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Research Paper

## Solar driven carbon dioxide Brayton cycle power generation with thermal compression

Pramod Kumar<sup>a,\*</sup>, Pradip Dutta<sup>a</sup>, Srikantiah Srinivasa Murthy<sup>b</sup>, Kandadai Srinivasan<sup>b,c</sup><sup>a</sup> Department of Mechanical Engineering, Indian Institute of Science, Bangalore 560 012, India<sup>b</sup> Interdisciplinary Centre for Energy Research, Indian Institute of Science, Bangalore 560 012, India<sup>c</sup> School of Mechanical and Chemical Engineering, University of Western Australia, Crawley, WA 6009, Australia

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## ABSTRACT

Solar thermal power generation by and large uses Rankine cycles with organic working fluids or steam when highly concentrated solar source is used. Brayton cycle is seldom used because the work of compression forms a major fraction of turbine output. We investigate removal of this lacuna by adopting thermal compression using the adsorption route. Here we propose a two source operated cycle with the low temperature source used for thermal compression and the high temperature source to increase the inlet temperature of the working fluid to the turbine. We adopt carbon dioxide as the working fluid in view of its excellent heat transfer properties and environmentally friendly disposition. Activated carbon is used as the adsorbent in the thermal compression process. It is shown that, though the First law thermal efficiency does not show the real merit of the cycle, the exergetic efficiency is substantially high even for low side operating pressures in the range of 15–25 bar. In particular, where the waste heat is available around 100 °C and concentrated solar power is available even at 200–300 °C, exergetic efficiencies in excess of 25% can be realised. When other advantages such as dispensing with one whole segment of moving parts (compressor) are taken into account, the running costs of such a cycle can be so low that it would be a viable proposition. However, we appreciate that, large scale use of adsorption compression is associated with huge technological problems for which we propose some remedies.

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## 1. Introduction

The concept of Brayton cycle with thermal compression was introduced by Dutta et al. [1]. After comparing several working fluids they recommended that carbon dioxide (CO<sub>2</sub>) can be a working fluid. This fluid has been proposed for concentrated solar power generation. However, high efficiencies achievable come with a collateral premium of high pressures and temperatures imposed by adoption of supercritical Brayton cycle [2]. The obvious weakness of subcritical Brayton cycle is that the work of compression consumes more than 60% of the work of expansion [3]. Analysis of Dutta et al. [1] was limited to the heat source being available at 100 °C, both for thermal compression and expansion. Obviously, at such low temperatures the thermal efficiencies are deplorably low (<5%). Even conceding that conventional first law efficiency when low grade heat sources are used does not give an encouraging picture, it will be inevitable to raise the temperature of the working fluid before expansion not only to extract more work

but also to realise higher efficiencies. Partitioning of low and high grade thermal energies for thermal compression and enhancing enthalpy of the working fluid respectively could be a useful concept. For example, thermal energy (which may be at about 100 °C) from good quality evacuated tubular collectors can be used for thermal compression. Heat derived from a field of concentrating collectors can increase the enthalpy of the working fluid before expansion. In doing so, it may be possible to obviate the drawbacks of low pressure Brayton cycle power generation. This approach is also logical extension of methodology used in sorption refrigeration for augmenting the refrigerant pressure [4–7]. Garg et al. [3] showed that, in the case of CO<sub>2</sub>, optimum compression ratios required for low pressure CO<sub>2</sub> Brayton cycles are ~2, which is easily obtainable with reasonable uptake efficiencies of adsorption thermal compressors [1]. As a next step post compression, the enthalpy of CO<sub>2</sub> can be raised with another heat source for which concentrating solar collectors can be used. This paper explores the potential and limitations of this approach and characterizes it through a few performance indicators such as specific work output per unit mass of adsorbent and thermal efficiency based on exergy component of the heat. For the sake of proving the concept, this

\* Corresponding author.

E-mail address: [pramod@mecheng.iisc.ernet.in](mailto:pramod@mecheng.iisc.ernet.in) (P. Kumar).

**Nomenclature**

C	uptake (kg of CO <sub>2</sub> /kg of adsorbent)
E	exergy (kJ)
h	enthalpy (kJ/kg)
m	mass (kg)
$\dot{m}$	mass flow rate (kg/s)
p	pressure (bar)
r	expansion ratio ( $p_H/p_L$ )
s	entropy (kJ/kg K)
T	temperature (K or °C)
$\eta$	efficiency
$\rho$	density (kg/m <sup>3</sup> )

<i>Subscripts</i>	
1	source 1
2	source 2
ch	adsorbent
eff	effective
H	high side
L	low side
s	isentropic
u	uptake

analysis focuses on thermal compression through solid sorption route with Maxsorb III as the adsorbent analogous to the previous work of Dutta et al. [1].

**2. Upper cycle pressure limitations in thermal compression Brayton cycle**

The processes on a temperature-entropy plane are depicted in Fig. 1

It is not possible to show the thermal compression process from 1 to 2 or 1 to 2<sub>1</sub> or 1 to 2<sub>3</sub> on the pure fluid T-s diagram, although the two end states of compression process are known. If another heat source at a higher temperature of T<sub>H2</sub> is available for enhancing the enthalpy of the working fluid, heat addition can occur along the isobar p<sub>H2</sub> (2–3) such that h<sub>3</sub> > h<sub>2</sub>. The processes 3–4 and 4–1 are the same as in a conventional Brayton cycle. Unlike in the earlier case of Dutta et al. [1], the state 2 can now be moved along the T<sub>H1</sub> isotherm (e.g.: 2<sub>3</sub> or 2<sub>1</sub>). Consequently several high side pressures, such as p<sub>H1</sub> and p<sub>H3</sub> can also be considered. It is also apparent that higher this pressure, more is the work of expansion. For example, since p<sub>H3</sub> > p<sub>H2</sub> > p<sub>H1</sub>, ideal specific work output  $\Delta h_{33-43} > \Delta h_{3-4} > \Delta h_{31-41}$ . For the sake of explanation, the lower cycle pressure will be fixed at p<sub>L</sub>, although this can also be varied.

A schematic layout of the Brayton cycle with a thermally driven compressor is shown in Fig. 2.

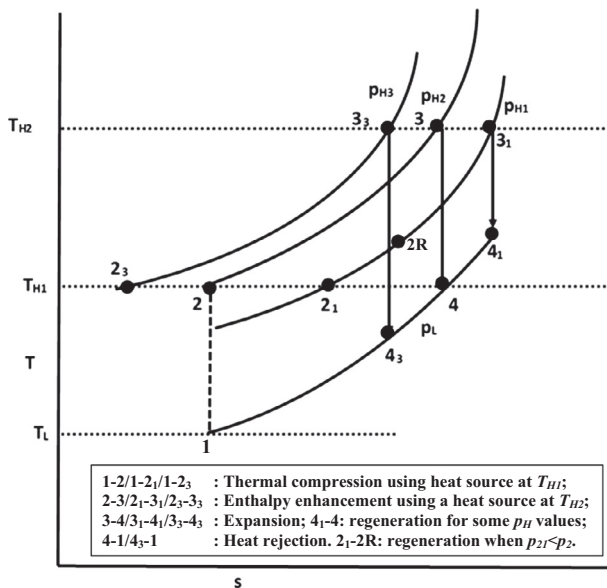


Fig. 1. Processes in a thermal compression Brayton cycle.

In this figure states corresponding to Fig. 1 are shown for the case where a regenerator is required. When it is not the case, the dotted line shows the path. In the previous analysis of Dutta et al. [1] the cycle comprised of thermal compression from state 1 to 2 followed by mechanical expansion from 2 to 1. In that configuration, the additional heat input, regenerator and heat rejection unit were absent. Consequently, the pressure at state 2 was dictated by the entropy at state 1 [ $s_L(p_L, T_L)$ ] and the temperature of the heat source (T<sub>H1</sub>). This is nothing but solution of

$$s(p_2, T_{H1}) = s(p_L, T_L) \tag{1}$$

Although, it appears from Fig. 1 that the high side pressure can be varied, there is only a narrow range of pressures through which the cycle can operate which is restrained by adsorption characteristics of CO<sub>2</sub> on activated carbons. The thermal compression process is shown in Fig. 3 and has been described in detail in several publications [8–11].

The throughput of the adsorption compressor depends on the difference between uptakes (C<sub>b</sub>-C<sub>d</sub>). For an ideal compressor which has no effect of void volume in the adsorber and in which the equilibrium conditions prevail, the upper limit of p<sub>H</sub> is given by the solution of

$$C(p_L, T_L) = C(p_H, T_{H1}) \tag{2}$$

This is represented by state E in Fig. 3. The isotherm equations for CO<sub>2</sub> adsorption by Maxsorb III will be required which are taken from [1] and is summarized below:

$$C = 3.06 \frac{1.17 \times 10^{-5} e^{\frac{463}{RT}} p}{\left[ 1 + \left( 1.17 \times 10^{-5} e^{\frac{463}{RT}} p \right)^{0.66} \right]^{1.515}} \tag{3}$$

In the above equation, p is in bar, T in K, and R is the characteristic gas constant for CO<sub>2</sub> (0.1889 kJ/kg K). The above equation is a refit of experimental data for adsorption of CO<sub>2</sub> on Maxsorb III reported by Saha et al. [12]. However, a realistic compressor has enormous void volume contributed by macropores and interparticle void between particles of activated carbon. Consequently, the thermal compression cycle in Fig. 3 changes to a'-b'-c'-d, implying that the effective uptake difference across which the adsorber operates is (C<sub>b</sub>-C<sub>a'</sub>) < (C<sub>b</sub>-C<sub>d</sub>). This is analogous to the clearance volume of a positive displacement compressor. Similar to volumetric efficiency in a positive displacement compressor, Akkimaradi et al. [13] describe a parameter called uptake efficiency, which is given below

$$\eta_u = \frac{C_b - C_{a'}}{C_b - C_a} = 1 - \frac{\rho(p_H, T_{H1}) - \rho(p_L, T_L)}{C(p_L, T_L) - C(p_H, T_{H1})} \left( \frac{1}{\rho_{eff}} - \frac{1}{\rho_{ch}} \right) \tag{4}$$

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