Solar Driven Co₂ Brayton Cycle Power Generation with Thermal Compression

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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.

Key words: S-CO₂, Sub-critical, Thermal Compression, Adsorption, Brayton Cycle

Nomenclature

C uptake (kg of CO₂/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)

η efficiency

ρ density (kg/m³)

Subscripts

1 source 1

2 source 2

ch adsorbent

eff effective

H high side

L low side

s isentropic

u uptake

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_2) 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 merely 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 thermal 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_2 , optimum compression ratios required for low pressure CO_2 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_2 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 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.

Figure 1: Processes in a thermal compression Brayton cycle

It is not possible to show the thermal compression process from 1-2 or 1-2₁ or 1-2₃ 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_{H2} is available for enhancing the enthalpy of the working fluid, heat addition can occur along the isobar p_{H2} (2-3) such that $h_3 > h_2$. 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_{H1} isotherm (eg: 2_3 or 2_1). Consequently several high side pressures, such as p_{H1} and p_{H3} can also be considered. It is also apparent that higher this pressure, more is the work of expansion. For example, since $p_{H3} > p_{H2} > p_{H1}$, 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_L , although this can also be varied.

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

Figure 2: Schematic diagram of thermal compression Brayton

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 1-2 followed by mechanical expansion from 2-1. 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_{H1}). This is nothing but solution of

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

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 CO₂ on activated carbons. The thermal compression process is shown in Fig.3 and has been described in detail in several publications [8-11].

Figure 3: Thermal compression process on p-C-T plane. a-b-c-d: ideal with no void volume effect, a'-b'-c'-d': practical cycle with void volume effect; pressure at E: maximum p_H for ideal case; pressure at F: when $\eta_u=0$

The throughput of the adsorption compressor depends on the difference between uptakes (C_b - C_d). 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_H is given by the solution of

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

This is represented by state E in Fig. 3. The isotherm equations for CO₂ adsorption by Maxsorb III will be required which are taken from [1] and is summarized below:

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

In the above equation, p is in bar, T in K, and R is the characteristic gas constant for CO₂ (0.1889 kJ/kg K). The above equation is a refit of experimental data for adsorption of CO₂ 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 $(C_b - C_{a'}) < (C_b - C_d)$. 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

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

The uptake efficiency will be zero when the lines d-a' and b-c' coincide which can happen when the upper cycle pressure is at state F in Fig. 3. Thus, the upper limit with void volume effect taken into account and when equilibrium states are possible will be the solution of eq (3) when $\eta_u=0$. But, the thermal compressor fails to provide any throughput. For a realistic

process $\eta_u > 0$, implying that p_H values should be even smaller than that derived from this criterion. It is not difficult to visualise that upper cycle pressure limits with thermal compression will depend on the lower cycle pressures, the effective packing density one can obtain for the adsorbent. The latter is typically 300-600 kg/m³ for highly activated carbon powders. ρ_{ch} is the solid carbon density ~2200 kg/m³. Srinivasan et al.[8] propose that $\eta_u=0.75$ should be targeted because equilibrium conditions are seldom possible in a practical adsorber due to slow kinetics of adsorption and desorption processes. Consequently, eq (3) gets modified as follows

$$C(p_H, T_{H1}) = C(p_L, T_L) - \frac{[\rho(p_H, T_{H1}) - \rho(p_L, T_L)]}{(1 - \eta_u)} \left(\frac{1}{\rho_{eff}} - \frac{1}{\rho_{ch}} \right) \quad (5)$$

Solution of the above equation yields the practical upper limit of p_H for a given T_{H1} .

3. Enthalpy enhanced thermal compression Brayton cycle

If cycle upper operating pressures can be enhanced, a Brayton cycle output also increases. In any case it is apparent that p_H should be less than pressures listed under column $\eta_u=0$ in Table 1.

Table 1 High side operating pressure limits of CO₂ for various criteria with Maxsorb III as adsorbent in a thermal compressor

p_L (bar)	p_H (bar)			
	Ideal	$\eta_u=0$	$\eta_u=0.7$	$s(p_H, T_{H1})=s(p_L, T_L)$
10	39.9	31.8	24.4	23.6
15	59.9	45.3	34.3	34.9
20	79.9	57.6	43.3	45.8
25	99.9	69.0	51.8	56.6
30	120.0	79.7	59.8	67.2

Whether or not even those pressures before expansion lead to optimization of the cycle will be explored further.

The specific work of expansion is given by

$$w = \eta_s [h(p_H, T_{H2}) - h\{p_L, s(p_H, T_{H2})\}] \quad (6)$$

In the above equation η_s the isentropic efficiency of the expander.

If the cycle designed to generate a power of P , the mass flow rate required will be

$$\dot{m} = P/w \quad (7)$$

Unlike a mechanical compression, an adsorption compression process is a slow process because of kinetics of adsorption and slow heat transfer in activated carbon beds. As a result, to maintain a steady flow more than one adsorber will be required as shown in Fig.1. If τ is the time required for the adsorption process to be complete, an adsorber must be able to have an uptake of $\dot{m}\tau$. In order to realise it, the amount of adsorbent that needs to be packed into an adsorber is given by

$$m_{ch} = \frac{\dot{m}\tau}{[C(p_L, T_L) - C(p_H, T_{H1})]\eta_u} \quad (8)$$

Using eq (4) in eq (8) the amount of adsorbent required will be

$$m_{ch} = \frac{\dot{m}\tau(1 - \eta_u)}{\eta_u [\rho(p_H, T_{H1}) - \rho(p_L, T_L)] \left(\frac{1}{\rho_{eff}} - \frac{1}{\rho_{ch}} \right)} \quad (9)$$

The heating inventory for the source at T_{H1} comprises of three components: i) thermal energy required to change the enthalpy of CO_2 from state (p_L, T_L) to (p_H, T_{H1}) , ii) that required to raise the temperature of the adsorbent from T_L to T_H and iii) heat of adsorption. The second and third components are parasitic. There could be other parasitic components, such as the heating of the adsorber body, the heat exchangers which will be neglected in further analysis

as they form only a small fraction of the total [7]. The net thermal inventory for thermal compression can be written as

$$Q_{TH1} = \dot{m} \tau [h(p_H, T_{H1}) - h(p_L, T_L)] + m_{ch} c_{pch}(T_{H1} - T_L) + \dot{m} \tau \Delta h_{st} \quad (10)$$

The last term on the right hand side has the isosteric heat of adsorption which can be derived from isotherm equations and is assumed to be constant in the Toth isotherm used in this analysis. It is 463 kJ/kg as given in eq (3).

To raise the enthalpy of CO₂ from state 2 to 3 (in Fig.2), the thermal energy required is

$$Q_{TH2} = \dot{m} \tau [h(p_H, T_{H2}) - h(p_H, T_{H1})] \quad (11)$$

Since thermal energies at two different temperatures are involved as per eqs (10) and (11), it will not be appropriate to add them together in the calculation of efficiencies. To obviate this, we propose to use the physical exergy of the two heat quantities with T_L as the reference temperature. As a result, total exergy equivalent supplied to the cycle will be

$$E_{net} = Q_{TH1} \left(1 - \frac{T_L}{T_{H1}} \right) + Q_{TH2} \left(1 - \frac{T_L}{T_{H2}} \right) \quad (12)$$

Normally, the performance indicators would be the specific work output based on unit mass of the working fluid and the thermal efficiency. However, in the present thermal compression based Brayton cycle with two heat sources from different temperatures, we propose the following performance indicators:

a) specific work output per unit mass of the adsorbent used w ($C_b - C_a$) which can be summarized as

$$\Delta h_{ch} = \eta_s [h(p_H, T_{H2}) - h\{p_L, s(p_H, T_{H2})\}] \eta_u [C(p_L, T_L) - C(p_H, T_{H1})] \quad (13)$$

b) exergetic efficiency defined as

$$\eta_{ex} = \frac{\dot{m} \tau \eta_s [h(p_H, T_{H2}) - h\{p_L, s(p_H, T_{H2})\}]}{Q_{TH1} \left(1 - \frac{T_L}{T_{H1}} \right) + Q_{TH2} \left(1 - \frac{T_L}{T_{H2}} \right)} \quad (14)$$

The specific work output of the cycle (eq (7)) which now is purely that derived from the turbine will monotonically increase with T_{H2} . Further, as this happens, the mass flow rate steadily reduces and so will the amount of CO_2 that needs to be adsorbed, as well as the amount of activated carbon required. Thus, the thermal inventories as per eq (10) will decrease. But the heat requirement from eq (11) will have a different trend because while the mass flow rate is decreasing, the enthalpy added will increase, which will be discussed in the next section.

In the analysis of Dutta et al. [1] there was neither a need for a regenerator nor a heat rejection unit prior to compression because after expansion the fluid was at T_L . Even in the case of enthalpy enhanced Brayton cycle, the regenerator can be eliminated if the temperature after expansion (at state 4 in Fig. 2) is equal to or less than T_{H1} . This temperature is the solution of the following equation

$$s(p_H, T_{H2}) = s(p_L, T_{H1}) \quad (15)$$

With reference to Fig. 1, expansion process 3-4 is drawn to satisfy this condition, from which the pressure p_H (same as p_3 or p_2) will be when $s(p_H, T_{H2}) = s(p_1, T_{H1})$, since $p_L = p_1$. However, a heat rejection unit will still be required to cool the turbine exhaust (T_4 to T_L) before it can be adsorbed in the thermal compressor. As a corollary, for each value of p_H , there will be a value of T_{H2} at which the expander and compressor exit temperatures will be equal.

The upper cycle pressure needs to be optimised between two conflicting criterion. If it is too large, the uptake difference across which the thermal compressor will operate will be small and hence its size will be large and vice versa. This will tend to increase the magnitude of Q_{TH1} , although Q_{TH2} will decline with increase in p_H . It will be necessary to specify p_L , T_{H1} and T_{H2} in order to make a quantitative estimate. In the next section a detailed discussion is provided based on some specified operating conditions.

4. Results and Discussion

Results are presented for a fixed $T_L=35^\circ\text{C}$ and $T_{HI}=100^\circ\text{C}$. p_L is varied from 10 to 30 bar. T_{H2} is varied between 100-300°C. Adsorption is allowed to occur over 600 s. Table 1 lists the values of p_H for various p_L values assuming a conservative $\rho_{eff}=320\text{ kg/m}^3$ [14]. The last column corresponds to the condition used by Dutta et al. [1], that is, no further heat addition after thermal compression. From this table it is evident that expansion ratios that can be used are at best the ratios of pressures in columns 3 and 1. More practically, it will be the ratio of higher of columns 4 or 5 and of 1.

Calculations were performed for a notional net power of 1 kW. Although, since most results are presented per unit mass basis, the performance indicators are expected to remain the same barring the effects of scaling up. The expander isentropic efficiency of 65% was assumed as done by Garg et al. [15].

Fig. 4 shows the relation between expansion ratio and the minimum value for T_{H2} for which the regenerator can be dispensed with. This temperature seems to be nearly independent of p_L values and the following empirical relation is obtained with a regression coefficient of 99.72%:

$$T_{H2}|_{\text{no-regen}} (^\circ\text{C}) = 88.32 \ln(r) + 97.98 \quad (16)$$

The conventional specific work output of the turbine increases with source temperature and expansion ratio for all p_L values as shown in Fig. 5.

Figure 4: Uniformity in dependence of maximum temperature of second heat source on expansion ratio for dispensing with the regenerator

Figure 5: Work output of the expander dependence on expansion ratio

The relations between the expansion ratio and the work output per unit mass of CO₂ is again well represented by a logarithmic curve for each temperature which are given below:

$$\Delta h \text{ (kJ/kg of CO}_2\text{)} = 94.51 \ln(r) + 3.594 \text{ at } 300 \text{ }^\circ\text{C} \quad (17)$$

$$\Delta h \text{ (kJ/kg of CO}_2\text{)} = 74.15 \ln(r) + 3.433 \text{ at } 200 \text{ }^\circ\text{C} \quad (18)$$

$$\Delta h \text{ (kJ/kg of CO}_2\text{)} = 53.786 \ln(r) + 3.596 \text{ at } 100 \text{ }^\circ\text{C} \quad (19)$$

The above empirical correlations have regression coefficients of 99.92%, 99.74% and 98.19% respectively. A large scatter at a lower T_{H2} is due to strong dependence on the magnitude of p_L . The absence of such dependence at high temperatures suggests that whatever may be p_L , the specific work output depends mainly on the expansion ratio, which is invariably true for a Brayton cycle if compressor work is not considered because of large enthalpy change of the working fluid in the expander.

In the case of thermal compression, however, there is a symbiosis of CO₂ and activated carbon notwithstanding the parasitic component of thermal energy required for rising the temperature of the adsorbent. Consequently, in this case, the specific work output based on unit mass of adsorbent also needs to be evaluated. It is also necessary to appreciate that a minimum of two adsorbers will be required for a continuous in-flow to the power block. This will double the adsorbent inventory for continuous power generation. All further references are to kJ of work produced by adsorption thermal compressor cycle using activated carbon in one adsorber.

Figure 6: Expansion ratio dependence of adsorbent mass specific work output of the expander. Vertical lines indicate the trace of maxima

In the Fig. 6 this specific work per unit mass of adsorbet is used where only two extreme cases of p_L are shown so as not to clutter the figure. The variations for other values of p_L will be between these two cases. An interesting feature is that the expansion ratio for maximum adsorbent mass based specific work shifts to lower values as the p_L increases. Further, the sensitivity of expansion ratio is stronger at higher values of p_L than lower ones. This can be attributed to pressure differential across which the adsorbers operate at high p_L values, though the expansion ratio may be the same. At large pressure differentials across the adsorber, the uptake efficiency diminishes necessitating a larger mass of adsorbent despite a decline in the mass flow rate of CO_2 through the expander. Indeed, Fig.6 also shows that beyond a specific expansion ratio, increasing p_L yields diminishing returns. The positive feature is that maximum specific work output does increase with p_L .

An even more intriguing outcome of thermal compression Brayton cycle is captured in Fig. 7, which extracts the influence of expansion ratio on not only the specific work per

unit mass of adsorbent but also the exergetic efficiency defined in eq (14). This figure is plotted for isobars of p_L as against isotherms in previous figures.

Figure 7: Exergetic efficiency and adsorbent mass specific work output of the cycle on the expansion ratio

The following deductions are made: i) maxima in efficiency and specific work output per unit mass of adsorbent occur at different expansion ratios, ii) while the specific work output per unit mass of adsorbent steadily increases with p_L , the occurrence of its maxima shifts to lower expansion ratios, iii) the curvature of the trend at this maximum increases with p_L indicating that its maximum becomes quite sensitive to p_H values used in the cycle. In the case of the efficiency, the monotonic influence of p_L is absent. The maximum efficiency itself seems to occur in the range of p_L of 15-20 bar. Except at $p_L = 10$ bar, at all other pressures the initial increase in efficiency with expansion ratio appears to be nearly independent of p_L . Yet, for all p_L values 15-30 bar there is only a marginal change in the efficiency (24.5 to 25%) and in specific work output based on adsorbent mass (11.4-12.4 kJ/kg of adsorbent). Considering

the primary objective of this Brayton cycle of reducing the operating pressures, even with $p_L=30$ bar, p_H will be limited to 60 bar. This is expected to be a boon despite the complexities introduced by thermal compression.

Figs. 8a and 8b indicates the relative magnitudes of exergy components of thermal inventories for compression and enthalpy enhancement for the expansion ratio corresponding to best exergetic efficiency for each combination (~ 2). The compression component is significantly large at all conditions shown there, although, their ratios seem to decline as p_L and T_{H2} are increased.

Figure 8a: Exergy components of thermal inventories- Thermal compression

Figure 8b: Exergy components of thermal inventories- Enthalpy enhancement

The fact that optimum expansion ratio is different for maximum adsorbent mass based specific work and efficiency, can be more emphatically assimilated on a plane involving these two parameters. This is similar to the analysis of Garg et al. [16] for supercritical CO₂ Brayton cycles. It is shown in Fig. 9 in which the abscissa is eq (6) and the ordinate is eq (14). Each of the lines are loci of p_H for a given p_L and T_{H2} with T_{H2} anchored at 100°C. The maximum efficiency seems to be almost invariant in the range of $15 < p_L < 25$ bar, although the specific work output per unit mass of adsorbent increases steadily. In solar energy driven applications, where the cost of the collector field is quite crucial, one may like to operate at the maximum efficiency state rather than at the maximum specific work. For waste heat driven thermal compression Brayton cycles, perhaps, one would operate for maximum work per unit mass of the adsorbent. Further, even in solar energy systems, considering the cost of highly microporous adsorbents such as Maxsorb III one may prefer to operate at the maximum specific work state because the sacrifice in the efficiency between the two maxima is only ~3%.

Figure 9: Exergetic efficiency vs adsorbent mass specific work output of the cycle

5. Limitations of thermal compression and possible solutions

One should hasten to add that at large power levels, the amount of adsorbent such as Maxsorb III required will be several 100's of kg. For example, in the calculations presented above, ~65 kg of activated carbon will be required per kW at the best possible efficiency of ~25% (which occurs for a $p_L = 15$ bar and $T_{H2} = 300^{\circ}\text{C}$) allowing for 10 min for adsorption/desorption. This will yield a specific work output of ~77 kJ/kg of CO_2 . To achieve

this magnitude of net specific work, a supercritical CO₂ cycle will require a $p_L > 80$ bar and an expansion ratio of about 4 requiring a $p_H > 320$ bar and a heat source at $\sim 300^\circ\text{C}$.

The major challenge is slow heat transfer and adsorption kinetics of activated carbon beds. For example, the thermal conductivity of the activated carbon bed is as low as 0.1 W/m K and its thermal diffusivity is of the order of $10^{-8} \text{ m}^2/\text{s}$ [4]. Saha et al. [12] show that the thermal kinetics for their test bed ranges up to 900 s and adsorption kinetics up to 400 s, the bed size being just $\sim 80 \text{ cm}^3$ and with heating and cooling from outside the bed. To accelerate the kinetics, it will be necessary to embed the heat transfer surfaces inside the adsorbers as proposed by Ismail et al. [17] and adopted by Mitra et al. [18] for silica gel + water system of thermal compression in the sub-atmospheric pressure levels. This could add new dimensions to design of adsorbers.

One obvious result emerging from this study is that, the expansion ratios are still quite small and nearly same as those in conventional Brayton cycles. Indubitably, larger work output can be derived by enhancing it, which is inhibited by the adsorption characteristics of the pair studied here if done in one stage. The exergy component of thermal compression is substantially larger than that for enthalpy enhancement. This trend can be altered favourably by embarking on two stage thermal compression as proposed by Banker et al. [19] for activated carbon + HFC 134a system and Prasad et al. [20] for activated carbon + nitrogen cryocooler. This will reduce the net adsorbent inventory by two means, namely, i) reduced CO₂ flow rate due to augmented expansion ratio and amount to be adsorbed and ii) each stage of the adsorber operating across a smaller pressure differential resulting in not only a high uptake difference but also a high uptake efficiency of each stage. In tandem, all these factors can lead to a compact thermal compression system. There exists another option of two stage hybrid compression consisting of thermal and mechanical compression. This has been proposed for refrigeration in general by Banker et al. [21] and Koyama et al. [22] for CO₂ air

conditioning. Results of these additional compression methods will be published in an ensuing publication.

4. Conclusions

The contingent disadvantage of small net power output of a low pressure Brayton cycle can be overcome by adopting thermal compression of the working fluid. Through an example of CO₂ as a working fluid with activated carbon as adsorbent in thermal compressor, the efficacy of realising pressure rise is established. Merits of agglomerating a low grade heat source such as an evacuated tubular collector for thermal compression and a field of concentrating collectors for enhancement of enthalpy of working fluid before expansion are evaluated. Even a thermal compression based Brayton cycle is associated with optimal compression ratios of the order of only 2. Specific work output based on mass of adsorbent used and efficiency derived from exergy based thermal component are shown to be good performance indicators. This paper also reiterates that cycle operating conditions for maximising these two performance indicators are different. For the example chosen, the optimal low pressure is found to be in 15-20 bar range. Even with a high temperature heat source at only 300°C, the work output is comparable to that obtainable from supercritical CO₂ Brayton cycle operating at higher cycle pressures and temperatures. Despite these positive aspects, the thermal compression model suffers from large adsorbent inventories, slow kinetics and limited range of operating conditions. Adoption of two stage thermal compression could mitigate this lacuna to an extent.

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