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Research Paper

# Feasibility and operating costs of an air cycle for CCHP in a fast food restaurant

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## HIGHLIGHTS

- A power, heating and cooling Brayton cycle to serve fast food restaurants is presented.
- The numerical cycle model relies on existing correlations to calculate compressor/turbine performance.
- The power, heating and cooling loads for fast food restaurants are calculated for "mixed-humid" and "cold/very cold" climates.
- The model shows that the loads can be met.
- Capital costs are relatively low, but operational costs are high for the Brayton cycle.

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## ABSTRACT

This work considers the possibilities of an air-based Brayton cycle to provide the power, heating and cooling needs of fast-food restaurants. A model of the cycle based on conventional turbomachinery loss coefficients is formulated. The heating, cooling and power capabilities of the cycle are extracted from simulation results. Power and thermal loads for restaurants in Knoxville, TN and in International Falls, MN, are considered. It is found that the cycle can meet the loads by setting speed and mass flow-rate apportionment between the power and cooling functional sections. The associated energy costs appear elevated when compared to the cost of operating individual components or a more conventional, absorption-based CHP system. A first-order estimate of capital investments is provided. Suggestions for future work whereby the operational costs could be reduced are given in the conclusions.



## 1. Introduction: why the Brayton cycle

Brayton cycle-based gas turbines are used for aircraft propulsion and for ground-based power generation. Their dynamic response and fuel efficiency plus ability to operate at high altitudes have made them the engine of choice for almost all types of aircraft. The low heat rates, load-following capabilities, together with straightforward permitting have made them a definitive player in the power generation arena.

Aircraft cabins flying at high altitudes or in cold days on the tarmac need heating for comfort and operability. When flying low altitudes or in a hot day on the tarmac, cabins need cooling air. To provide heating, engine compressor air is used for heating via valves controlling the discharge rate and stage. Compact A/C units operating on reverse Brayton cycles are used to provide chilled air [1]. Some aircraft have conventional air conditioners, but the simplicity and direct application of the air cycles currently dominate the aircraft heating and cooling industry.

The Brayton (power and heating) and reverse Brayton cycle (cooling) are described briefly in this paragraph and the next. Air is assumed to be a perfect gas, and ideal standard cycles are considered for description. In a *T*–*s* diagram (Fig. 1), the constant pressure lines are shown schematically. Air at ambient conditions is compressed isentropically (1c to 2pt), heated (2pt to 1pt) and expanded in a turbine (1pt-3pt). The air can be delivered directly for heating, or cooled to 1c to recommence the cycle. In aircraft, pressurized and hot air is obtained from state points between 1c and 2pt. In real turbines, the process 2pt-1pt includes combustion, whereby the mixture at 3pt is unsuitable for use in conditioned spaces and constitutes the exhaust of the engine.

A particularity of the *T*–*s* diagram is that the vertical spacing between two constant pressure lines increases (exponentially, in

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**Nomenclature**

| | |
|---|---|
| $a_{xyz}$ | coefficients |
| $A$ | duct area, m$^2$ |
| $ah$ | absolute humidity |
| $cp$ | specific heat, J/kg K |
| $CHP$ | combined heat and power |
| $ct$ | cooling turbine |
| $D$ | diameter, m |
| $DF$ | diffusion factor |
| $FM$ | figure of merit for CHP |
| $h$ | specific enthalpy, J/kg |
| $\Delta h_{vf}$ | specific enthalpy of vaporization, J/kg |
| $k$ | isentropic exponent |
| $mfr$ | mass flow rate, kg/s |
| $M$ | mach no. |
| $M_{ie}$ | tip mach no. |
| $Nv$ | number of vanes |
| $OA$ | outdoor air |
| $p$ | pressure, Pa |
| $pt$ | power turbine |
| $PR$ | pressure ratio |
| $PM$ | primary energy |
| $Q$ | heat, kJ |
| $\dot{Q}$ | heat rate, kW |
| $R$ | radius, m |
| $Ra$ | air gas constant, kJ/kg K |
| $RA$ | return air |
| $s$ | specific entropy, kJ/kg K |
| $SA$ | supply air |
| $Su$ | summer |
| $SPR$ | rotational speed ratio |
| $SCCHP$ | seamless cooling combined heat and power |
| $T$ | absolute temperature, K |
| $TCA$ | cooling turbine air |
| $THA$ | power turbine air |
| $U$ | metal velocity, m/s |
| $V$ | absolute velocity, m/s |
| $Vfr$ | volume flow rate, m$^3$/s |
| $V_s$ | velocity of sound, m/s |
| $V_{spt}$ | spout velocity, m/s |
| $W$ | work, kJ; relative velocity, m/s |
| $Wi$ | winter |
| $\dot{W}$ | mechanical power, kW |

*Greek letters*

| | |
|---|---|
| $\beta$ | vane angle |
| $\varepsilon$ | recuperator thermal effectiveness |
| $\eta$ | isentropic efficiency |
| $\rho$ | density, kg/m$^3$ |
| $\sigma$ | slip coefficient |
| $\omega$ | angular velocity, 1/s |

*Super and subscripts*

| | |
|---|---|
| $\cdot$ | indicates rate |
| $a$ | humid air |
| $c$ | compressor |
| $C$ | cooling |
| $ct$ | cooling turbine |
| $e$ | exit |
| $eff$ | effective, required |
| $H$ | Heating |
| $i$ | inlet |
| $inp$ | input |
| $o$ | stagnation conditions |
| $pt$ | power turbine |
| $r$ | reference or relative; radial direction |
| $s$ | isentropic, or sound |
| $st$ | steam |
| $t$ | turbine |
| $u$ | azimuthal |

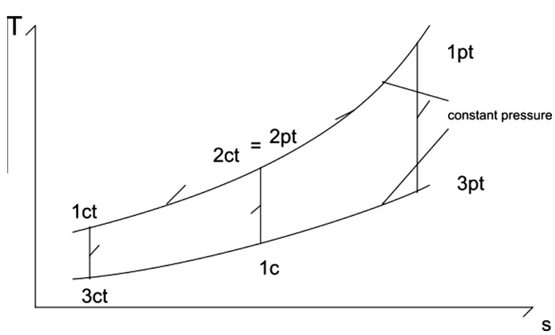


**Fig. 1.** Brayton cycle (1-2pt-1pt-3pt) and reverse Brayton cycle (1-2ct-1ct-3ct) for an ideal gas. Note that the temperature of 3ct is below that of 1.

fact) with absolute temperature. For a perfect gas undergoing an adiabatic compression or expansion, the power exchanged is given by

$$\dot{W} = mfr \cdot cp \cdot (T_{final} - T_{initial}) \quad (1)$$

Since the turbine temperature difference (1pt-3pt) is greater than the one in the compressor (1c-2pt), it follows from Eq. (1) that the power delivered by the Brayton cycle turbine exceeds that required by the compressor, and the turbine can drive a fan or generator. In the reverse Brayton cycle, the air is compressed (1c to 2c), cooled close to ambient (2c to 3c) and expanded (3c to 4c). The cooling cycle needs to receive external power for sustained operation, as could be concluded from Eq. (1). The chilled air can be delivered to the conditioned space, previous to dehumidification at an opportune point in the cycle.

The reverse Brayton cycle has been proposed for a number of A/C applications. For racing car settings, where light weight and simplicity are of importance, Forster and Lemiuex [2] implemented and tested an automotive cooling system using automotive turbocharger components. Compressor pressure ratios of 1.7 at speeds bordering 110,000 rpm were reported. With dry air the cooling COP hovered around 0.38 with a capacity of 1.6 kW. Whereas the COP is not high, the work showed the feasibility of using existing components to produce cooling. Another novel cooling application is suggested in [3], where it is proposed to use the reverse Brayton cycle to cool the inlet air stream into a gas turbine. Cooling the inlet stream reduces the heat rate and increases the capacity of the machine. The extra cooling power is calculated to be smaller than the increment of power due to inlet cooling. In this case, by cooling the air stream with seawater a net power gain in the order of 19% is projected.

Brayton cycles can be used for simultaneous production of heating and cooling. In [4], an experimental system was developed to ascertain if air for freezing and cooking could be provided using compressor air and a reverse Brayton cycle. Functionally the somewhat complicated arrangement worked well: cooling loads ranging from 1.7 to 3.6 kW at temperatures between −140 °C to −80 °C were obtained. The total heating load provided was in the order