SIMULATION OF THE COUPLED BETWEEN AN ABSORPTION REFRIGERATION SYSTEM (H_2O-NH_3) AND A TURBO-CHARGED DIESEL ENGINE

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Abstract. *This work presentes a coupling between an absorption refrigeration system (AAR) and a diesel internal combustion engine (ICE) equipped with turbocharger system. The thermodynamic modeling considers the energy and exergy analysis. Besides the evaluation of heat exchanged and mechanical work the heat exchanger area is calculated based on the temperature mean logarithmic and global heat transference coefficient. The heat source considered is the hot water from engine's block and the refrigerator effect produced by AAR é applied for cools the compressed air from turbocharger system. Further, the coefficient of performance (COP) and device's irreversibility is analyzed. The main conclusion of this study shows that, as heat source, the thermal energy from hot water is enough to cools the compressed air to conventional vehicle level. However, the vehicle implementation (commercial truck and passeger car) of this system (AAR + ICE) is not feasible because of devices's size is very larger especially of the fan (power) and the radiator's area, suggesting the application on the systems that have enough space like locomotives and boats.*

Keywords*: Internal Combustion Engine, Absorption Refrigeration, Waste Heat Source*

1. INTRODUCTION

Internal combustion engine (ICE) converts chemical fuel energy into useful mechanical energy. This conversion process in an ICE (gasoline or diesel) is very inefficiencies, Ford (2010). While most of the fuel energy is dissipated in energy conversion in the engine, significant amounts are lost in the vehicle systems. Typically, ICE dissipates 76% of the fuel energy in the heat form; 24% remains are applied to power the vehicle systems and drive the vehicle down the road, Ford (2010).

Besides the low thermal efficiency, another matter is related to emission of pollutants from ICE. The air pollution has provoked health problem to people, mainly in metropolitan areas, especially because the polluted air carry toxic substances that affect, among others, the human respiratory and cardiovascular system, Gouveia et. al. (2003).

After 100 years of evolution, the automobile industry is on the edge of revolution, responding for increase urgent need for new mobility solutions that decrease dramatically the environmental impacts. The evidence of this urgency includes the increase of:

- Global concerns about the environment;
- Volatility of energy cost;
- Negative perception of the automotive industry;
- Importance of fuel economy in new projects;
- Global regulatory pressures.

Therefore, deserves particular care, any proposed technology that comes to discuss the possibility of improving engine's thermal efficiency and pollutants emissions reduction.

Herold et. al. (1996) shows that when a heat source is available (waste heat), like found in ICE (exhaust system, hot water from engine's block and compressed air by turbo system), an AAR can be used to make cool (refrigerator effect). Besides the cools the compressed air, from turbo system, (proposal of this present work), the refrigerator effect produced by AAR could be used in passenger thermal comfort like, nowadays, is done by conventional air conditional system (AC). Thus, the engine's thermal efficiency could be increased becoming the waste heat from ICE in power thermal energy to AAR. Manzela et. al. (2010) shows still that the coupling between ICE and AAR resulted in decrease of the carbon monoxide, increase of hydrocarbons and insignificant changes of carbon dioxide.

Overall, the coupling between ICE and AAR can be considered an alternative technology for the engine's thermal efficiency improvements and pollutant emission reduction, promoting synergistic effects with the some improvements done by automotive industry like weight reduction, aerodynamic drag, rolling resistance tires and reducing vehicle parasitic losses.

2. ABSORPTION REFRIGERATION SYSTEM

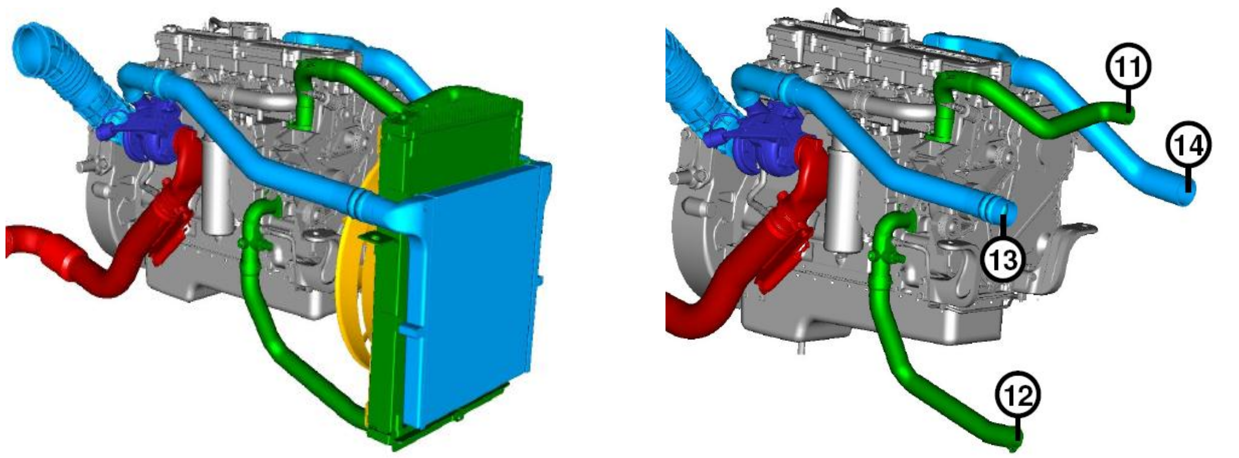
The refrigeration cycle based on absorption have experienced up and downs since its first application around in 1859, Bjurstrom & Raldow (1981). Predominantly in refrigeration and air-conditioning industry, these systems became no competitive when the advent of electric energy distribution at low cost between the end century 19^{th} and beginning century 20th. Newly, has seen a renewed interest in the absorption cycles in face to growing concerns about energy and environmental matters.

Similar to vapor compression cycle, the basic absorption cycle operates at two pressure levels correspondent to the condensation and evaporation temperatures, Niebergall (1981). In absorption cycle, the compressor is replaced by a binary solution, refrigerant (NH_3) + absorbent (H_2O), and three components: absorber, where the mixture absorbs the refrigerant, recirculation pumps where raises the solution's pressure and generator where occurs the refrigerant separation from solution through the addition of heat. Apart from these differences, the refrigerant has the same processes of condensation, expansion and evaporation in a vapor compression cycle. An absorption cycle (with regenerative heat exchanger) and main components are shown in figure 2.

Regarding the application of this system on vehicle project, Salviano (2011) shows a brief historic which covers from 1971 till 2010. However, the main studies about it were showed by Mostafavi & Agnew (1996a, b, c, d, 1997b, c, d) between the years of 1996 and 1997. These studies focus in the use of thermal energy from exhaust system (gas) and the refrigerator effect produced is used for replace the current vehicle air-conditioning, while the present work considers the hot water from engine's block like heat source and the refrigerator effect produced is used for cools the compressed air from turbo system.

3. PROPOSAL DESCRIPTION

Figure 1a shows a diesel engine coupled to the air intake system (AIS) and cooling package (fan + radiator + intercooler). Following, the figure 1b shows this same configuration uncoupled of these components.



(a) (b)

Figure 1. Schematic (a) turbo engine coupled to standard cooling package (b) turbo engine uncoupled.

Air from ambient is admitted through the pipe towards the turbo-compressor, device that increases the pressure and consequently the temperature. Thus, it is necessary to cools the air through a heat exchanger, commonly known as the intercooler (Figure 1a, blue way). After the combustion process, heat energy is released and absorbed by water that circulates inside the engine's block (jackets), therefore, it is necessary to cools the water before returning it to the engine's block to continue the cooling process the components; this is also done through a heat exchanger known as the radiator (figure 1a, green way). In order to improve the efficiency of the intercooler and radiator, a fan is attached to the heat exchangers, increasing the mass air flow through the tubes and fins, consequently, increasing the heat rejection (Figure 1a, yellow components). It is shown in figure 1b an ICE uncoupled of the fan, radiator and intercooler, where the point 11 provides hot water while there is air in point 13 and the points 12 and 14 have warm water and air cooled, respectively. The proposed cycle use the energy from hot water (point 11) and produces the refrigerator effect to cools the hot air from turbo system (point 13).

Briefly, the system cools the water that through the radiator, releasing the thermal energy to generator (absorption cycle). It refrigerator effect produced at evaporator cools the hot air that flow through of the intercooler. Thus, both desirable and necessary effects have been reached. In Figure 2, is present the AAR system with water and air inlet points, besides the AAR's components (generator, condenser, evaporator, absorber and regenerator).

4. THERMODYNAMIC MODELING

The most classic procedure to evaluate the thermal system performance is through of the 1ª Law of Thermodynamic, Horlock (1997). This analysis allows defining, from the energy view, the thermal efficiency of each equipment as well as the overall system. Although very popular, this methodology has limitations because it not counts the energy quality, e.g., not concerned with the inherent irreversibility of the processes. Considering this aspect, the thermal evaluation ought to consider the 1ª and 2ª Law of Thermodynamic coupled, known as Exergy Analysis, Kotas (1995).

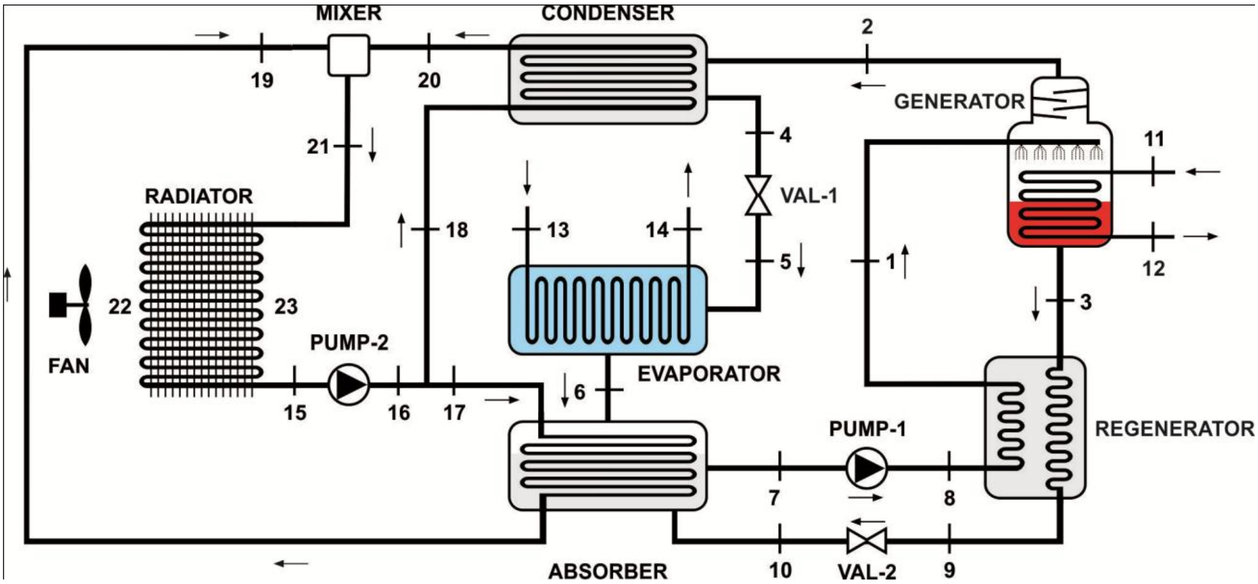


Figure 2. AAR basic schematic with regenerator coupled to external circuit (radiator, pump-2 and mixer).

Around of each device shown in Figure 2, is drawn one imaginary line representing a control volume, where the mass balance (continuity), 1ª of Thermodynamics and Exergy Balance, Wylen (1994), equations 1, 2 and 3, respectively, have been applied. The following considerations were observed:

I. The cycle operates under steady state.
II. Variations of kinetic and potential energy are negligible.

$$\sum \dot{m}_s - \sum \dot{m}_e = 0 \tag{1}$$

$$\dot{Q}_{V.C.} + \sum \dot{m}_e h_e = \sum \dot{m}_s h_s + \dot{W}_{V.C.} \tag{2}$$

$$\dot{I}_{V.C.} = \left(\sum \dot{m}_e \psi_e - \sum \dot{m}_s \psi_s \right) + \sum \left(1 - \frac{T_0}{T} \right) \dot{Q}_{V.C.} - \dot{W}_{V.C.} \tag{3}$$

The total exergy has been evaluated as follows:

$$\psi_i = \psi_i^{ch} + \psi_i^{ph} \tag{4}$$

Chemical exergy $\left(\psi_i^{ch}\right)$ for the mixture H_2O-NH_3 (points: 1, 2, 3, 4, 5, 6, 7, 8, 9, 10) has been calculated according to Rossa & Bazzo (2009), equation 5:

$$\psi_i^{ch} = \frac{x_{NH_3}}{M_{NH_3}} e_{o,NH_3}^{ch} + \frac{\left(1 - x_{NH_3}\right)}{M_{H_2O}} e_{o,H_2O}^{ch} \tag{5}$$

Where e_{o,NH_3}^{ch} and e_{o,H_2O}^{ch} is the standard chemical exergy of ammonia and water, respectively, as given by Szargut et. al. (1988). The standard chemical exergy unit is *kJ/kmol*. x_{NH_3}, M_{NH_3} and M_{H_2O} represents the mass fraction of ammonia, ammonia and water molecular mass, respectively.

The other points of the cycle shown in figure 2, only the physical exergy $\left(\psi_i^{ph}\right)$ have been considered, and calculated according to Szargut *et. al.* (1988), equation 6:

$$\psi_i^{ph} = (h_i - h_0) - T_0 (s_i - s_0) \tag{6}$$

Reference state of exergy analysis: ammonia at 30ºC and 1bar.

Cycle's coefficient of performance (COP) has been calculated as follow, equation 7:

$$COP = \frac{\dot{Q}_E}{\dot{Q}_G + \dot{W}_{BB1} + \dot{W}_{BB2} + \dot{W}_{FAN}} \tag{7}$$

4.1. Mixture's properties of H_2O-NH_3

The performance of an absorption refrigeration system is critically dependent on the chemical and thermodynamic properties of working fluids, Perez-Blanco (1984). A fundamental characteristic of the pair absorbent/refrigerant is that, in liquid phase, the pair is soluble within the cycle temperature range. The mixture should also be chemically stable, non-toxic and non-explosive. Further it follows:

- The difference between the fusion points of the pure refrigerant and mixture, at the same pressure, should be as large as possible.
- The refrigerant should have high vaporization heat and high concentration in the absorbent in order to maintain low rates recirculation between the generator and absorber per unit cooling capacity.

Several working fluids are suggested in the literature. Macriss (1988) suggests that there are at least 40 kinds of refrigerants and 200 kinds of absorbent for application in absorption refrigeration systems. However, the most common pairs are: ammonia-water and water-LiBr.

Since the discovery of absorption refrigeration system, the water-ammonia pair has been widely used. Both ammonia (refrigerant) and water (absorbent) are highly stable over a wide operating temperature and pressure range. The mixture can be used in application requiring low temperatures, because the freezing temperature of ammonia is close to -77°C. However, as ammonia and water are volatile, the cycle requires a rectifier between the generator and condenser to condense the evaporated water and return to generator. Without the rectifier, the water would accumulate in lower parts at evaporator harming the cycle's performance.

In this work, the mixture's properties H_2O-NH_3 have been calculated according to the correlations proposed by Ibrahim & Klein (1993) and implemented into the software EES® (Engineering Equation Solver) developed by Klein & Alvarado (1995). This software enables the rapid resolution of equation systems through the thermodynamic properties evaluation, such as: enthalpy and entropy, making it unnecessary access the thermodynamic tables.

4.2. AAR Pressures and Concentrations

The project assumptions define the pressure levels of the cycle. Niebergall (1981) suggests that the evaporation pressure (low pressure) should be defined by the evaporating temperature and condensing pressure (high pressure) by the condensation temperature. Thus, it is possible write:

$$p_{low} = p_{sat}\left(T_{EVAP}^{NH_3}\right) \tag{8}$$

$$p_{high} = p_{sat}\left(T_{COND}^{NH_3}\right) \tag{9}$$

Where: p_{sat} represent the saturated pressure.

Niebergall (1981) also suggests that the weak solution concentration should be defined by the pressure and temperature at generator while the rich solution concentration by pressure and temperature at absorber. Thus, the concentrations have been calculate according to simplified formulation proposed by Pátek & Klomfar (1995) where the mixture temperature is function of the pressure and molar fraction of ammonia in liquid phase, equation (10). Salviano (2011) provides detailed explanation about this methodology.

$$T(p,x) = T_o \sum_i a_i (1-x)^{m_i} \left[\ln\left(\frac{p_o}{p}\right) \right]^{n_i} \tag{10}$$

Where: $T_o = 100K$ and $p_o = 2MPa$. The parameters are shown in the table 1.

Table 1. Parameters of the equation (10) by Pátek & Klomfar (1995).

i	m_i	n_i	a_i	i	m_i	n_i	a_i
1	0	0	+0.322302 x 10^{1}	8	1	2	+0.106154 x 10^{-1}
2	0	1	-0.384206 x 10^{0}	9	2	3	-0.533589 x 10^{-3}
3	0	2	+0.460965 x 10^{-1}	10	4	0	+0.785041 x 10^{1}
4	0	3	-0.378945 x 10^{-2}	11	5	0	-0.115941 x 10^{2}
5	0	4	+0.135610 x 10^{-3}	12	5	1	-0.523150 x 10^{-1}
6	1	0	+0.487755 x 10^{0}	13	6	0	+0.489596 x 10^{1}
7	1	1	-0.120108 x 10^{0}	14	13	1	+0.421059 x 10^{-1}

4.3. Thermal area of the heat exchanger and power fan of the radiator

Considering the global heat balance on the heat exchanger (hot and cold fluid stream), Incropera & De Witt (2002) defines the equation (11) that can be used for exchanger's area determination, since known the global heat transference coefficient (U), heat rate $(\dot{Q})$ and temperature mean logarithmic (ΔT_{lm}) defined by the equation (12). The table 2 shows the results of the equations (11) and (12) applied on the each heat exchanger of the cycle (figure 2).

$$\dot{Q} = UA\Delta T_{lm} \tag{11}$$

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\dfrac{\Delta T_1}{\Delta T_2}\right)} \tag{12}$$

Table 2. Summary of the results according to equations (11) and (12) applied on the cycle (figure 2).

Device	ΔT_{lm}	$U\left(\dfrac{W}{m^2.^{o}C}\right)$
Generator	$\dfrac{(T_{11}-T_3)-(T_{12}-T_2)}{\ln\left(\dfrac{T_{11}-T_3}{T_{12}-T_2}\right)}$	600*
Condenser	$\dfrac{(T_2-T_{20})-(T_4-T_{18})}{\ln\left(\dfrac{T_2-T_{20}}{T_4-T_{18}}\right)}$	480*
Evaporator	$\dfrac{(T_{13}-T_6)-(T_{14}-T_5)}{\ln\left(\dfrac{T_{13}-T_6}{T_{14}-T_5}\right)}$	240*
Absorber	$\dfrac{(T_{17}-T_6)-(T_{10}-T_{19})}{\ln\left(\dfrac{T_{17}-T_6}{T_{10}-T_{19}}\right)}$	480*
Regenerator	$\dfrac{(T_9-T_8)-(T_3-T_1)}{\ln\left(\dfrac{T_9-T_8}{T_3-T_1}\right)}$	480*
Radiator	$\dfrac{(T_{21}-T_{23})-(T_{15}-T_{22})}{\ln\left(\dfrac{T_{21}-T_{23}}{T_{15}-T_{22}}\right)}$	194**

***** Pratts (1997), ****** Salviano (2011)

The power fan and the mass air flow of the radiator, between the points 22 and 23, is determined in function of the rotation speed, regarding the "Law of Fan", like shows by the follow equations:

$$\dot{m}_{air} = \dot{m}_{base}\left(\frac{n}{n_{base}}\right) \tag{13}$$

$$\dot{W}_{FAN} = \dot{W}_{base}\left(\frac{n}{n_{base}}\right)^3 \tag{14}$$

4.4. Vehicle test

The data shown in table 3 were measured during validation test on the commercial truck. The measured points indicated are according to the figure 1b. The data are referent the test condition known as *Maximum Torque*, where the vehicle is fully load and is accelerated around of 22 km/h. This condition is considered too critical of the thermal view point.

Table 3. Input data from validation test on the commercial truck considering maximum torque condition.

Point	Fluid	$T(°C)$	$p(bar)$	$\dot{m}\left(\frac{kg}{s}\right)$	$\dot{Q}(kW)$
11	Water	92	0.83	2.865	84.32
12	Water	85	0.62	$\dot{m}_{12} = \dot{m}_{11}$	
13	Air	162	1.24	0.251	27.96
14	Air	52	1.17	$\dot{m}_{14} = \dot{m}_{13}$	

The *base* data below, equation (13) and (14) were obtained from 3D numerical simulation through of commercial software UH3D® (Under Hood Three Dimension), kindly provided by the big automotive industry from São Paulo State (Brazil). These results also consider the maximum torque condition.

$$n_{base} = 1300\,RPM \tag{15}$$

$$\dot{m}_{base} = 3.37\frac{kg}{s} \tag{16}$$

$$\dot{W}_{base} = 3.518kW \tag{17}$$

5. RESULTS AND CONCLUSIONS

Initially, the main question that needs to be answered is: *"The thermal energy from hot water leaving the engine's block (point 11) is sufficient to power an absorption refrigeration unit (H_2O-NH_3), and produce the enough refrigerator effect to cools the hot air from turbocharger system to the temperature levels found in the conventional vehicle system?"*

Looking at it, firstly the AAR must meet the difference of temperature between the points 11 and 12, i.e., there is a minimum amount of thermal energy that must be supplied to the generator. Whereas that supplied energy, the AAR should be able to produce the refrigerator effect, at evaporator, at least to maintain the difference of temperature between the points 13 and 14. Namely, the simulation strategy of the coupled system is to fix the points 11 and 13 and after adjust the various parameters of the cycle like evaporator and condensation temperature, mass flow of rich solution, ensuring that temperatures at the points 12 and 14 are according to table 3.

Important remark is relation about boundary conditions assumed in thermodynamic modeling, as the saturated steam condition at the generator and evaporator outlet, point 2 and 6, respectively, besides the saturated liquid condition at the condenser and absorber inlet, point 4 and 7, respectively. Another boundary condition assumed considers that the mass flow of "pure" ammonia leaving the generator (point 2) is in thermal equilibrium with the rich solution entering at the generator (point 1), i.e., $T_2 = T_1$. It condition allows to vaporize less steam water together the ammonia, and the simulation results, table 4, shows it, the concentration of ammonia at the generator outlet (point 2) is 99.88%. This result could lead to the decision to eliminate a device known by *rectifier*. It device is commonly added to the AAR when the pair refrigerant-absorbent used is water-ammonia. The purpose of the rectifier is to condense the steam water that vaporized together ammonia returning it to the generator.

The coefficient of performance (COP) is 0.271, according to equation 7. It COP is relatively low if compared to COP of the refrigeration vapor compression, which typically has exceed the value of 1.0. However, COP of 0.27 is acceptable and very common for AAR, Niebergall (1981).

Table 4. Results of the thermodynamic modeling considering the coupled of the ICE and AAR (figure 2).

Point	$\dot{m}\left(\frac{kg}{s}\right)$	$T\left({}^{\circ}C\right)$	$p\,(bar)$	x_{NH_3}	x	$h\left(\frac{kJ}{kg}\right)$	$s\left(\frac{kJ}{kg.K}\right)$	$\psi_{ph}\left(\frac{kJ}{kg}\right)$	$\psi_{ch}\left(\frac{kJ}{kg}\right)$
1	0.415	60.0	20.33	0.4996	-	31.07	0.7155	270.9	9938
2	0.0266	60.0	20.33	0.9988	1	1329	4.173	520.5	19817
3	0.3884	87.9	20.33	O.4654	-	159.8	1.088	286.5	9262
4	0.0266	49.9	20.33	0.9988	0	240.1	0.8156	449.5	19817
5	0.0266	8.0	5.74	0.9988	0.1644	240.1	0.8683	433.5	19817
6	0.0266	29.6	5.74	0.9988	1	1332	4.725	356.8	19817
7	0.415	44.3	5.74	0.4996	0	-40.74	0.5004	264.3	9938
8	0.415	44.6	20.33	0.4996	-	-38.34	0.5022	266.1	9938
9	0.3884	71.9	20.33	O.4654	-	85.59	0.8783	276.0	9262
10	0.3884	55.0	5.74	O.4654	0.0541	85.59	0.8899	272.5	9262
11	2.865	92.0	0.83	-	-	385.4	1.216	473.5	-
12	2.865	84.9	0.62	-	-	355.9	1.134	468.8	-
13	0.251	162.0	1.24	-	-	437	7.185	-1284	-
14	0.251	47.6	1.17	-	-	321.2	6.893	-1312	-
15	4.000	45.0	1.00	-	-	188.5	0.6385	451.6	-
16	4.000	45.0	3.00	-	-	188.8	0.6387	451.8	-
17	2.600	45.0	3.00	-	-	188.8	0.6387	451.8	-
18	1.400	45.0	3.00	-	-	188.8	0.6387	451.8	-
19	2.600	52.9	2.25	-	-	221.7	0.7411	453.7	-
20	1.400	49.9	2.25	-	-	209.5	0.7035	452.9	-
21	4.000	51.9	1.00	-	-	217.4	0.7284	453.3	-
22	6.092	30.0	1.00	-	-	303.5	6.881	-1326.0	-
23	6.092	48.8	1.00	-	-	322.5	6.942	-1325.0	-

From table 4, the first result to be confronted is the difference of temperature between the points 11 and 12 in comparison with results shown in table 3. Of easy verification, the difference of temperature of 7°C between the points 11 and 12 (table 4) is the same shown in the table 3. Met this minimum condition, the points 13 and 14 should be analyzed. The temperature at the point 14 (47.6ºC, table 4) is less than of conventional system (52°C, table 3). This is an excellent result because the lower the temperature at this point, the greater the air-density and, consequently, the greater the amount of mass of air that can be included at constant volume (volume between the engine's block and piston). Therefore, the question done earlier in this section could be answered as follows: *"Yes. The thermal energy from hot water leaving engine's block is sufficient to power an AAR and produce refrigerator effect enough to cools the compressed air from the turbocharger system to temperature levels found in a conventional vehicle system of ICE turbo."*

It is noteworthy that the results above were obtained considering the combination of several parameters, among the main are: evaporation temperature (8°C, point 5), condensation temperature (50°C, point 4), cooling temperature (45ºC, point 15) difference of temperature between the regenerator inlet and outlet (16°C, T_3-T_9) and mass flow rate of rich solution (0.415 kg/s, point 1).

The difference between the concentration of ammonia in rich and poor solution was 3.419%. This result is lower than suggested by some work, around of 10%, Marques (2005).

Another interesting result is the quality of 16.44% (point 5) after valve 1. This is not a good result because about 1/6 of ammonia has already been evaporated before it enters the evaporator, which reduces the overall refrigerator capacity of the AAR. Salviano (2011) shows that increasing the evaporation temperature decrease the quality (at point 5) whereas increasing the condensation temperature the quality (at point 5) increase, in both case about the same amplitude.

The table 5 shows the added/rejected heat by the devices, as well as the irreversibility generated and the pumps and fan mechanical work.

Table 5. Irreversibility, heat and work of the device of the cycle (figure 2).

Device	$\dot{I}_{VC}\,(kW)$	$\dot{Q}_{VC}\,(kW)$	$\dot{W}_{VC}\,(kW)$
Generator	0.9869	84.51	-
Condenser	0.4221	28.97	-
Val-1	0.4254	-	-
Evaporator	8.876	29.06	-
Absorber	0.8318	85.59	-
Pump-1	0.2255	-	0.9955
Val-2	1.362	-	-
Regenerator	2.103	28.81	-
Pump-2	0.2566	-	1.077
Mixer	0.5192	-	-
Radiator/Fan	23.96	115.6	20.78

Comparing the results shown in table 3 and 5, it is possible verify that the thermal energy dissipated by conventional vehicle system is equal to dissipated by AAR, which ensures the minimum temperature difference of 7°C. However, the refrigerator effect produced by AAR (29.06 kW) is 4% higher than needed by vehicle application (27.96 kW). As already understood, the AAR is commonly known to be primarily powered by thermal sources, unlike the system based on vapor compression (conventional air-conditional, for example) which high energy is necessary for power the compressor through of electric motor. The evidence of this affirmation can be verified by the results shown in table 5, where the pump mechanical work is lower than thermal energy, for example, generator and condenser.

The irreversibility rate generated by the devices, according to table 5, is visually better shown in the figure 3.

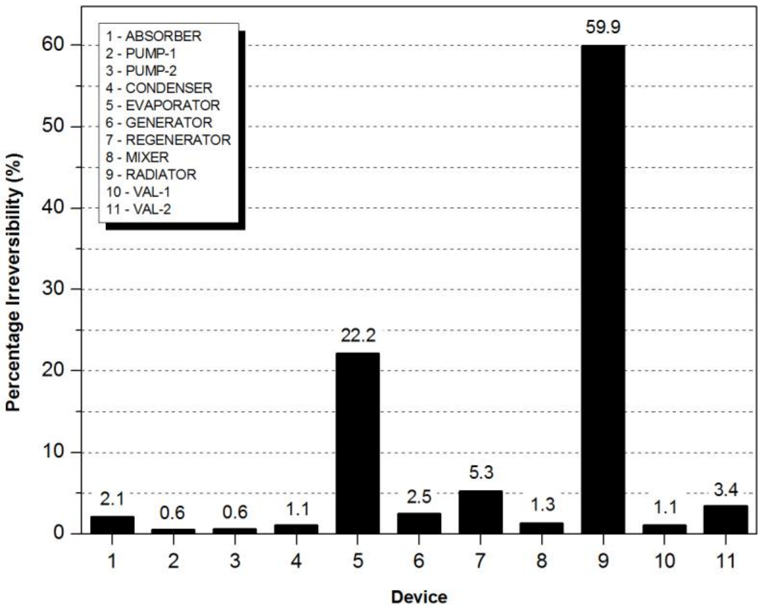


Figure 3. Percentage irreversibility of each device of the cycle according to results from table 5.

Evidently, it is possible observed that the radiator is responsible for more than half the AAR total irreversibility, about 59.9%, mainly due to necessary power mechanical work of the fan, about 20.78 kW, according to equation 7. Further, the evaporator had about 22.2% of the AAR total irreversibility. Evaluating the evaporator irreversibility, according to equation 7 and table 4, despite of much larger than physical exergy, the chemical exergy at point 5 and 6 are almost equivalent not contributing significantly on the evaporator irreversibility balance. Thus, the main cause of the evaporator's high irreversibility is primarily due to cooling of the air from 162°C to 47.6°C, and further due to ammonia evaporation between the points 5 and 6, of course, the mass flows at each point of the evaporator (inlets and outlets) had been considered. The other devices showed low percentage of irreversibility, not exceeding 5.3% as identified on the regenerator.

The figure 4 together the data from table 5 allows to analyze the added/rejected heat and heat exchanger area of devices.

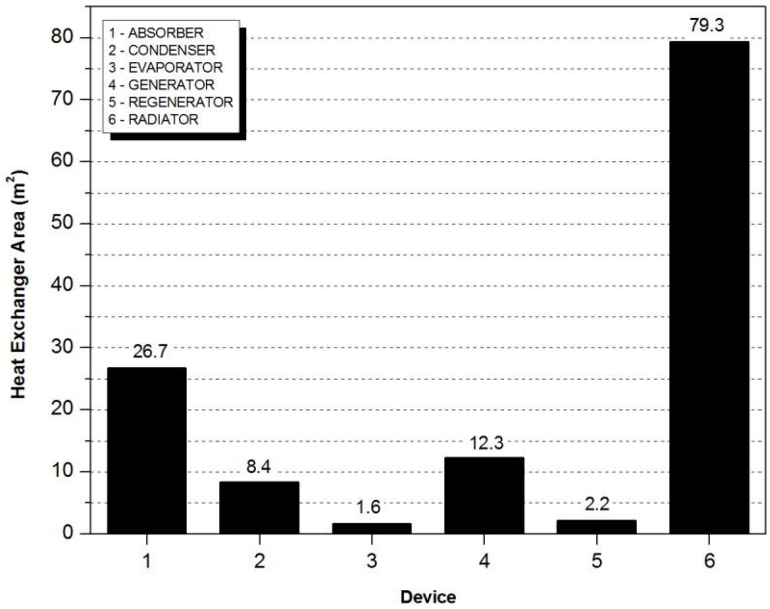


Figure 4. Heat exchanger area of devices according to equation 12 and table 2.

Evidently, the thermal energy dissipated by the radiator is approximately the sum of heat added to generator and evaporator, about 115.6 kW. The radiator's area is about 4 times larger than conventional vehicle configuration because of the difference of temperature between the points 21 and 22 is lower than the difference of temperature between the points 11 and 22, i.e., 79.3 m^2 (radiator from figure 2) against 22.17 m^2 (radiator from figure 1a). The fan rotation has also changed, from 1300 RPM (conventional configuration) to 2350 RPM (AAR plus ICE). This adjustment was needed to ensure that the temperature at point 21 is, minimally, larger than at point 23, which resulted in an increase significant on the fan power mechanical work, from about 4 kW (conventional configuration) to 20.78 kW (AAR plus ICE), 5 times higher. These last results and results from figure 4 leads to the preliminary conclusion that for automotive applications, the coupling between AAR and ICE is not feasible because the device's size, especially for radiator and fan specification. This conclusion is based on the physical space observed in vehicle's underhood that is too tight, Salviano (2011), especially in commercial trucks and passenger cars.

6. ACKNOWLEDGEMENTS

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