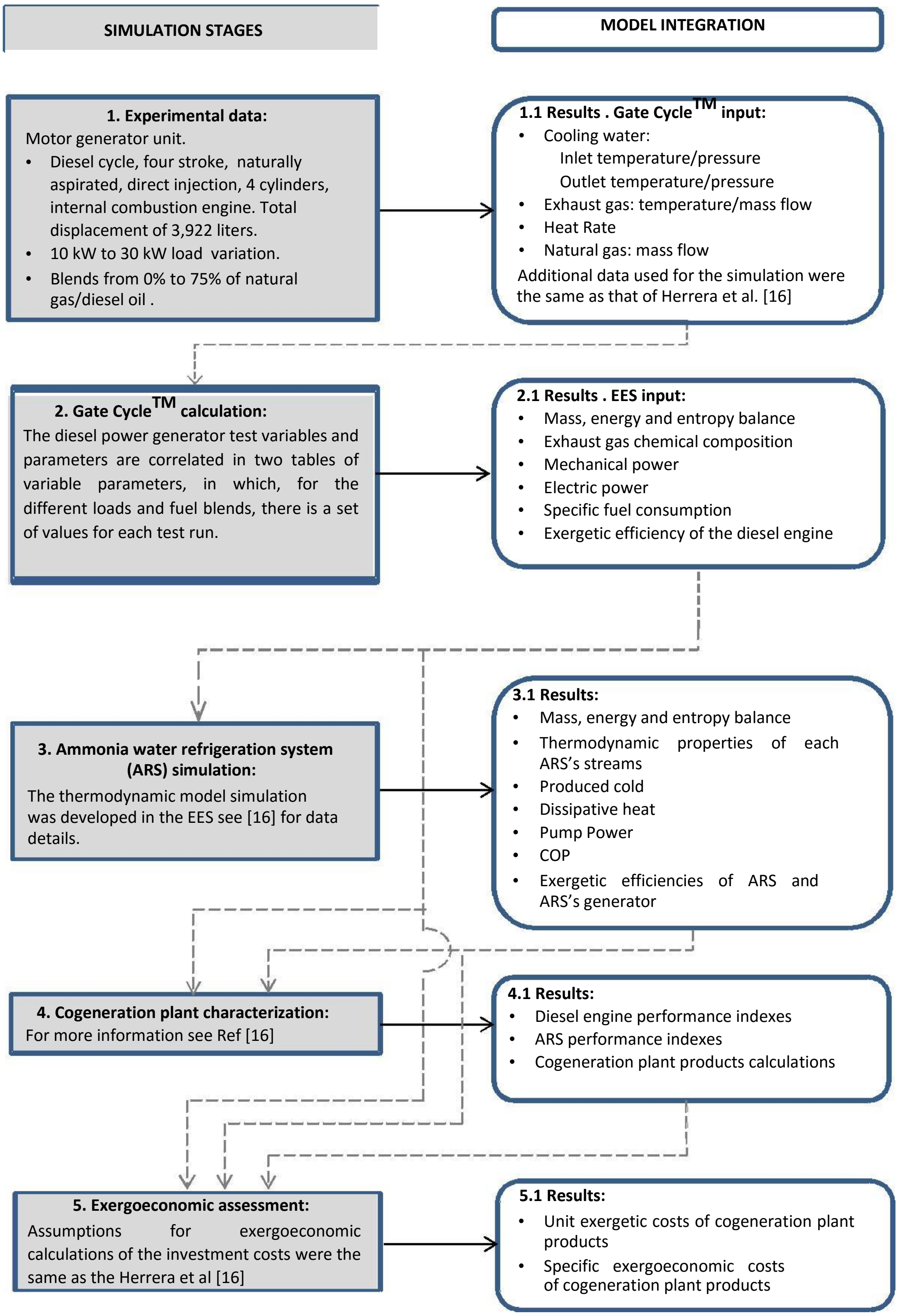

Figure 2 – Summary of the stages of the simulation model.

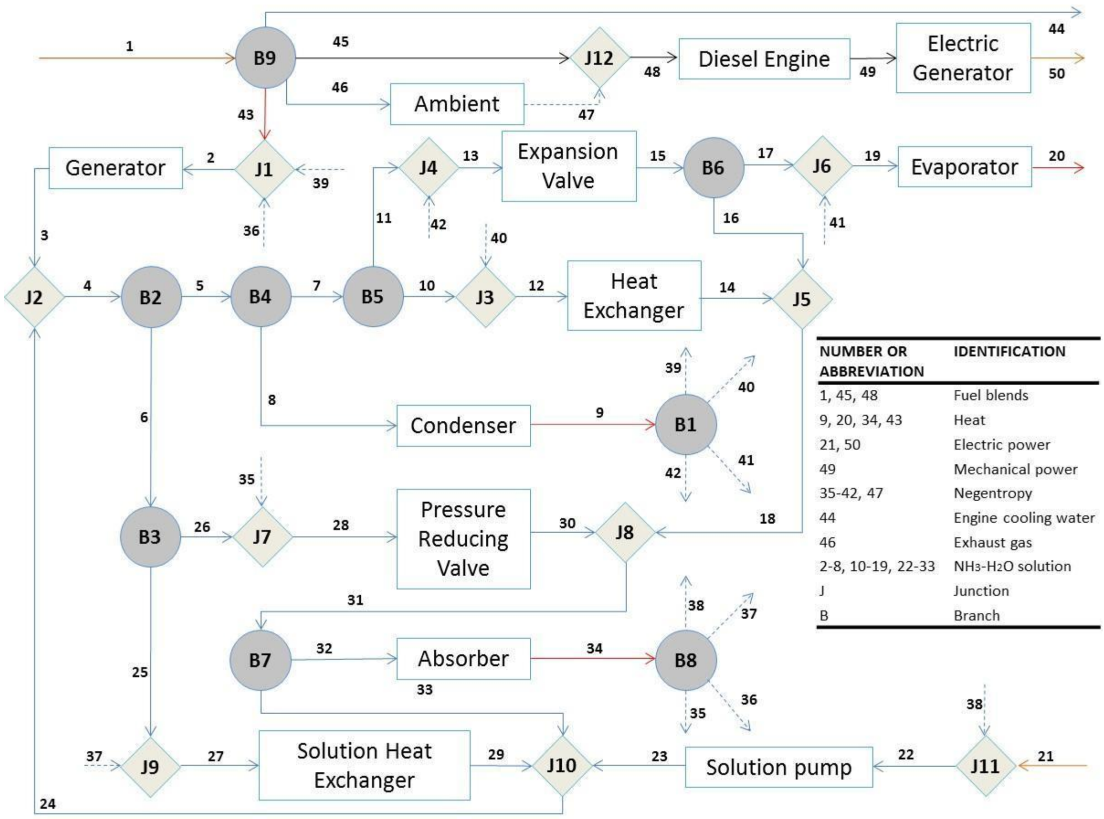

Figure 3 – Cogeneration plant productive structure.

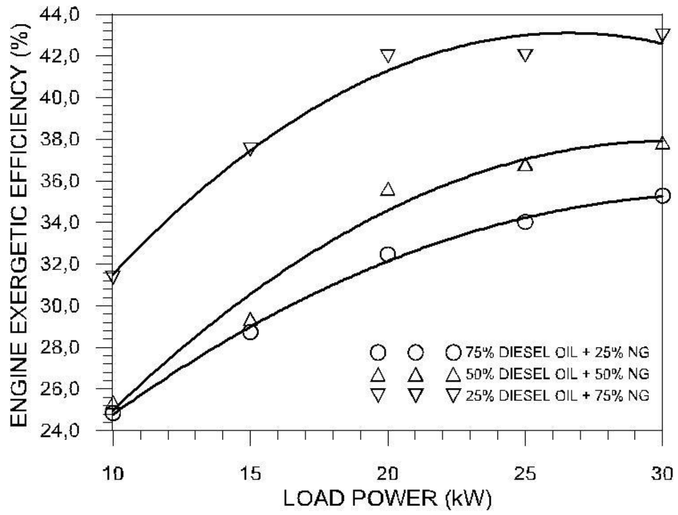

Figure 4 – Variation of engine exergetic efficiency with load power and natural gas concentration in the fuel.

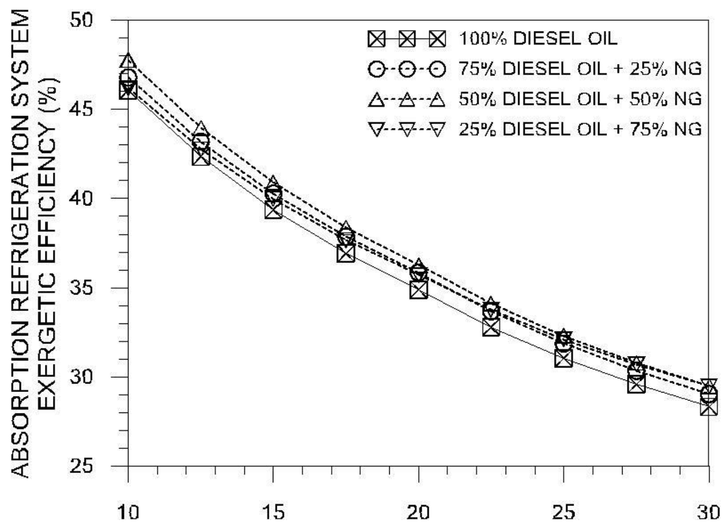

Figure 5 – Variation of absorption refrigeration system exergetic efficiency with engine load power and natural gas concentration in the fuel.

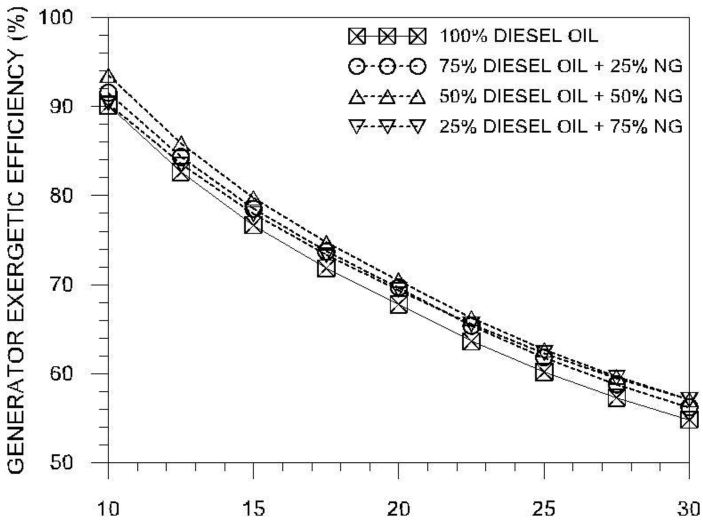

Figure 6 – Variation of generator exergetic efficiency with engine load power and natural gas concentration in the fuel.

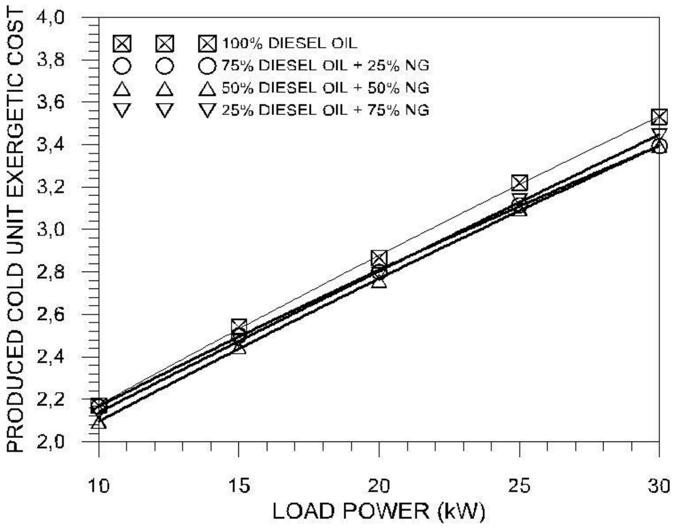

Figure 7 – Variation of produced cold unit exergetic cost with engine load power and natural gas concentration in the fuel.

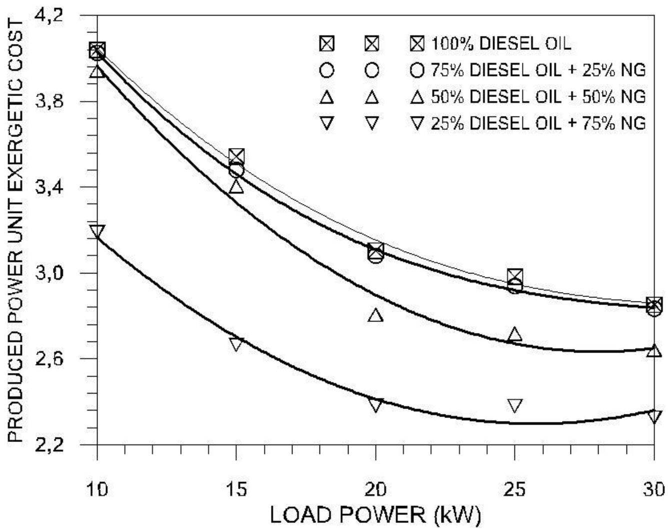

Figure 8 – Variation of produced power unit exergetic cost with engine load power and natural gas concentration in the fuel.

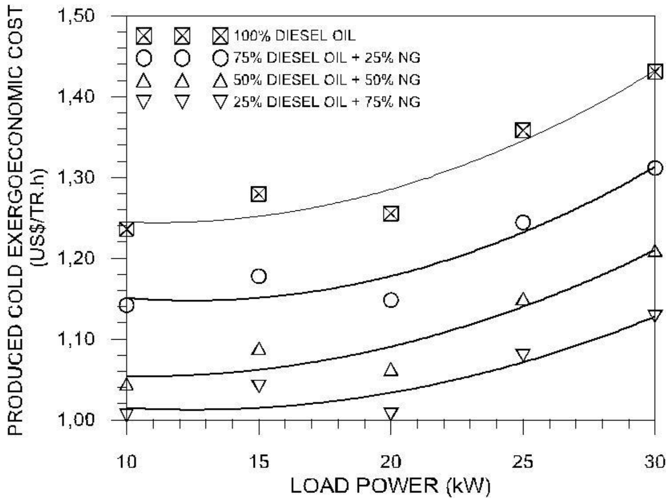

Figure 9 – Variation of produced cold exergoeconomic cost with engine load power and natural gas concentration in the fuel.

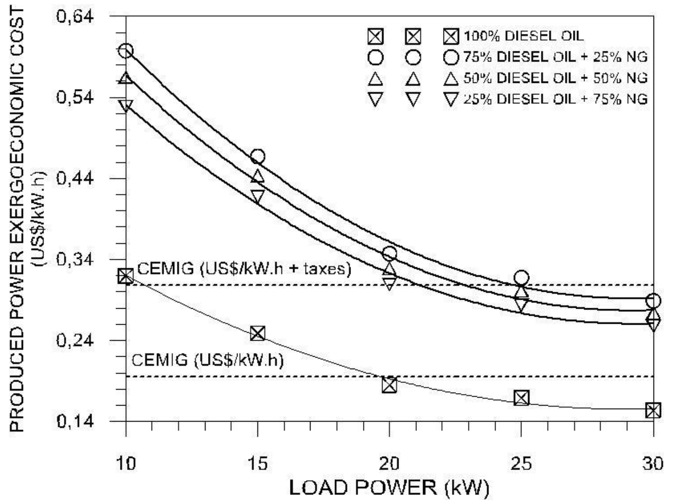

Figure 10 – Variation of produced power exergoeconomic cost with engine load power and natural gas concentration in the fuel.