Carbon Dioxide based Power Generation in Renewable Energy Systems

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ABSTRACT

After a substantial impact on refrigeration, carbon dioxide (CO₂) is gaining considerable attention as a working fluid for thermal power generation. This can be attributed mainly to its excellent heat transfer properties and compactness of components arising from its high density. It has the merit of being amenable to operation in sub-, trans- or super-critical Brayton cycle modes. However, inhibiting factors are high pressures needed when operated in trans- or supercritical cycles and the work of compression eroding most of the work of expansion in sub-critical cycle operation. Some of the lacunae of CO₂ such as high work of compression can be alleviated by using non-mechanical means such as thermal compression using the adsorption technique either for partial compression in high pressure Brayton cycles or for total compression in low pressure cycles. CO₂ has also been proposed as an additive to flammable hydrocarbons such that their flammability can be suppressed and yet retaining their other desirable thermodynamic qualities. This review explores the potential and limitations of thermodynamic

cycles where either CO₂ is used alone or as a component in mixture of working fluids. *Inter alia*, it also highlights the issues of regulation of load management using the efficiency-specific power output plane. When used as a blending component, pinch point in the regenerators affects the cycle performance. The objective is to identify research and developmental challenges involving CO₂ as a working fluid specifically for solar power generation.

Key words: Solar thermal power, supercritical CO₂, Brayton cycle, Rankine cycle, thermal compression, mixtures with CO₂, heat exchangers

1. Introduction

Productivity whether it be agricultural, industrial or commercial is given enormous importance in developing countries as a path to prosperity. Availability of high grade energy, and specifically electrical power is intricately interlinked to productivity. While demand for the former continues to grow at a very rapid rate, environmental issues are inhibiting large scale (at several 100's of MW level) generation of electricity using commonly opted resources such as hydro-, fossil fuel or nuclear. As a result, more governments are turning to renewable energy sources for electricity. Among them, solar energy is the most sought after. The quickest way of electrifying has been through photovoltaics, though, long term sustainability of this option is also to be weighed *vis-a-vis* use of materials like tellurium, indium and cadmium which are in limited supply and some of them may be associated with health hazards [1]. Comparatively, solar thermal based electricity generation has only a limited penetration. Most of them use concentrated solar energy for generating steam with the rest of the processes being as in a Rankine cycle [2]. Organic Rankine cycle (ORC) power plants, with working fluids other than water, powered by waste heat or solar thermal energy have also been explored at a few 10's MW level and there are several in operation too [3]. One of the main concerns in ORC power plants is the direct greenhouse gas emission arising out of some of the working fluids (such as R-245fa) which have a high global warming potential (GWP). Environmental impact of solar technologies are also often assessed because there are several issues that need to be considered to establish their viability [4]. A notable feature of solar thermal electricity is that no exotic materials are required in their hardware.

For servicing settlements which are quite far from conventional electrical grid, distributed localized power generation is quite attractive because it obviates losses occurring from long

distance transmission followed by local distribution of electricity. Steam based solar thermal power plants become unviable at 100's of kW or smaller levels. ORC power plants need to be operated with working fluids which are either flammable such as propane or toxic such as ammonia if GWP issue has to be addressed. The other option is to search for alternate cycles, eg: Brayton. Air is the most sought after working fluid which needs high temperatures –such as those obtainable from combustion of natural gas. With solar energy as the thermal source where temperature levels are at best 800°C, a large fraction of the turbine work needs to be allocated to the compressor. It is a common knowledge from thermodynamics that to realize a large energy output, one requires a large enthalpy change of the working fluid which can be either brought about by pressure or temperature changes (or both) with temperature effect being much stronger than the pressure effect. It is apparent that the engineering community is challenged to come up with cycles and working fluids for generating solar thermal electricity using totally environmentally friendly disposition.

On the other hand, looking at the end user of energy, among which is refrigeration and air conditioning industry, natural working fluids are in the lime light with carbon dioxide (CO₂) emerging as a good choice from every indicator of environmental performance [5]. One must hasten to add that even this sector has been cautious in promulgating this fluid because of its high operating pressure requirements and also higher work of compression compared to halocarbon refrigerants in vogue. More notably, the inhibition arises from transcritical operation of refrigeration cycle and the lack of trained manpower in this area. It is logical to infer that, if CO₂ consumes more power during compression than other working fluids, it should also deliver more power while expanding in the same region. Incidentally, CO₂ as a Rankine cycle working fluid has been in the reckoning of nuclear power industry for over a decade [6-9]. More recently,

that option now seems to be applicable even in the renewable energy sector because of the possibility of generating concentrated solar power at temperature in excess of 500°C [10-14]. This paper reviews the work that has been in progress in this area and presents an overview of prospects and limitations of this option. How the lacuna of high operating pressures can be overcome by using low pressures and thermal compression is also investigated. Another possibility of lowering the upper cycle pressures and simultaneously elevating condensing temperatures by blending CO₂ with flammable hydrocarbons is presented. The latter option takes into account the requirement of suppressing the flammability of hydrocarbons and yet retaining their positive thermodynamic features. While the general focus of this paper is on thermodynamic aspect, the fall outs of some of promising choices from the heat transfer point of view are also highlighted.

2. CO₂ as a Standalone Working Fluid in Brayton Cycle

CO₂ as a working fluid has a wide domain of operation whether it be for refrigeration or power generation. For refrigeration, it is in vogue as a primary working fluid in sub-critical and transcritical zones. It is also used as a secondary refrigerant in supermarket applications. In view of peculiarities of this refrigerant, Srinivasan et al. [15] proposed the temperature-exergy plane for obtaining a better perspective of the cycle than the conventional pressure-enthalpy plane. In the area of power generation also CO₂ has a very wide operational domain, namely in sub-, trans- and super-critical pressure ranges as Brayton cycle working fluid or even as transcritical Rankine cycle where the heat sink temperatures can be lower than its critical point of about 31°C. Garg et al. [16, 17] have used representation of these cycles on pressure-enthalpy and temperature-

entropy planes. Fig. 1 shows such possibilities. Srinivasan et al. [18] proposed a two intensive property coordinate system of sound velocity and pressure, the basic premise being that useful processes (in this case expansion) would occur in the decreasing direction of sound velocity. Accordingly, an alternate plane for representing these cycles is shown in Fig.2. These representative cycles are merely chosen to show a qualitative comparison without any reference to whether or not they are optimal. The direction of decreasing sound velocity during expansion is evident for all the three modes of CO₂ Brayton cycles. However, a peculiarity of the supercritical cycle is that the isobaric heat addition process seems to be in the decreasing direction of sound velocity unlike in the other two cases. It also arises because of the compression isentrope being much steeper than the expansion one on this plane. This is also the case with the pressure-enthalpy plane. Further, there is an overshoot which is due to the sound velocity going through a minimum along some of the isobars as shown in Fig. 3. A generalized sound velocity-pressure plane for CO₂ in the zone of interest is shown in Fig.4. In Brayton cycles the compression process is viewed as a collateral liability. In fact, Garg et al. [16] show that the ratio of expansion to compression work is as low as 1.5 in the sub-critical region and marginally increases to just over 2 in the transcritical region. However, there is an appreciable reduction in the compressor work in the supercritical cycle *albeit* with congruent penalty of much higher operating pressures.

Another interesting feature of CO₂ Brayton cycles is that the optimal expansion ratios are quite small compared to those in Rankine cycles or high temperature air Brayton cycles. In the case of sub-critical operation it is as low as just over 2 and even in the trans-critical region it is under 3 [16] . However, in the supercritical region it raises up to about 4 for a low pressure of

~83 bar. But this increase in the expansion ratio, which has a symbiotic benefit of enhanced work output, is neutralized by impractical high side operating pressures of 300 bar or higher.

A patent of Osgerby [19] describes a wide spectrum of possibilities from fully supercritical to a combination of supercritical and transcritical condensing cycles. A common feature is two stage compression and expansion. A wholly supercritical operation has been shown with heat source temperatures as high as 1500°C and as low as 520°C. The operating pressures range from critical to 200 bar. One of the embodiments of this patent is a cycle (Fig. 5) where after expansion and regeneration, the flow stream is bifurcated at 8. One stream of the working fluid operates in the Brayton cycle regime (0-9-0) and the other in transcritical Rankine cycle (8-10-11-1-8).

In solar thermal power systems, one aims to work with maximum efficiency because the cost of the energy collection system, namely the collector field, is far higher than the power block. However, CO₂ Brayton cycle operations are laden with deficits of either low expansion ratio, or high operating pressures. The consequence of low expansion ratio compared to conventional gas turbine cycles is that turbine outlet temperatures are significantly large. The enthalpy associated with it is invariably recovered by using a regenerator for preheating the compressor discharge. Perhaps, one could argue that the actual heat input in the solar receiver reduces and hence its size. But, at the time of start still the entire heat has to be given from some source or the other. Garg et al. [16] also analyzed the fraction of heat recovered in the regenerator to the net work output of various cycles. For S-CO₂ cycles it is more than 15% and for other cycles it is somewhat lesser. Indeed, a real picture of the heat appropriation is visualized through entropy generation in various components and it emerges that the heat exchanger systems- the regenerator and the gas precooler- are the largest sources for it [16, 17].

Chacategui et al. [13] elucidate that an S-CO₂ cycle has many advantages when operating at high cycle pressures of 225 bar and source temperatures >1000 K, among them being their superior efficiency compared to supercritical steam cycles. They also assert that if such pressures have been used in steam cycles, there is no reason why it cannot be so for S-CO₂. The added advantage is compact power block due to low specific volume of the latter. Enthalpy of low pressure stream remaining after recovery in the regenerator can further be used by dovetailing with another ORC for which this remaining heat acts as a heat source. A common conclusion of Chacategui et al. [13] and Dostal et al. [7] is that instead of simple Brayton cycle, a recompression cycle delivers a better performance if heat is available in temperature > 500°C.

All the foregoing studies uniformly find that operational readiness is a shortcoming for CO₂ Brayton cycles. Also there is a clear lack of awareness and trained personnel to handle the situations arising in design and operation of these systems, particularly high pressures and temperatures, despite the fact that they are already in vogue with supercritical steam power plants. Another technological issue that needs to be settled is the design of heat transfer equipment which need a good data base of correlations for designers. This is still work in progress although considerable progress has been made to date [7, 13]. Certain amount of trial and error is inevitable which brings in financial constraints.

The seemingly positive aspect of S-CO₂ cycles of very small components because of low specific volumes at which the cycle operates has a flip side of the need for very high rotational speeds of the turbine. Ma and Turchi [20] report that for a power of 300 kW the turbine rotor is just 40 mm in diameter for a flow rate of 3.5 kg/s, but has to rotate at 125,000 rpm. Dostal [6] points out that the compactness of the power block is negated by voluminous heat exchangers in the form of regenerators and gas coolers. Turchi et al. [12] give a good summary of the role of

effectiveness and loop pressure losses in heat exchangers in recuperated Brayton cycles over and above their contribution to cost. The recuperated heat can be many times larger than that absorbed from the primary heat source. Future work is needed to bring combined effects of heat exchanger size and pressure drop into the cycle performance analyses. In order to minimize these deficiencies, printed circuit/ micro-channel heat exchangers are proposed. In such a case, cleanliness of CO₂ and the heat transfer fluid become paramount and it has to be devoid of any contaminants such as oil/suspended particles. But the exact effect can only be quantified in situ. In addition, because of environmental problems of use of cooling towers (eg: legionella regulations), many industries are dispensing with it by switching to air cooled heat exchangers. In the case of CO₂ Brayton cycles, this inducts some other contingencies because the compressor performance is quite sensitive to gas inlet temperature [21], which essentially is the temperature one can achieve that is as close to ambient temperature dictated by diurnal and seasonal fluctuations. Singh et al. [22] further reaffirm the strong interdependence of mass flow rate, compressor operating parameters and power output in S-CO₂ solar power systems. They stress the need for in tandem assessment of three important entities, namely, i) regulation of amount of charge for stable turbine inlet temperature and power output during summer operation, ii) control of compressor inlet conditions for sustained fully supercritical operation and improved net-power output during winter and iii) sizing of the hot and cold-sides of S-CO₂ loop because of variable densities under widely fluctuating dynamic operating conditions.

2.1 Transcritical non-condensing Brayton cycle

Garg et al. [23] propose an option of operating with CO₂ in the Brayton cycle with a subcritical lower cycle pressure. This non-condensing cycle (TN-CO₂) is devoid of the

consequences of high pressures at high temperatures because they are well below 150 bar which is due to a reduction in the optimum expansion ratio as well. However, this benefit accrues at a premium, with a drop in the efficiency which is not too significant. For example, their analysis shows that for a 900 K source temperature, a low side pressure of 50 bar, the optimal high side pressure is 112.5 bar which yields an efficiency of just under 28% whereas for the same source temperature, a S-CO₂ cycle has an upper cycle pressure of ~225 bar and yields an efficiency of 30.5%. It is evident that a small sacrifice of efficiency is worthwhile for reducing the risks associated with operating at double the pressures.

Another feature of transcritical Brayton cycles is their amenability to load regulation. In solar operations, seldom it is possible to secure a constancy of heat source temperature. As this changes, the point of maximum efficiency also changes implying that the corresponding optimal expansion ratios also change. Garg et al. [23] analyzed the performance of both TN-CO₂ and S-CO₂ cycles on efficiency- specific work output plane. This two path function coordinate system is shown to provide a lot of useful information on operational aspects such as load regulation and associated drop in efficiency. It was shown that a TN-CO₂ cycle has a wider range of load regulation adoptability than S-CO₂ cycle at any given temperature, although at the cost of a larger drop in efficiency when operating away from the optimal efficiency point.

2.2 CO₂ Subcritical Brayton cycle with thermal compression

CO₂ as a subcritical Brayton cycle mode is unlikely to make any impact because of low optimal expansion ratios (~2) and the fact that the compression work is more than half of the turbine output even with heat source at 800 K [16]. This demerit can be overcome if compression can be achieved by means other than turbine work. Dutta et al. [24] proposed a novel scheme

wherein the compression is achieved using low grade thermal energy, which in this case can be from a field of flat plate or evacuated tubular collectors. The adsorption route was proposed for this process which is based on the phenomena of pressure and temperature dependent adsorption characteristics of some highly microporous activated carbons for CO₂. They also point out that efficiencies are as low as 4% for CO₂ if the heat source and sink are at 100 and 35°C respectively for both compression and expansion. Optimal pressure ratios are also about 2.3 and are virtually independent of low side operating pressure in the low side pressure range of 2-55 bar. But, if low grade heat source can be used only for thermal compression and another high grade heat source such as from a concentrating field can be used for preheating the compressed CO₂, a complete delinking of compressor and expander work will happen. This will in turn widen opportunity for deriving as much work as possible from the turbine by expanding down to compression inlet pressure. However, because of poor adsorption characteristics of activated carbons for CO₂ at near atmospheric temperature and pressure [25], the low side pressure of CO₂ cycle needs to be boosted to 10-30 bar level. Yet, for single stage thermal compression, there is also a theoretical limit to maximum pressure that can be generated (Fig. 6), according to the condition that

$$C(p_L, T_{ads}) = C(p_H, T_{des}) \quad (1)$$

where C is the uptake (kg/kg).

However, at that p_H value the throughput of the adsorbers will be zero corresponding to uptake efficiency being zero [26]. In the case of Maxsorb III+CO₂ system this pressure is about 40 bar for an adsorption pressure of 10 bar, $t_{ads}=30^\circ\text{C}$ and $t_{des}=100^\circ\text{C}$. The uptake efficiency is defined as the ratio of uptake difference across which an adsorber operates under actual and ideal conditions [27]. The ideal condition is the one under which the effect of void volume is absent and equilibrium conditions exist [27]. To realize a meaningful throughput Srinivasan et al. [28]

recommend that a minimum uptake efficiency of 40% must be targeted. Because of this limitation thermal compression ratio diminishes further and is found to remain fairly constant at about 2.3 [23] which is again comparable to the optimal expansion ratio for a sub-critical conventional Brayton cycle. The thermal efficiency of adsorption compression Brayton cycle against specific work per unit mass of activated carbon required is shown in Fig. 7. In this figure Δh is the specific work per unit mass of CO₂ and ΔC is uptake in kg of CO₂ per kg of activated carbon. This plot is generated for same cycle temperature limits of 35 and 100°C and for $2 < p_L < 55$ bar. However, if larger compression ratios are desired, it is possible to adopt two stage thermal compression. A typical transcritical Brayton cycle with two stage thermal compression is shown in Fig. 8 [29]. Here p_I is an intermediate pressure and an intercooling will be necessary to cool the discharge gas emerging from low stage adsorber at p_I from t_{des} to t_{ads} . The cycle is shown in Fig. 9. A schematic diagram of a Brayton cycle with two-stage thermal compression is shown in Fig.10. However, the draw backs of such a compression is a necessity of a large number of adsorbers and an enormous parasitic heating inventories demanded by the adsorbent, adsorber metal container and the heat of adsorption over and above the change in enthalpy of CO₂. It may be possible to eliminate the regenerator after expansion if the expander downstream temperature is equal to or below the discharge temperature of the compression process which will essentially be the low grade heat source temperature.

Koyama et al. [30] proposed an innovative method of using hybrid compression of thermal and mechanical means for CO₂ for refrigeration applications. This technique can easily be used for sub-critical Brayton cycles as well. A schematic of such a cycle with low stage thermal and high stage mechanical compression is shown in fig.11. It will be an interesting exercise to determine what optimal turbine expansion ratios will be because only a small fraction

of it will be absorbed by the compressor. One has to optimize also the sequence of compression as to which method has to be used in the lower stage. Further, the inter-stage pressure needs to be optimized for maximum net work output of the cycle.

For adsorption based thermal compression, there is an imminent need for experimental isotherm data covering pressures of the order of 10's of MPa and temperatures well in excess of 100°C. Because such an application was not perceived to date, activated carbon + CO₂ adsorption measurements covered only temperatures well below 360 K, although pressure range extended up to 14 MPa [25, 31, 32].

3. CO₂ as a Rankine cycle fluid

A lot of research has occurred with CO₂ as a Rankine cycle working fluid where heat rejection can occur at subcritical temperatures implying ambient temperatures being quite low (well below critical temperature of CO₂ ~31°C). Desired upper cycle operating pressures can also be <100 bar. But in this Rankine cycle, heat addition is in the supercritical region. Angelino and Invernizzi [33] report an analysis of such a cycle with liquefied natural gas (LNG) as the heat sink. This cycle has the potential of covering the vast range from -50 to 600°C with a single fluid. This indeed is a highly viable option in locations endowed with abundant direct solar radiation and where there is a need to convert LNG in to compressed natural gas (CNG). They also report the application of a transcritical condensing cycle for which the heat sink can be the low temperature sea water which has been the topic of ocean thermal energy too.

Yamaguchi et al.'s [34] work covers a range of ~90 bar and 165°C, the latter being achievable in high quality evacuated tubular collectors. Their laboratory simulation tests yielded good results but were not conclusive enough in the absence of a turbine which was replaced by

an expansion valve and a heat exchanger. Zhang et al. [35] confirm that CO₂ is the best Rankine cycle working fluid among ammonia, water and HFC 134a in temperature range up to about 175°C, although in this range even transcritical CO₂ cycle has an efficiency under 12%. For this reason, Chen et al. [36] do not favor CO₂ as a transcritical Rankine cycle working fluid for heat source temperatures less than 433 K. An analogous conclusion is arrived at by Chen and Lundqvist [37] who attribute the weak performance to drastic changes in the properties of CO₂ during the heat addition process. Garg et al. [17] compare irreversibilities in CO₂ and steam Rankine cycles and conclude that at heat source temperatures below 1000 K the CO₂ cycle has a fairly larger irreversibility due to high turbine exhaust temperatures and inevitable use of a regenerator which contributes the maximum irreversibility.

Chen et al. [38] also bring out a salient point that only considering the thermodynamic efficiencies of cycles could be misleading, because the power output and cycle thermal efficiency cannot be maximized at the same point of when utilizing a certain heat source. This is also one of the conclusions of Cayer et al. [39] in a context of a free heat source like industrial waste gases where the focus should be on maximising the specific net output rather than the thermal efficiency. This aspect has been investigated in greater detail by Garg et al. [23]. For the transcritical Rankine cycle the efficiency vs specific net work output plane is shown in Fig. 12. As seen here for heat source temperatures over 700 K, the points of maximum efficiency and maximum specific work occur for high side pressures well over 300 bar. If the cycle pressures are to be limited, then sub-optimal operation becomes inevitable. The working plane of two path functions, namely the cycle efficiency and the specific work output was also used by Chacategui et al. [13]. However, their range of analysis did not extend to pressures where maximum work output and efficiency can be visualized.

All the CO₂ cycles suffer from poor performance under off design conditions [14]. The key role of performance of the compressor is already highlighted. When used in concentrated solar power without energy storage, vagaries in sunshine cause wide fluctuations in turbine inlet temperatures under which both power output and efficiency decline sharply. Garg et al. [23] have mapped it for various operating conditions.

4. CO₂ as a blend with other working fluids

In all the afore mentioned applications of CO₂ as a Brayton cycle working fluid, the negative features are either high operating pressures if used in mechanical compression or large heat inventories if used with thermal compression. Thermal compression also is still conceptual and it is unlikely that in the immediate future any practical power plants will be built. If used as a Rankine cycle fluid, the sink temperatures have to be well below critical point of CO₂. In this context, it is worthwhile exploring CO₂ as a component in mixtures of working fluids. Being a good suppressor of flammability, it can be used along with flammable hydrocarbons which are themselves known to be good working fluids in power generation. Their collateral disadvantage can be obviated by suitably mixing with CO₂. Garg et al. [40] have used this philosophy and evaluated some of the possible mixtures with saturated hydrocarbons such as propane, butane, pentane etc. A similar approach of blending hydrocarbons with some halocarbon refrigerants has also been proposed in the past [41-43]. It is important that all components of the mixture be environmentally friendly. Invernizzi and Stelt [43], however, proposed a mixture of benzene and CO₂. An important contribution to this method is from Li and Ma [44] who found out that most hydrocarbon and CO₂ blends achieve an optimal efficiency at a CO₂ mass fraction of 0.5. Because of non-isothermal condensation the exergetic and thermal efficiency are enhanced. Only butanes were the alkanes covered in their investigation. Garg et al. [40] covered other alkanes

such as propane and isopentane. Their analysis proposes that a 30% mole fraction of CO₂ is adequate to suppress flammability. Such a mixture yields a glide range of slightly over 20°C for propane CO₂ mixtures and 80-100°C for isopentane CO₂ mixtures. The latter could be run as a conventional Rankine cycle with substantial heat addition occurring in the two-phase region and also in the regenerator. In the case of propane CO₂ mixtures the upper pressure isobar happens to be supercritical. Yet, even in the case of these mixtures the expansion ratios are seldom greater than 3 implying that substantial turbine tail side heat recovery needs to be effected in a regenerator. In order to make propane CO₂ mixture as a serious contender, Garg et al. [45] consider an appropriate non-flammable mixture which has a high critical point and also normal boiling point lower than typical ambient temperatures encountered in tropical countries. For example, for a typical condensation temperature of ~35°C, a mole fraction mixture of 48.5 % propane and 51.5 % CO₂ the critical temperature is ~62°C and the corresponding critical pressure is 68.3 bar. However, for a condensing pressure of ~50 bar, the optimum expansion ratio is found to be >6 implying a high side pressure in excess of 300 bar, which is again burdened with the same disadvantage of a S-CO₂ cycle.

5. Heat transfer issues

There will be at least three heat exchangers in the CO₂ Brayton cycle: i) the regenerator where the high pressure streams emerging from the compressor is heated by the hot low pressure exhaust stream of the expander, ii) enthalpy enhancement of the high pressure stream by the concentrated solar heat derived at the receiver and iii) heat rejection in the gas cooler before CO₂ enters the compressor. Since the expansion ratios are quite small, the low pressure stream at expander exit is quite warm (of the order of 500°C), whereas the high pressure stream from the

discharge of compressor is often of the order of 100°C. Thus, there is enormous potential for heat recovery. Because of large pressure dependence of specific heat, the temperature change of the high pressure stream in the regenerator tends to be much smaller than for the low pressure stream. A recent review by Dutta [47] summarises the important issues to be addressed in the design of all these heat exchangers. Several configurations of regenerator have been suggested, the prominent ones being tube-in-tube type [48], printed circuit heat exchangers (PCHE) and microchannel heat exchangers (MCHE) [49]. Considerable amount of work on heat transfer in supercritical CO₂ has been reported mainly for refrigeration gas cooling applications [50]. Whether there are any peculiarities due to combined effects of pressures and temperatures well above the critical point is still a topic of research. Enhancement/curtailment of heat transfer in supercritical CO₂ seems to be a cumulative effect of sharp changes in the thermophysical properties across a particular temperature range. Thermal conductivity, viscosity and density decline sharply around the critical point (~31°C). This should affect the design of the gas cooler upstream of the compressor. The specific heat has a more peculiar behavior. It rises sharply, but the magnitude of enhancement and its steepness reduce as temperature increases. All these effects decrease with increasing supercritical pressure to become insignificant at about one and a half times the critical value, confirming that they are caused by the rapid near critical variation in physical properties [51]. This feature suggests that the unit used for actual enthalpy enhancement of CO₂ before its entry to expander could be designed in the conventional way. Zhang and Yamaguchi [48] observed a substantial enhancement in heat transfer and attributed it to i) thinner thermal boundary layer than in the case of water and ii) the decrease in viscosity and the increase in heat capacity and iii) velocity and temperature fields not reaching a fully developed state. Pioro et al. [52] compared various correlations for supercritical heat transfer and suggest that

most of them can be used for preliminary estimations of heat transfer in tubes and bundles. However, no one correlation is able to describe deteriorated or improved heat transfer in tubes. In another review article Duffey and Pioro [53] aver that there are all three possibilities of heat transfer, namely normal (as occurring in other fluids), a zone where it is worse than usual heat transfer and a zone where it is enhanced due to properties of supercritical CO₂. Flow orientation also seems to make a difference in the magnitude of heat transfer coefficients. In general, there is a distinct possibility of circumferentially non uniform temperatures which are attributable to large variations in density of CO₂ at conditions prevailing in the core and near the wall. Deteriorated conditions seem to be dominant at high heat fluxes and low mass flow rates and turbulence promoters need to be inserted in these zones.

Garg et al. [16] show that a substantial amount of energy is exchanged in the regenerator and the irreversibilities in this unit and final gas cooler are quite large in S-CO₂ cycles. Hence it is imminent that these two units deserve a critical design. Pulsed jet impingement cooling, perhaps, could be considered for these two units. A schematic arrangement is shown in fig. 13.

Pinch point in the regenerator is a limiting issue in general in all Rankine/Brayton cycle power generation methods. The two fluid temperatures approach each other with driving force for heat transfer. Conboy et al. [54] report that this can occur under two general conditions: i) when the hot-stream temperature reaches the inlet temperature of the cold-stream which can happen if the heat exchanger is over designed or at off-design conditions in a counter current heat exchanger if the heat capacities of the two fluids greatly differ as in the case of S-CO₂ case and ii) when one of the fluid streams approaches its critical point (or a phase change), causing large changes in heat capacity, for which, the heat transfer at the location of the pinch point can still be very high, because the local heat transfer coefficient can increase close approach to the

critical point (or phase change). In the case of a regenerator heat capacity of the high pressure CO₂ stream is larger than in the low pressure leg by a factor of 2. Its avoidance is the very reason the S-CO₂ recompression cycle has gained favour over the simple recuperated Brayton cycle.

Important conclusion of Garg et al. [40] is that as the number of carbon atoms in alkanes decreases, their blend with CO₂ moves the pinch point to the colder end of the regenerator. The shift of the pinch point towards the warm end of the regenerator is found to be a major cause of irreversibility and accordingly they suggest use of propane CO₂ mixture as a possible transcritical Rankine cycle fluid if heat source is available at about 300°C. Adoptability of this mixture over a wide range of source temperatures is found to be an added advantage. However, the upper limit has to be <900 K, due to disintegration of propane [46]. A minor advantage of large temperature glides associated with alkanes of larger carbon atoms, despite their association with large irreversibilities, is that the heat recovery can be stretched well in to the two phase dome.

As far as the pressure drop is concerned, it seems to play only a limited role because of the fact that the isotherms at the operating levels are quite steep (p-h plane), such that the changes in enthalpy are extremely small. Even 2-5 bar drops of the low side pressure and up to 10 bar on the high side have marginal effect on the changes in enthalpy. Hence, many designs have focused predominantly on the heat transfer rather than the pressure drop aspects. The work of Suzuki et al. [55] shows that when PCHE is used, discontinuous S-shaped fins in flow paths provide a fivefold reduction in the pressure drop without compromising the heat transfer. They ascribe this feature to a superior uniform velocity profile in the flow area and elimination of reverse flows and eddies that occur around bends corners of zigzag flow channels in conventional PCHE. Perhaps, this configuration can be used in the channels shown in fig. 13.

6. Concluding remarks

In this paper range of possibilities of using CO₂ as a working fluid for solar thermal power generation have been reviewed. Although, in complete supercritical state it has several benefits, inhibiting factors are high operating pressures and need of higher source temperatures. In the transcritical non-condensing mode, with some sacrifice in efficiency more amenable turbine inlet pressures can be realized. Complete sub-critical operation is impractical since the work of expansion is virtually consumed by the compressor. But if thermal compression is adopted in two stages or hybrid compression of mechanical and thermal means is resorted to, a subcritical cycle could be feasible. However, this needs to be supported through extensive data on CO₂ adsorption on adsorbents such as activated carbon. As a transcritical Rankine cycle working fluid, CO₂ needs condensing temperatures below about 20°C which makes it a difficult choice in tropical regions. However, positive features of CO₂ can still be used as a component in a mixture with hydrocarbons. Here again, non-isothermal phase change leads to some design limitations, which are possible to be accommodated. A good design of all the heat exchangers is paramount to successful implementation of the cycles.

Acknowledgement

This research is based upon work supported by the Solar Energy Research Institute for India and the U.S. (SERIUS) funded jointly by the U.S. Department of Energy subcontract DE AC36-08G028308 (Office of Science, Office of Basic Energy Sciences, and Energy Efficiency and Renewable Energy, Solar Energy Technology Program, with support from the Office of International Affairs) and the Government of India subcontract IUSSTF/JCERDC-SERIUS/2012 dated 22nd Nov. 2012. In addition, the authors would also like to thank

Professors J.A.White and S. Velasco of University of Salamanca, Spain for helping with the development of sound velocity- pressure plane of Brayton cycles for CO₂.

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