ECONOMIC VIABILITY OF AN ABSORPTION CYCLE FOR AUTOMOTIVE AIR CONDITIONING EMPLOYNG SOLAR HEATING

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Abstract. *Automotive air conditioning commonly employs vapor compression cycles, where the compressor extracts its power from the vehicle engine. Although the actual devices had reached high efficiency, such procedure always implies in lost of power and major fuel consume. An alternative is the use of absorption cycles, which demands minor amounts of mechanical power for working, using heat instead. This work analyses the economic viability of the use of an absorption cycle heated by solar energy. The work fluid is the combined water-ammonia solution. A solar collector installed in the vehicle supplies the heat. The advantages are the energy saving and the reducing of emissions, through the use a cheap and renewable energy.*

Keywords: *Absortion cycle, Automotive air conditioning, Solar heating.*

1.Introduction

Automotive air conditioning has been based in the vapor compression cycle. Despite of decades dedicated to its improvement, such systems continue presenting difficulty in reducing energy consumption. Since it needs mechanical energy to work, such energy has to be extracted from the engine power supply. As an example, Kaynakli *et al.* (2003) mentioned a compressor that operates with a 1.6 kW power at 3000 rpm, with environment temperature of 16° C, input air temperature at evaporator of 26° C, refrigerant mass flux of 0.030 kg/s and condenser temperature of 41° C.

As a consequence, the fuel consumption increases, what affects the mechanical efficiency, and yields a reduction in the power available for moving the vehicle. In the absence of a control system to manager the mechanical power during critical situations, like an over passing, this effect may be dangerous for the occupants.

Since has been difficult to improve the system working upon the cost reduction and energetic optimization, an option may be the use of an absorption cycle, employing solar energy as heat source.

In the vapor compression cycle mechanical work is necessary to increase pressure by a compressor. In the absorption cycle, such a pressure rising is obtained by thermal energy. Actually, a little work is necessary, in order to operate the pump present in this cycle, but the amount of work for a specific refrigeration capacity is minimal, whem compared to the work used in a compressor for the same thermal load.

The heat necessary for the absorption cycle can extracted from diverse sources. The use of solar energy is an excellent alternative compared to conventional energy sources, since it is an infinite and non-pollutant source, available all over the world. Its use has become attractive, as the costs have been reduced. The continuous raising of the price of fossil fuels also justify its use.

In this work, the vapor compression cycle employing refrigerant R134a, usually used in this kind of application, will be replaced by an absorption cycle employing water and ammonia, being the water the absorbent and ammonia the refrigerant. The heat source will be a flat solar collector. The main objective of investigation is to study economic viability of such replacement, through the comparison with a vapor compression cycle working under the same conditions. It should be pointed out that, although its toxicity, ammonia presents good technological properties for such application, what justify the choice for this refrigeration fluid.

Edward Naime invented the absorption cycle in 1777 (Cortez *et al.*, 1994). The Water-LiBr based cycles appeared in 1945, but its massive use was limited to the 50's, when it began to be replaced by the vapor compression cycle (Silva, 1994).

Now a day, diverse developments have been carried on. Dince *et al.* (1996) studied the performance of absorption cycles feed by solar energy using R-22 as refrigerant and DMETEG (Dimetil Ether Tetraethilene Glycol) as absorbent. Bulgam (1997) studied the use of low temperature heat sources (85^o C to 110^o C) in water-ammonia systems, comparing the results with the literature and obtaining an optimum operation condition, with better results in thermal processes. Li *et al.* (2000) presented a literature review for air conditioning system feed by solar energy, using the water-LiBr pair and reporting the efforts for improving its performance. Furthermore, authors showed that the input temperature of vapor generator is the most important parameter for the design and manufacturing of solar energy feed units. Other factors to take in account are solar collector choice and the design and shape of system. Srikhirin *et al.* (2001) described several systems and its status of implementation. Seara *et al.* (2001) studied the control system for the optimum generator temperature in absorption cycles of single effect, using water-ammonia, emphasizing its influence upon the coefficient of performance. Illane *et al.* (2002) presented a prototype of a water-ammonia absorption cycle, moved by solar energy, and expected to be useful in the countryside. The generator and heat exchanger have multipipe components for heat transfer among the fluids, and finned pipes formed the rest of the device, in order to improve the heat transfer to the environment. Solar energy was collected through a cylindrical parabolic solar collector that reached temperatures over 150^o C, high enough to operate a water-ammonia system. Despite of this, the system was considered unsatisfactory, since it has presented a low coefficient of performance.

2. Proposed absorption cycle

Figure 1 shows a schematic of the absorption cycle. Its components are: a solar absorption plate that acts as a heat source, a separator for water droplets and ammonia, a condenser, an expansion valve, an evaporator to extract heat from the cabin, an absorber device to receive the water from the dryer pipe and mix it with the weak solution of ammonia from the evaporator, and a pump to rise the refrigerant pressure until the solar absorption plate.

The heat exchange in these components occurs in the following sequence: Heat from the high temperature source (Q_H') comes in the generator, while the heat from a low temperature heat source (Q_L) comes in the evaporator. The heat release in the cycle occurs in the absorber (Q_L') and condenser (Q_H) at temperature that allow these rejections to the atmosphere. The major operational cost of the system is in the heat addicted to the generator (Q_H'). The other energy costs are relatively small when compared to that.

In the proposed system, the pipeline where ammonia flows is made with aluminum, due its light weight, mechanical strength and resistance to ammonia corrosion (since its surface is anodized). After tests in heavy truck with aerodynamic cover, it was verified that the most convenient place for the solar plate of 3 m2 would be above the aerodynamic cover. The purpose of the solar plate is to warm the water-ammonia solution to 110^o C, in order to extract the heat from the cabin internal environment by the evaporator, to maintain the thermal comfort. Among the diverse solar plates available for this application, the flat one with selective surface was chosen. It has an energy concentrator and a glass cover, in order to be able to absorb all the light wavelengths, improving the plate efficiency.

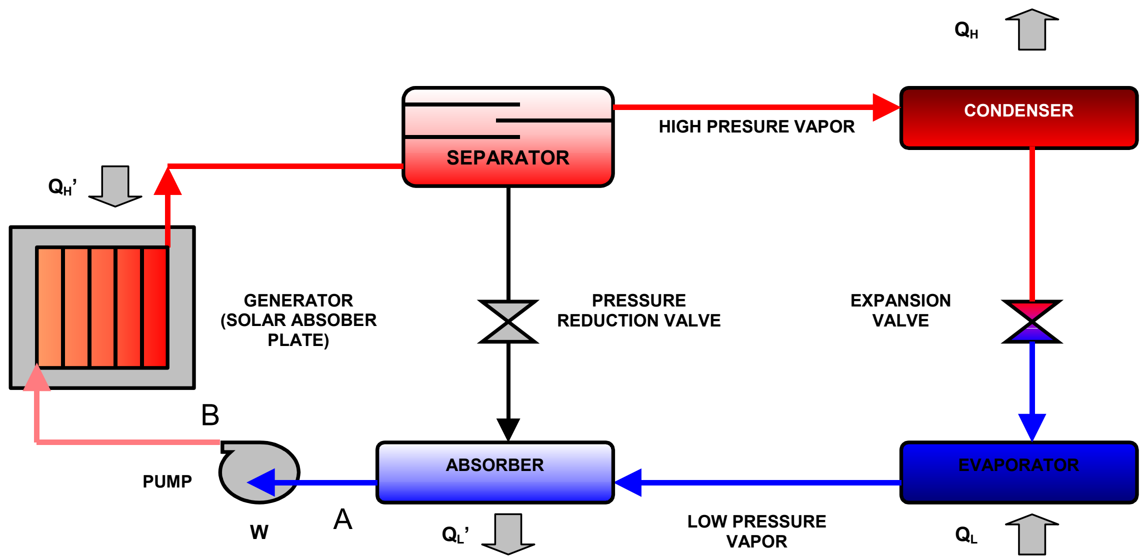

Figure 1. Absorption cycle.

3. Dimensioning

The system dimensioning was done with parameters extracted from a compression cycle already available. The main objective was replace to compressor, with 1.6 kW power consumption, to the solar absorber plate. It was also necessary to change the working fluid for ammonia, resulting in the addiction of new components from the system. The system was adapted for the original range of work of the vehicle. In order to compare the cycles, data extracted from a standard efficiency test for the vapor compression cycle in a heavy truck (57 tons and 370 CV, with tall roof and bed within the cabin) were employed. Measurements are showed in the Fig. 2. Temperature was measured with type K thermocouples, and transdutors were used for pressure measurements. The standard measurement procedure is described as follows:

1. Environment temperature has to be above 25^o C and the vehicle cabin has a hermetic closure.
2. The steady state has to be reached with the vehicle stopped, with engine working. The air recirculation is switched on and the front panel ventilation has to be in speed 3. This condition is to be kept for 15 minutes.
3. Vehicle at 23 km/h, beginning of the green stripe (optimum rotation level). The air recirculation is switched on and the front panel ventilation has to be in speed 3. This condition is to be kept for 15 minutes.
4. Vehicle at 96 km/h, in the last gear. The air recirculation is switched on and the front panel ventilation has to be in speed 1. This condition is to be kept for 5 minutes.
5. Vehicle stopped with engine working. The air recirculation is switched on and the front panel ventilation has to be in speed 1. This condition is to be kept for 5 minutes.

In these conditions, it is desirable that the vehicle air conditioning shall be able to keep a thermal comfort condition in a range of 20^o to 24^o C.

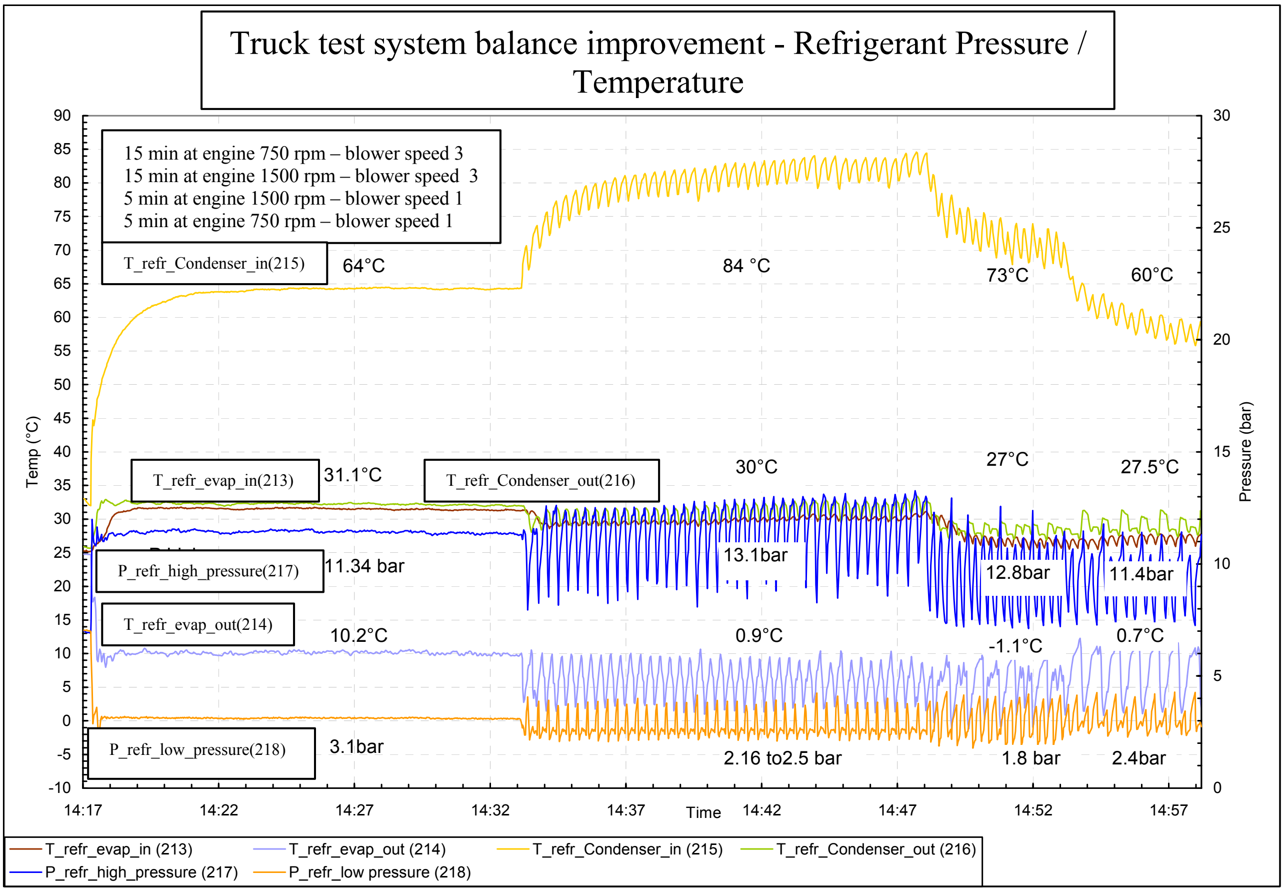

Figure 2. Data acquisition for the vapor compression system.

The meanings of the lines in Fig. 2 are:

Line “T_refr_Condenser_in(215)” : data measured in the condenser entrance.
Line “T_refr_evap_in(213)” : data measured in the evaporator entrance.
Line “T_refr_Condenser_out(216)” : data measured in the condenser outlet.
Line “T_refr_evap_out(214)” : data measured in the evaporator outlet.
Line “P_refr_ high_pressure(217)” : highpressure line.
Line “P_refr_ low_pressure(218)” : low pressure line.

After compiling these data, arithmetic averages for each curve were inserted in a pressure x enthalpy diagram for refrigerant R134a, Fig. 3, where the properties were extracted from. Table 1 shows the estimatives for heat transfer in condenser and evaporator, and the work in the compressor, considering steady state operation, applying the mass and energy conservation:

$$\dot{m}_e = \dot{m}_s \tag{1}$$

$$q + h_e = h_s + w \tag{2}$$

where $\dot{m}$ is the mass flux, q is the heat flux per mass unity, h is the enthalpy and w is the specific work. Subscripts *e* and *s* refers to entrance and outlet conditions in the equipments, respectively.

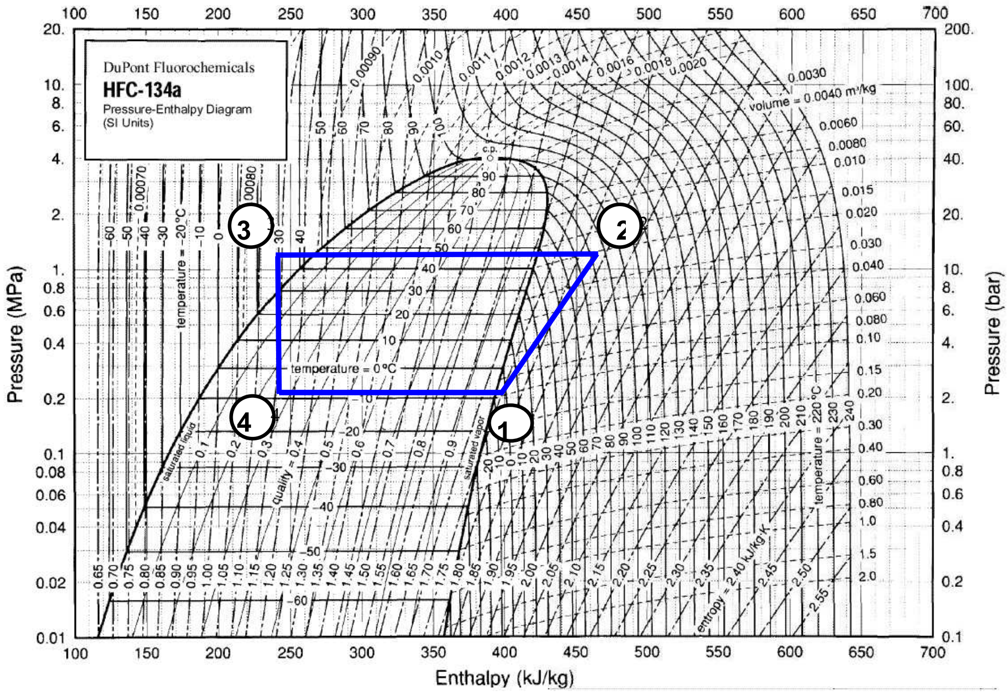

Figure 3. Pressure x enthalpy for R134a (Wang, 2000).

Table 1. Compression cycle quantities.

Point	h (kJ/kg)	q (kJ/kg)	w (kJ/kg)
1	400.0	$q_{41} = 160.0$	$w_{12} = 75.0$
2	475.0		
3	240.0	$q_{23} = -235.0$	
4	240.0		

The mass flux estimative was done through the reduction rate between the engine and the compressor, 1.44, using and average rotation for the engine. It was considered a Diesel engine, running in a specific part of the *Presidente Dutra* Highway (between *Resende* and *Serra das Araras*), resulting in 1700 rpm. Volumetric efficiency of the compressor was 63 % and displacement rate of compressor was 152.8 cm^2/rotation. With the volumetric efficiency and displacement rate at the compressor entrance, a value of 96.3 cm^3/rotation is found at that point. With a specific mass of 0.00528 g/cm^3 of R132a as saturated vapor, the mass flux is obtained from Eq. (3), and is equal to 0.0021kg/s.

$$\dot{m}_c = \rho . V_v . R_c \tag{3}$$

were ρ is the specific mass, V_v is the volumetric flux and R_c is the compressor rotation.

Once mass flux is known, the total amount of heat or work can be expressed in total units: q_{41}, q_{23} and w_{12} are 3.4 kW, - 4.9 kW e 1.6 kW respectively. The high and low pressures will be the same as in the compression cycle, for dimensioning the absorption cycle. All the new values needed were extracted from literature, for pure ammonia. These values are showed in Tabs. 2,3, where *s* is the entropy.

Table 2. Saturation data for ammonia at a pressure of 216 kPa.

Point	h (kJ/kg)	s (kJ/kgK)
1	1441.0	5.9
3	121.4	
4	121.4	

Table 3. Saturation data for ammonia at a pressure of 1310 kPa.

Point	Temperature (K)	h (kJ/kg)	s (kJ/kgK)
1			5.9
2	383.15	1700.0	5.9
3		121.4	
4		121.4	

After an energy balance, the ammonia mass flux is estimated as 0.0037 kg/s. At the absorber entrance, the ammonia leaving the evaporator mixes with the solution coming from the generator. This solution was assumed to be 50 % water and 50 % ammonia (Santos, 2005). After leaving the absorber (point A), the solution crosses the pump (point B), where its pressure rises until 1310 kPa, and arrives the generator entrance, where the separation occurs. The solution properties were extracted from Wang (2000). The Selected values are resumed in Tab.4.

Table 4. Properties of ammonia at pressure of 1310 kPa.

Place	Temperature (K)	h (kJ/kg)
Ponto 2		928.0
Ponto B	347.0	87.2

Since the proportion among water and ammonia in the solution is 1 to 1, the solution mass flux is considered to be twice water or ammonia mass flux. Using data from Tab.4 and energy conservation, the necessary generator heat supply ($Q_H^{'}$) was estimated to be 6.5 kW.

The solar flat plate selected to work as the generator is assumed to have a global heat transfer coefficient of 8 W/ m^2K. This value was the minimum in the range provided by the manufacturer. The surface available for its installation upon the vehicle cabin is 3 m^2. Concerned to this surface extension, it was verified that such installation would not prejudice the cabin structure. The temperature difference between entrance and outlet of the generator was supposed to reach 36.15 K, and the temperature in the separator outlet was assumed to be the same as the generator outlet. The amount of heat supply available for the cycle from the solar flat plate is given by:

$$Q_H^{'} = U.A.\Delta T \tag{4}$$

resulting in 0.9 kW. This is not enough to attend the energy demand of the cycle. As a possible solution, a heat exchanger can be added to the cycle, in order to supply its demand. The additional amount of heat could be extracted straight from the engine, through the refrigeration water and/or exhausting gases.

4. Economic analysis

Considering the results for dimensioning the system, the economic analysis was done over a cycle with a solar plate and a complementary heat exchanger, which in theory should be able to supply the necessary heat for warming. An analysis was also carried on considering only the presence of a heat exchanger. Both analyses should involve not only economic considerations, but also environment effects. However, since it is difficult to quantify such evaluation, the analysis will be restricted to the financial aspects, i. e., the money saved in a truck that runs 12000 km/month during 10 years, with the air conditioning always turned on.

The values used in this work were extracted from the market. Consumption was estimated from measurement data, and the oil costs were extracted from http://www.anp.gov.br/petro/levantamento_precos.asp (ANP, 2007).

4.1. Net Actual Value (*Valor Presente Líquido* - VPL)

VPL is a deterministic method that consists in evaluate at present time the initial investment and the money saved for implementation when compared to the *Minimum Rate of Attractiveness* (*Taxa Mínima de Atratividade* - TMA), which corresponds to the minimum financial return accepted by the investor (the interest rate, for example). The VPL is the value obtained from a cashier flux when brought to the present time, taking into account the TMA. The VPL is calculated from (Buarque, 1989):

$$VPL = \sum FC(1+i)^{-n} \tag{5}$$

where
:
FC = cashier flux, R$;
i = interest rate, %;
n = number of months

A typical criterion for decision is that VPL should be bigger than zero.

4.2. Rate of Internal Return (*Taxa Interna de Retorno* - TIR)

As VPL, it is considered an exact method. TIR is the interest rate that will produce a VPL equal to zero (Buarque, 1989)

$$TIR = \sum FC(1+i)^{-n} = 0 \tag{6}$$

The final investment decision is taken after comparing *TIR* and *TMA*. In order to assure the investment, TIR should be greater than TMA.

4.3. Payback (*PB*)

PB will be the necessary time to recovery the investment (Buarque, 1989):

$$PB = \frac{I_0}{\sum FC_{year}} \tag{7}$$

where:

I_0 = Cost of initial investment, R$;
FC = cashier flux, R$;

5. Economic viability

Some hypotheses were assumed, in order to complete the inputs for the financial analysis. A period of 10 years will be considered for depreciation of equipment (Motta, 2002). TMA will be 14.25 % / year, which corresponds to the Federal interest rate (SELIC tax) at 31 august, 2006. Initial investment will be applied in equipament's buying and installation. Values selected are shown in Tab.5.

Table 5. Average values of equipments and amortization time.

	Investment with the absorber plate (R$)	**Investment without the absorber plate (R$)**
Compressor	-1,900.00	-1,900.00
Absorber plate	2,000.00	
Pump	2,500.00	2,500.00
Heat exchanger	4,000.00	4,000.00
Absorber	2,000.00	2,000.00
TOTAL	**8,600.00**	**6,600.00**
Amortization time	**13 months**	**10 months**

According to ANP, the average price for diesel was R$ 1.85/l. The average vehicle consumption, considering a running of 12000 km/month and a full time use of the air conditioning is R$ 13,374.61/month (with air conditioning working all year long) and R$ 12,705.88/month (with air conditioning working 8 months a year).

Installation cost is considered to be part of vehicle cost. Maintenance costs are variable, according to the vehicle utilization. Since this cost is not attached to an equipment replacement, it is not measurable.

The cumulative flux will be the difference between the VPL of the vapor compression system and the VPL of the absorption cycle. After the calculation, the rate of investment return, the liquid present value and the payback will be available. Tables 6 and 7 resume the financial analysis for the two cases along 10 year.

Table 6. Results of *VPL* and *TIR* for the system proposed, with solar plate and heat exchanger.

Initial Investment = R$ 8,600.00												
Years		0	1	2	3	4	5	6	7	8	9	10
VPL	*FC*	-8,600.00	6,352.90	6,352.94	6,352.94	6,352.94	6,352.94	6,352.94	6,352.94	6,352.94	6,352.94	6,352.94
	FCD	-8,600.00	5,560.60	4,867.01	4,259.96	3,728.63	3,263.57	2,856.52	2,500.24	2,188.39	1,915.44	1,676.53
	FCA	-8,600.00	5,560.60	10,427.56	14,687.50	18,416.20	21,679.70	24,536.30	27,036.50	29,224.90	31,140.30	32,816.90
TIR = 51%												

Table 7. Results of *VPL* and *TIR* for the system proposed, with heat exchanger only.

Initial Investment = R$ 6,600.00												
Years		0	1	2	3	4	5	6	7	8	9	10
VPL	*FC*	-6,600.00	6,352.94	6,352.94	6,352.94	6,352.94	6,352.94	6,352.94	6,352.94	6,352.94	6,352.94	6,352.94
	FCD	-6,600.00	5,560.56	4,867.01	4,259.96	3,728.63	3,263.57	2,856.52	2,500.24	2,188.39	1,915.44	1,676.53
	FCA	-6,600.00	5,560.56	10,427.60	14,687.53	18,416.20	21,679.70	24,536.30	27,036.50	29,224.90	31,140.30	32,816.90
TIR = 71%												

6. Conclusion

After the viability study, we conclude that the use of the solar plate is not economically justified. The solar plate feeds the system with just 0.9 kW, considering a surface of 3 m^2, while the system needs 6.5 kW to assure the internal conditions prescribed for thermal comfort. However, with the use of a heat exchanger that extracts heat straight from the engine, it is possible to obtain the desired power supply, since the refrigeration water and the exhaust gases reach 90^o C and 570^o C, respectively. With this new configuration, the payback time will be 13 months, with TIR of 53 %. If only the heat exchanger is used, the payback time will be 10 months, with TIR of 71 %. This second case seems to be the best option, considering technical and economic viability.

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