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# THERMAL AND COST ASSESSMENT OF AN ABSORPTION REFRIGERATION SYSTEM

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***Abstract.*** *The objective of this work is thermal and cost assessment of a micro-capacity absorption refrigeration system. The simple effect absorption refrigeration system we modeled in EES (Engineering Equation Solver), in order to recovery the exhaust energy provided by the exhaust gases of an internal combustion engine. The exergy analysis of the cycle components we performed considering the heat transfer modelling. The cost of the produced cold we computed. A cold of up to 17.9 kW was produced at the maximum engine load of 37.5 kW at a unit cost of 0.058 US$/kW. The minimum engine load was required for the operation of the 22.5 kW cycle, the cold produced was 9.7 kW at a unit cost of 0.164 US$/kW.*

***Keywords:*** *Absorption refrigeration, $H_2O$-LiBr, COP, Exhaust gas energy, Exergy.*

## 1 INTRODUCTION

The current Brazilian scenario has a concern with the emission of polluting gases being one of the villains the combustion engines. According to the MINISTÉRIO DE MINAS E ENERGIA (2014) the transport sector was responsible for 215.3 million tons of carbon dioxide emitted in the year 2013. Among the most variable energy consumption needs of this sector is that of cold, which can be used, for example, for thermal comfort and food preservation. Most of the equipment for this purpose uses compressors driven by the engine, which causes a considerable energy expenditure and consequently decrease its efficiency. It is known that combustion engines release about 30% of all energy consumed in the form of exhaust gas energy, whereby it is an interesting idea to use this energy to produce cold in a micro-capacity absorption refrigeration system.

## 2 COMPUTATIONAL PROCEDURE

The developed thermal and cost assessment of an absorption refrigeration system was according to the following steps:

• Characterization of the thermal source based on experimental from Justino, *et al*., (2012). Definition of the thermal energy from the engine exhaust gas to be used at different engine loads;

• Definition of the refrigeration cycle as simple effect presented in Fig. 1.;

• Modeling the cycle in the EES. Modeling includes mass, energy, entropy and exergy balances in the cycle components as well as the heat transfer modelling in the heat exchanger of the cycle;

• Cycle and cost optimization aiming the maximum COP at low cost of cold production. Table 1 shows some formulas used for the cold cost calculation base on thermoeconomic analysis presented by Misra, *et al*., (2003).

### 2.1 Characterization of the thermal source

The thermal source considered in this work includes the information reported Justino, *et al*., (2012) that is shown in Tab. 1. The gas composition without consider the NOx, CO and THC is presented in mole fraction. Exhaust gas temperature and flow as well as air and fuel flow are also listed in the table at different engine load. EES procedures were developed to compute the thermodynamic properties.

Table 1 – Thermal source data at different engine load.

| Engine load [kW] | Exhaust Gas temperature [°C] | Fuel flow [kg/h] | Air flow [kg/h] | Exhaust Gas flow [kg/h] | CO2 [-] | O2 [-] | N2 [-] |
|---|---|---|---|---|---|---|---|
| 37.5 | 533 | 9.34 | 148.3 | 157.6 | 0.1454 | 0.0697 | 0.7849 |
| 35.0 | 514 | 8.48 | 146.8 | 155.3 | 0.1365 | 0.0795 | 0.7840 |
| 32.5 | 481 | 7.71 | 146.8 | 154.5 | 0.1254 | 0.0905 | 0.7841 |
| 30.0 | 448 | 7.24 | 146.8 | 154.0 | 0.1147 | 0.1016 | 0.7837 |
| 27.5 | 418 | 6.66 | 146.8 | 153.5 | 0.1057 | 0.1104 | 0.7838 |
| 25.0 | 387 | 6.17 | 148.1 | 154.3 | 0.0969 | 0.1204 | 0.7827 |
| 22.5 | 357 | 5.59 | 148.2 | 153.8 | 0.0884 | 0.1294 | 0.7822 |
| 20.0 | 329 | 5.22 | 148.2 | 153.4 | 0.0802 | 0.1384 | 0.7815 |
| 15.0 | 285 | 4.37 | 148.1 | 152.5 | 0.0663 | 0.1520 | 0.7817 |
| 10.0 | 239 | 3.47 | 148.2 | 151.7 | 0.0525 | 0.1667 | 0.7808 |
| 5.0 | 196 | 2.74 | 149.4 | 152.1 | 0.0399 | 0.1804 | 0.7796 |
| 0.0 | 157 | 1.84 | 149.5 | 151.3 | 0.0276 | 0.1942 | 0.7782 |

## 2.2 Definition of refrigeration cycle

According to Fig. 1, a strong-liquid solution, with a large concentration of ammonia refrigerant leaves the absorber at state 1. This solution is pumped to the condensing pressure, and preheated in the heat exchanger to reduce heating at state 3. The heated strong solution enters the rectifying column. The column produces a weak liquid solution with a low concentration of ammonia refrigerant at the bottom, at state 4, and nearly pure ammonia vapor at the top, at state 7. The solution with a low concentration of ammonia refrigerant enters the solution heat exchanger, and flows through the expansion valve to enter the absorber. The ammonia refrigerant is sent to the condenser at state 7, which condenses it to sub-cooled liquid at state 8 before it enters in the expansion valve. The ammonia leaving the expansion valve at state 9 enters the evaporator, where the liquid phase vaporizes to absorb the refrigerant load in the system. The refrigerant enters in the absorber at state 10, and then it returns to state 1 to begin another cycle.

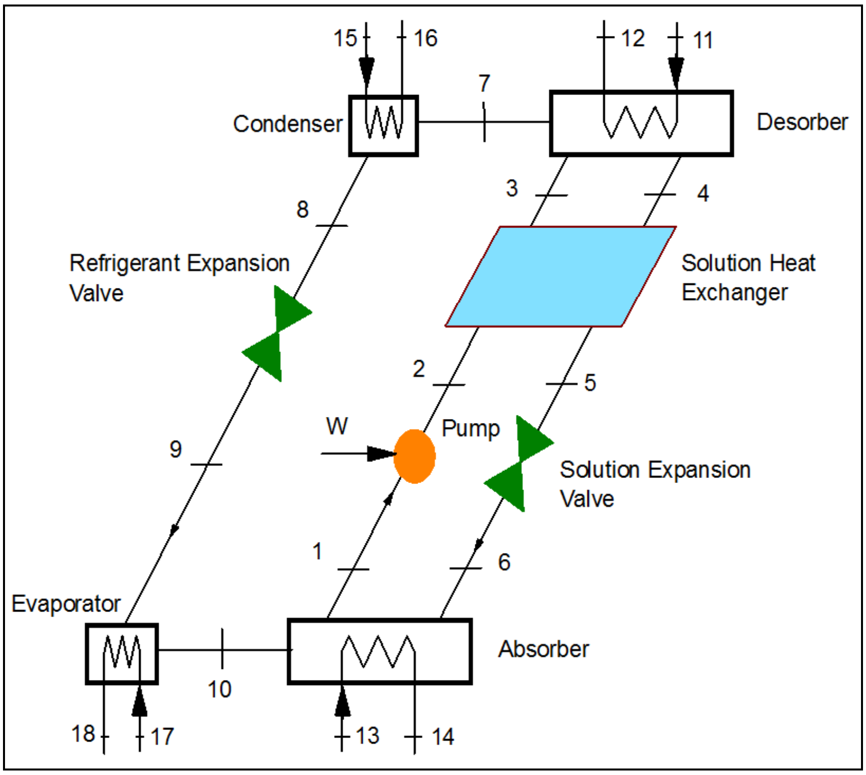



Figure 1 - Cycle of refrigeration by absorption of heat simple effect.

### 2.3 EES model

The EES developed model include the mass, energy, entropy and exergy balances. Additionally the heat transfer relations were considered. The relations included the effectiveness, the logarithmic mean temperature difference and the heat transfer expressions by components.

For the desorber:

$$Eff_d = (T_{11} - T_{12})/(T_{11} - T_7) \tag{1}$$

$$Lmtd_d = (T_{11} - T_4 - T_{12} + T_7)/\ln\{(T_{11} - T_4)/(T_{12} - T_7)\} \tag{2}$$

$$Q_d = Lmtd_d \cdot UA_d \tag{3}$$

For the condenser:

$$Eff_c = (T_{15} - T_{16})/(T_{15} - T_8) \tag{4}$$

$$Lmtd_c = (T_{16} - T_{15})/\ln\{(T_8 - T_{15})/(T_8 - T_{16})\} \tag{5}$$

$$Q_c = Lmtd_c \cdot UA_c \tag{6}$$

For the evaporator:

$$Eff_e = (T_{17} - T_{18})/(T_{17} - T_{10}) \tag{7}$$

$$Lmtd_e = (T_{17} - T_{18})/\ln\{(T_{17} - T_{10})/(T_{18} - T_{10})\} \tag{8}$$

$$Q_e = Lmtd_e \cdot UA_e \tag{9}$$

For the absorber:

$$Eff_a = (T_{14} - T_{13})/(T_6 - T_{13}) \tag{10}$$

$$Lmtd_a = (T_6 - T_{14} - T_1 + T_{13})/\ln\{(T_6 - T_{14})/(T_1 - T_{13})\} \tag{11}$$

$$Q_a = Lmtd_a \cdot UA_a \tag{12}$$

For the solution heat exchanger:

$$Chot_{she} = \dot{m}_4 \cdot (h_4 - h_5)/(T_4 - T_5) \tag{13}$$

$$Ccold_{she} = \dot{m}_2 \cdot (h_3 - h_2)/(T_3 - T_2) \tag{13}$$

$$Lmtd_{she} = (T_4 - T_3 - T_5 + T_2)/\ln\{(T_4 - T_3)/(T_5 - T_2)\} \tag{15}$$

$$Q_{she} = Lmtd_{she} \cdot UA_{she} \tag{16}$$

In the equation 1 to 16 *T* is the temperature at different states accordingly Fig. 1, in °C, *Eff* is the effectiveness, *Lmtd* is the logarithmic mean temperature difference, *Q* is the heat transferred, in kJ/s, *UA* is the thermal conductance, in kW/K, *C* is the capacitance, kW/K referent to the hot and cold side of the solution heat exchanger. The subscript refers to the different components.

For the pump the consumed power, $W_P$, in kW, is computed in function of the inlet specific volume and mass flow, $v_1$, in m³/kg and $m_1$, in kg/s, respectively, high pressures $P_h$ and low pressure $P_l$ of the cycle, in kPa, and the pump isentropic efficiency $\eta$:

$$W_p = v_1 \cdot m_1 \cdot (P_h - P_l)\eta_P \tag{17}$$

The coefficient of performance COP of the cycle is computed with the expression:

$$COP = Q_e / (Q_d + W_p) \tag{18}$$

For the exergy analysis, the physical and the chemical component of specific exergy were taken into account. The methodology to compute the chemical specific exergy consider the suggested by Palacios *et al.* (2010). Finally, for the simulation were considered the data shown in Tab. 2, which include the temperature, pressure and quality in some states of the cycle, the temperature and pressure at the dead state for the exergetic analysis, the pump efficiency and the thermal conductance for the different cycle components.

Table 2 – Inlet data for simulation.

| Parameter | Value | Unit | Parameter | Value | Unit |
|---|---|---|---|---|---|
| $T_{17}$ | 10.0 | ºC | $T_{15}$ | 25.0 | ºC |
| $P_{17}$ | 300.0 | kPa | $P_{15}$ | 180.0 | kPa |
| $q_{17}$ | 0.0 | - | $q_{15}$ | 0.0 | - |
| $T_{18}$ | 4.0 | ºC | $\eta_B$ | 0.8 | - |
| $P_{18}$ | 300.0 | kPa | $UA_s$ | 0.1320 | kW/K |
| $q_{18}$ | 0.0 | - | $UA_a$ | 1.8000 | kW/K |
| $T_{13}$ | 25.0 | ºC | $UA_g$ | 0.1478 | kW/K |
| $P_{13}$ | 180.0 | kPa | $UA_e$ | 2.2500 | kW/K |
| $q_{13}$ | 0.0 | - | $UA_c$ | 1.2000 | kW/K |
| $T_{14}$ | 37.0 | ºC | $T_0$ | 22.0 | °C |
| $q_{14}$ | 0.0 | - | $P_0$ | 101.3250 | kPa |

**2.4 Cycle cost and optimization**

From the exergetic analysis the cost of the system could be estimated using the Exergetic Cost Theory. The investment cost of each component of the system can be determined as follow:

$$z_k = \beta_k \cdot [\eta_k / (1-\eta_k)]^{x_k} \cdot Ep_{p,k}^{y_k} \tag{19}$$

For the pump:

$$z_{pK} = z_{Rp} \cdot \left(\dot{w}_p / \dot{w}_{Rp}\right)^{m_P} \cdot \left[(1-\eta_p)/\eta_p\right]^{n_P} \tag{20}$$

For the valve:

$$z_{Vk} = z_{RV} \cdot I_V \tag{21}$$

The total cost of the system is estimated by the sum of component costs:

$$z_{TOTAL} = \sum_{k=1}^{n} z_k \tag{22}$$

Finally, the cost of the produced cold is computed as:

$$c = \left( z_{TOTAL} \cdot \xi / E_{P,E} \right) \tag{23}$$

Were the amortization factor is:

$$\xi = \left( i \cdot (1+i)^N / \left[ (1+i)^N - 1 \right] \right) \cdot \left[ 1 / \left( N \cdot t \cdot 3600 \right) \right] \cdot s^{-1} \tag{24}$$

In equations 19 to 23, $z$ is the capital investment of each cycle component; $\eta$ refer to the exergetic efficiency; $Ep$ is the exergetic product of the component, in kW; $I_V$ is the irreversibility in the valve; $z_{rp}$ and $z_{rv}$ are the reference cost for the pump and the valve respectively, adopted in 800 US$ and 37 US$ from Misra, *et al*., (2003); $w_{rp}$ is the pump power for the reference pump, adopted in 10 kW, the values of $m_p$ and $\eta_p$ were assumed as 0.26 and 0.5 respectively. All aforementioned values for the pump were adopted from Accadia and Rossi (1998). The values of $\beta_k$, $x_k$ and $y_k$ were assumed as 40265, 0,6848 and 0,8 respectively accordingly to Misra, *et al*., (2003). The interest rate, $i$, was 15% per year, $N$, 10 years and the time operation per year $t$, 1000 h/year.

The absorption refrigeration system optimization were performed using the genetic algorithm of EES. The cycle parameters for optimization are shown in Tab. 3. As can be seen in these table the parameters include the approach temperatures between some states of the cycle, the temperature at state 12, the solution heat exchanger effectiveness and the thermal conductance of the desorber. The range values for the different parameter were defined after several simulations for the consistence of the results.

Table 3 – Optimization parameters.

| Parameter | Minimum value | Maximum value |
|---|---|---|
| $\Delta T_{10\text{-}18}$ | 3.000 | 3.50 |
| $\Delta T_{14\text{-}1}$ | 2.000 | 5.90 |
| $\Delta T_{8\text{-}16}$ | 2.000 | 5.50 |
| $EFF_{HX}$ | 7.945 | 8.05 |
| $T_{12}$ | 8.500 | 9.30 |
| $UA_g$ | 1.450 | 1.70 |

## 3 RESULTS AND DISCUSSION

Table 4 shown energy transferred by absorption refrigeration cycle component and COP as function of engine load. The thermal duty in the refrigeration cycle increase as the engine load rise. The thermal energy transferred in the component increase about 400% at nominal engine load with respect to the minimum one. The COP has low variation with the rise of engine load, which is explained by the high temperature of the engine exhaust gases to the regenerator inlet. Table 5 shown the properties at the refrigeration cycle states for 37.5 kW of engine load. At this load the concentration of the LiBr concentration is far away from the LiBr crystallization zone.

The results showed that a load of 22.5 kW is required in the motor for the operation of the cycle with the chosen parameters. The maximum cold output of 17.9 kW was obtained with the engine in maximum load of 37.5 kW, it presented the lowest cost for cold production 0.058 US$/kW, and the lowest initial investment cost of US $ 75.2 thousands.

With the engine running at a minimum load of 22.5 kW, it is necessary an equipment with higher thermal efficiency, thus generating a higher initial investment cost of US $ 115.0 thousand and US$ 0.164/kW of cold produced, respectively. In this load the cold produced was 9.7 kW.

The COP was virtually the same on all engine loads. However, after optimizing the cycle, a variation of up to 5.6% in the COP value was observed between the cycle operating with the minimum and the maximum engine load, and a low variation with a mean of 0.758.

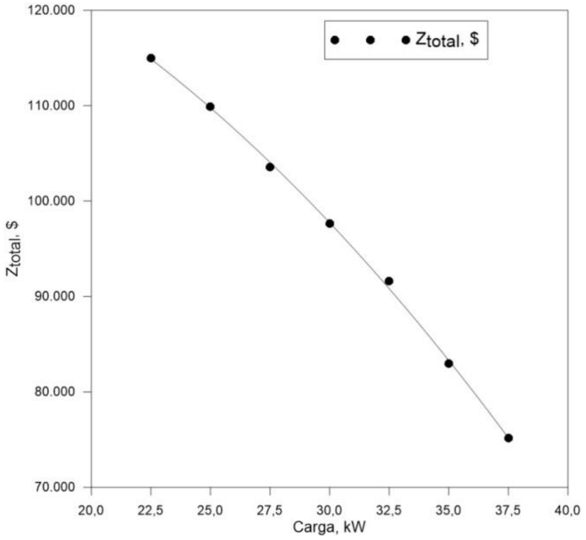
Figure 2 summarize the cost results. Figure 2 (A) show the initial investment as function of engine load. Figure 2 (B) show the cold as function of engine load. When the engine load increase the initial investment and the cold cost fall as a consequence of lower area in the desorber due to the high temperature of the engine exhaust gas when the engine load rise.

Table 4 – Energy transferred by absorption refrigeration cycle component and COP.

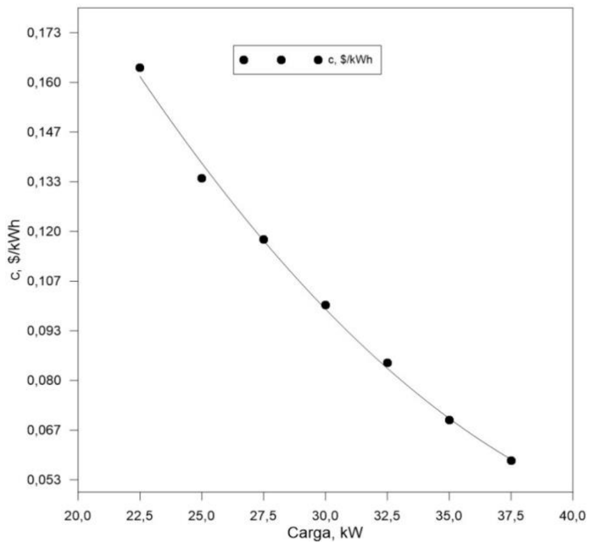
| Engine Load | Heat transferred [kW] | | | | Pump | COP |
|---|---|---|---|---|---|---|
| [kW] | Desorber | Condenser | Evaporator | Absorber | Power | [-] |
| 5.0 | 4.406 | 3.566 | 3.364 | 4.229 | 0.03148 | 0.758 |
| 10.0 | 6.398 | 5.190 | 4.897 | 6.139 | 0.04360 | 0.760 |
| 15.0 | 8.617 | 6.991 | 6.596 | 8.268 | 0.05857 | 0.760 |
| 20.0 | 10.801 | 8.748 | 8.253 | 10.367 | 0.07597 | 0.759 |
| 22.5 | 12.199 | 9.867 | 9.309 | 11.712 | 0.08904 | 0.758 |
| 25.0 | 13.724 | 11.084 | 10.456 | 13.181 | 0.10600 | 0.756 |
| 27.5 | 15.193 | 12.261 | 11.567 | 14.598 | 0.12380 | 0.755 |
| 30.0 | 16.759 | 13.521 | 12.756 | 16.114 | 0.15070 | 0.754 |
| 32.5 | 18.491 | 14.939 | 14.094 | 17.799 | 0.19160 | 0.754 |
| 35.0 | 20.288 | 16.458 | 15.527 | 19.566 | 0.26120 | 0.756 |
| 37.5 | 21.608 | 17.672 | 16.672 | 20.916 | 0.38450 | 0.758 |

Table 5 – Properties at the refrigeration cycle states for 37.5 kW of engine load.

| Estado | P [kPa] | T [°C] | $\dot{m}$ [kg/s] | h [kJ/kg] | X [%Li-Br] | s [kJ/kg-K] | V [m³/kg] |
|---|---|---|---|---|---|---|---|
| 1 | 0.681 | 33.000 | 0.075 | 86.093 | 56.749 | 0.201 | 0.617 |
| 2 | 7.353 | 35.040 | 0.075 | 90.211 | 56.749 | 0.215 | 0.618 |
| 3 | 7.353 | 57.984 | 0.075 | 136.575 | 56.749 | 0.358 | 0.622 |
| 4 | 7.353 | 72.467 | 0.068 | 191.613 | 62.744 | 0.403 | 0.583 |
| 5 | 7.353 | 44.771 | 0.068 | 140.351 | 62.744 | 0.248 | 0.577 |
| 6 | 0.681 | 44.771 | 0.068 | 140.351 | 62.744 | 0.248 | 0.577 |
| 7 | 7.353 | 76.886 | 0.007 | 2643.395 | 0.000 | 8.468 | 21.938 |
| 8 | 7.353 | 39.928 | 0.007 | 167.202 | 0.000 | 0.571 | 0.001 |
| 9 | 0.681 | 1.500 | 0.007 | 167.202 | 0.000 | 0.609 | 11.986 |
| 10 | 0.681 | 1.500 | 0.007 | 2503.279 | 0.000 | 9.114 | 185.979 |
| 11 | 101.320 | 533.000 | 0.044 | 1014.550 | 0.000 | 9.202 | 4.178 |
| 12 | 101.320 | 100.000 | 0.044 | 1508.010 | 0.000 | 8.620 | 3.187 |
| 13 | 180.000 | 25.000 | 0.417 | 104.753 | 0.000 | 0.367 | 0.001 |
| 14 | 180.000 | 37.000 | 0.417 | 154.954 | 0.000 | 0.532 | 0.001 |
| 15 | 180.000 | 25.000 | 0.443 | 104.753 | 0.000 | 0.367 | 0.001 |
| 16 | 180.000 | 34.528 | 0.443 | 144.613 | 0.000 | 0.499 | 0.001 |
| 17 | 300.000 | 10.000 | 0.662 | 41.988 | 0.000 | 0.151 | 0.001 |
| 18 | 300.000 | 4.000 | 0.662 | 16.819 | 0.000 | 0.061 | 0.001 |




(A)




(B)

Figure 2 – Initial investment and specific cost of cold as function of engine load.

## 4 CONCLUSION

A cold production up to 17.9 kW was achieved at the maximum engine load of 37.5 kW with a cost of 0.058 US$/kW. The minimum engine load of 22.5 kW was required for the operation of the absorption refrigeration system. In this condition, the cold produced was 9.7 kW with a cost of 0.164 US$/kW. The COP suffered little variation with the engine load. It is ideal that the engine works at full load, to obtain the highest cold production at a minimum cost.

## 5 ACKNOWLEDGEMENTS

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