Modeling and Simulation of a Desiccant Assisted Brayton Refrigeration Cycle

Carlos E.L. Nobrega
nobrega@pobox.com
Leandro Alcoforado Sphaier

Follow this and additional works at: http://docs.lib.purdue.edu/iracc

http://docs.lib.purdue.edu/iracc/1279
Modeling and Simulation of a Desiccant Assisted Brayton Refrigeration Cycle

C.E.L. Nobrega¹*, L.A. Sphaier²

¹CEFET-Rio, Departamento de Engenharia Mecânica
Rio de Janeiro, RJ, Brazil
(+55) 21 9901-1263, nobrega@pobox.com

²Universidade Federal Fluminense, Departamento de Engenharia Mecânica
Rio de Janeiro, RJ, Brazil
(+55) 21 2629-5576, lasphaier@id.uff.br

* Corresponding Author

ABSTRACT

The phase-out of CFCs has shed a new light over natural refrigerants, which have null global warming potentials. Air would be a natural choice, and although the Brayton cycle usually exhibits a lower coefficient of performance when compared to vapor-compression systems of same capacity, it has been considered in applications other than aircraft cooling. These include gas separation, food processing and preservation, refrigerated containers and train air-conditioning. Price perspectives in the oil market also make the Brayton cycle an alternative to be considered as an option for automotive air conditioning. Even though the Brayton cycle is often employed in low temperature applications, the ambient humidity level is essential for the uninterrupted operation. For applications far below the ambient air dew point, the condensate is likely to cause icing at the turbine outlet, causing duct obstruction and system failure. The use of a solid desiccant would provide a thorough humidity control, allowing for increased pressure ratios (and thus lower expansion temperatures) even for significant ambient humidity levels. In the standard Brayton refrigeration cycle, the air is collected by the compressor at ambient conditions, and compressed through a specified compression ratio. The air is then cooled back to the ambient temperature at a regenerator, and subsequently expanded through a turbine to the ambient pressure, at a low temperature. At the proposed cycle, the air is collected by a desiccant wheel and dehumidified, before it is admitted to the compressor. Accordingly, it can be compressed under a significant pressure ratio, without incurring in ice formation when later expanded. The desiccant wheel is dried using the hot air at the compressor outlet, by a heat exchanger which collects the heat that would be otherwise dumped by the regenerator. A mathematical model for the proposed cycle is developed, consisting of a system of non-linear equations which stems from mass and energy balances applied to each individual cycle component. The results show that the desiccant assisted cycle allows for frost-free operation even for temperatures below -60°C, which is required for fish preservation warehouses.

1. INTRODUCTION

The increasing consciousness allied to the enforcement of environmental friendly policies, has launched the HVAC industry in a search for refrigerants with low ozone depleting and global warming potential. The scheduled replacement of traditionally used refrigerants by some chlorine-free compounds has been inhibited by technical problems such as leak detection and lubrication (Spatz et al., 2011). Accordingly, natural refrigerants have been increasingly investigated. Air and water are an inexpensive choice, in addition to the absence of toxicity and flammability. This sheds a new light on Brayton refrigeration cycles, which were traditionally segregated due to their lesser performance when compared to vapor-compression systems of the same capacity. Brayton Refrigeration cycles have been restricted to the aerospace industry, since the compressed air can be harvested at the propulsion engines. However, it has been discussed as an option in a number of applications, such as train air-conditioning systems (Murphy et al, 1994), refrigerated containers (Spence et al, 2004), food display cases and drying (Peloscio, 2001). Moreover, for the ultra-low temperature field (below -50°C), HFC23/HCF22 binary vapor compression
systems are used, although HCF22 is regulated by the Montreal Protocol, which recommends its phase out by 2020. Although HFC23 has null ozone depletion potential, it has a high global warming potential of 11700, and is manufactured as a by-product or HCF22 (Machida and Boone, 2011). The use of a desiccant system in series with a Brayton air refrigeration cycle was first proposed by El-Sayed et al (2008). The results showed improved performance when compared to the standard Brayton cycle; however, the analysis neglected the control of humidity at the turbine outlet, which is a critical parameter of operation. Whenever the air stream expanding through the turbine is rich in water vapor, some condensation is bound to occur, as the air temperature drops. This phenomenon is detrimental from the energy conservation perspective, since it feeds on the work to be recovered by the turbine. Moreover, for low temperature applications, the condensate is likely to cause icing in the turbine outlet duct, which will eventually result in blockage and possible system failure (Hamlin, 1998). Figure (1) shows the standard air refrigeration cycle, which is ideally comprised of an isentropic compression, followed by an isobaric heat rejection to the ambient. The cycle continues with an isentropic expansion through a turbine, which significantly lowers the air temperature. The cycle is often designed in an open configuration, completed with a heat uptake occurs at ambient pressure, between the turbine outlet and the compressor inlet. Figure (2) shows that pressure ratio is significantly limited by the outside air relative humidity, if frosting is to be avoided.

Figure (3) shows that the coefficient of performance value is strongly dependent on the chosen warehouse temperature level. The increase in the value of the pressure ratio simultaneously increases the compressor work and decreases the turbine outlet temperature (thereby increasing the cooling effect). For low warehouse temperatures, the later effect overcomes the former, resulting in an increased value for the COP at low values of the pressure ratio. For higher warehouse temperatures, however, the COP value continuously decreases with the pressure ratio.

The proposed desiccant assisted Brayton refrigeration cycle is shown in Fig. (4). In contrast to the standard Brayton cycle, which compresses air at atmospheric conditions, the proposed cycle process the air stream in a desiccant wheel before it is admitted to the compressor. Accordingly, the air will be at a higher temperature and at a lower absolute humidity, when compared to the atmospheric condition. Moreover, the proposed cycle employs an evaporative cooler before the heat exchanger, allowing the turbine inlet temperature to lie below the outside ambient temperature.
2. MATHEMATICAL MODELING

The mathematical procedure relies on modeling each cycle component as an individual control volume. The outlet condition at each component outlet is determined by two equations which stem from energy and mass balances applied to each control volume. Some simplifying assumptions are required:

1. The moist air is regarded as a mixture of ideal gases (dry-air and water)
2. Specific heats are constants in the range of the present simulation
3. The evaporative cooling is an isenthalpic process
4. The effectiveness of each component is previously known, and independent from operation conditions.

Considering equal flow rates on the two branches of the cycle, the heat exchanger effectiveness is defined as

\[ \varepsilon_{\text{ex}} = \frac{T_1 - T_4}{T_5 - T_1} \]  

(1.a)

Whereas the mass and energy balances are written as
The evaporative cooler requires an energy balance and an effectiveness relation:

\[ i_b = i_r, \quad e_{ec} = \frac{T_e - T_i}{T_b - T_{i,\text{sat}}} \]  

(2)

The compression and expansion in the Brayton portion of the cycle are represented by

\[
T_3 = T_2 \left(1 + \eta_c \left(\frac{r_p^{k-1}}{k} - 1\right)\right), \\
T_2 = T_4 \left(1 - \eta_c \left(1 - r_p^{k-1}\right)\right)
\]  

(3.a)

The mass balances through the compressor and turbine are given by

\[ Y_3 = Y_2, \quad Y_6 = Y_4 \]  

(3.b)

The desiccant wheel outlet states can be obtained by the solution of discretized mass and energy transport equations (Spahier and Worek, 2009), (Nobrega and Brum, 2011), or using the analogy method (Mac-Laine Cross and Banks, 1972), (Panaras et al., 2007), (Nobrega and Sphaier, 2012), which consists on determining the outlet states by solving the following non-linear algebraic system

\[
\eta_1 = \frac{F_1(T_2, Y_2) - F_1(T_1, Y_1)}{F_1(T_2, Y_2) - F_1(T_1, Y_1)}, \\
\eta_2 = \frac{F_2(T_2, Y_2) - F_2(T_1, Y_1)}{F_2(T_2, Y_2) - F_2(T_1, Y_1)}
\]  

(4.a)

In which these effectiveness are previously known and depend on the type of desiccant wheel. The functions \( F_1 \) and \( F_2 \) are defined in terms of temperature and absolute humidity as

\[
F_1(T, Y) = -\frac{2865}{(T + 273.15)^{1.40}} + 4.344Y^{0.8624} 
\]  

(4.b)

\[
F_2(T, Y) = \frac{(T + 273.15)^{1.40}}{6360} - 1.127Y^{0.07969}
\]  

(4.c)

Finally, in order to fully determine the outlet states of the desiccant wheel, mass and energy balances across this component are employed:

\[ i_2 - i_1 = i_b - i_a, \quad Y_2 - Y_1 = Y_b - Y_a \]  

(4.d)

The aforementioned system is numerically solved using an iterative procedure based on Newton’s method, with a progressive initial estimate selection scheme (Wolfram, 2003). Once the system has been solved, the coefficient of performance (COP) and the heat balance error (HBE) are calculated for evaluating the cycle performance and the accuracy of the numerical results.
\[ COP = \frac{q_L}{w_s} , \quad HBE = \frac{w_s + q_L - q_H}{q_H} \]  

(5)

Where

\[ w_s = (i_3 - i_2) - (i_4 - i_3) \]  

(6)

\[ q_L = (i_{EXH} - i_2) , \quad q_H = (i_g - i_1) \]  

(7)

The enthalpy of humid air given as

\[ i = C_{pa} T + Y (i_{fg} + C_{pv} T) \]

\[ C_{pa} = 1.005 \text{ kJ/kgK} \]

\[ C_{pv} = 1.86 \text{ kJ/kgK} \]

\[ i_{fg} = 2501 \text{ kJ/kgK} \]  

(8)

The effectiveness of the evaporative cooler, heat exchanger and the turbine are used as input parameters. The compressor efficiency is a function of the chosen pressure ratio and mass flow rates, and is evaluated according to the following performance map (Leufvén, 2010).

3. RESULTS

The coefficient of performance (COP) was evaluated as a function of the pressure ratio for different values of the compressor efficiency, for a given value of the turbine efficiency. Figure 6 shows that, for \( T_{EXH} = 0^\circ \text{C} \), the COP increases with the pressure ratio, as expected. In contrast, Figure 7 shows that for a higher temperature of the warehouse, a continuous decrease of the COP takes place. This behavior is consistent with the behavior of the standard Brayton cycle depicted in Fig. (3). In both figures it is evident the effect of the compressor isentropic efficiency over the cycle performance. However, Figure (8) shows that a decrease in the compressor efficiency benefits the humidity control, since more heat will be available to drive the desiccant system. Allowing for a thorough air dehumidification. Moreover, decreased compressor efficiencies will imply in a greater temperature (and thus reduced relative humidity) at the turbine outlet. Figure (9) shows how increased specific cooling loads can be attended by increasing the pressure ratio. Finally, Fig.(10) shows the impact of using a compressor efficiency as a function of the mass flow rate and pressure ratio, following the performance map given by Fig.(5). A small
difference between the curves can be seen, which stems from similar values of the efficiency provided by the compressor performance map, in the range of interest.

Figure 6: COP as a function of compression ratio for $T_{EXH} = 0^\circ$C.

Figure 7: COP as a function of compression ratio for $T_{EXH} = 25^\circ$C.

Figure 8: Turbine outlet relative humidity as a function of compression ratio
4. CONCLUSIONS

The present work is dedicated to the modeling and simulation of a desiccant-assisted Brayton refrigeration cycle. It was seen that although the proposed cycle has a decreased coefficient of performance when compared to the standard Brayton cycle, the desiccant action allows for the Brayton cycle to operate at increased pressure ratios without incurring in frost formation, which is extremely detrimental to the refrigeration system performance and liability. An analysis using a typical compressor performance map was carried out, and the proposed system was also shown to be able to operate at different mass flow rates with a similar value for the COP, allowing for eventual fluctuations of the cooling load to be handled. A complete analysis would require the present cycle performance to be compared to that of the standard Brayton cycle, including the energy intensive defrost process. This analysis, however, is beyond the scope of the present effort.

NOMENCLATURE

\begin{align*}
  c_{pa} & \quad \text{air specific heat, kJ/(kg K)} \\
  c_{pv} & \quad \text{vapor specific heat, kJ/(kg K)} \\
  F_{1,2} & \quad \text{auxiliary functions} \\
  HBE & \quad \text{heat balance error} \\
  i & \quad \text{specific enthalpy, J/(kg K)} \\
  k & \quad \text{specific heat ratio}
\end{align*}
REFERENCES


Pelsoci, T.; Closed-cycle air refrigeration technology, 2001. NIST report GCR 01-819


ACKNOWLEDGEMENT

The authors would like to acknowledge the support of Centro Federal de Educação Tecnológica (CEFET-RJ) e Universidade Federal Fluminense (UFF), as well as the Brazilian funding agencies CAPES, CNpQ and FAPERJ.