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Carbon Dioxide As Working Fluid for Power Generation and Refrigeration

Thursday, 10 March 2022 | Technical Topic Webinar

Presented By

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CRICOS Provider Number: 03567C | Higher Education Provider Number: 14008 | RTO Provider Number: 51971

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Agenda

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1	Welcome and Introduction
2	Principle Properties of **CO_2**
3	**CO_2** as a Working Fluid in Closed Cycle Power Generation
4	**CO_2** as a Working Fluid in Vapour Compression Refrigeration
5	Conclusion and Q&A

Introduction - Presenter

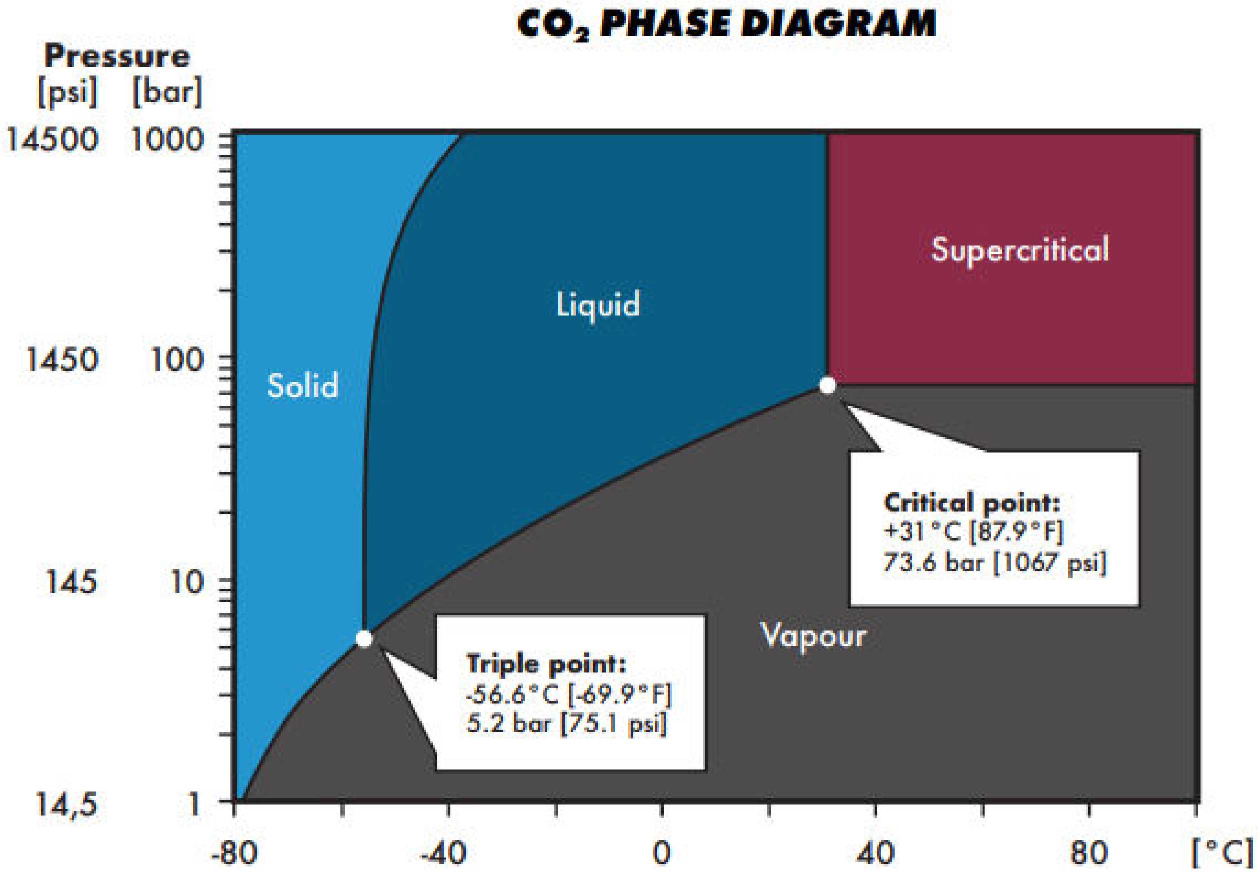
Professor Kandadai Srinivasan – EIT Lecturer

- Graduated in mechanical engineering.
- Obtained MS through research from IIT, Madras
- Diploma of Imperial College, London
- PhD from ANU
- Fellow of Engineers Australia, CPEng, NER
- Fellow, ASME
- Near 5 decades of experience in teaching at most renowned institutions across the world, 5 years of industrial experience.
- Specialising in refrigeration, air conditioning, solar energy.
- Numerous consultance projects on energy conservation
- 138 high impact journal publications
- near 4000 citations and h index of 36.

CO_2 as a working fluid for power generation and refrigeration

- CO_2 is a natural fluid, non toxic and non flammable
- A single “green” fluid for both applications
- Simple system but somewhat higher operating pressures than normal
- Energy efficient, especially in cold climates
- Small charge inventories
- Reduced pipe diameters due to high densities, so low installation cost
- Technically the best solution available today, given the furore on climate change, global warming
- A lot of research and development is needed for improvements and standardization.

Principal Properties of CO_2

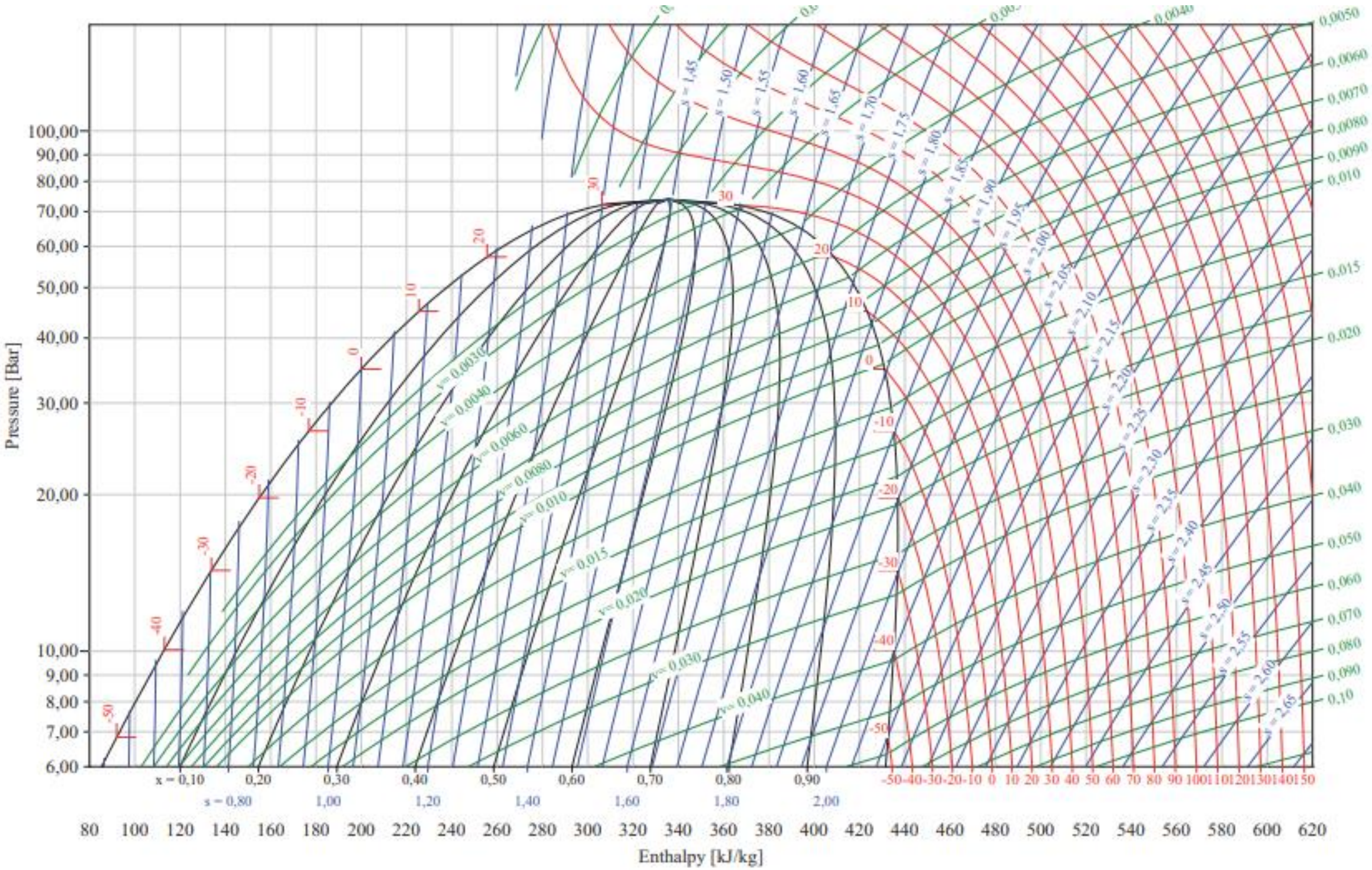


From: World Guide to trans-critical CO2 Refrigeration, Shecco Market Development 2020.

Data from REFPROP

Critical temperature	30.98°C
Critical pressure	73.773 bar
Critical density	467.6 kg/m^3
Triple point temperature	-56.56°C
Triple point pressure	5.176 bar

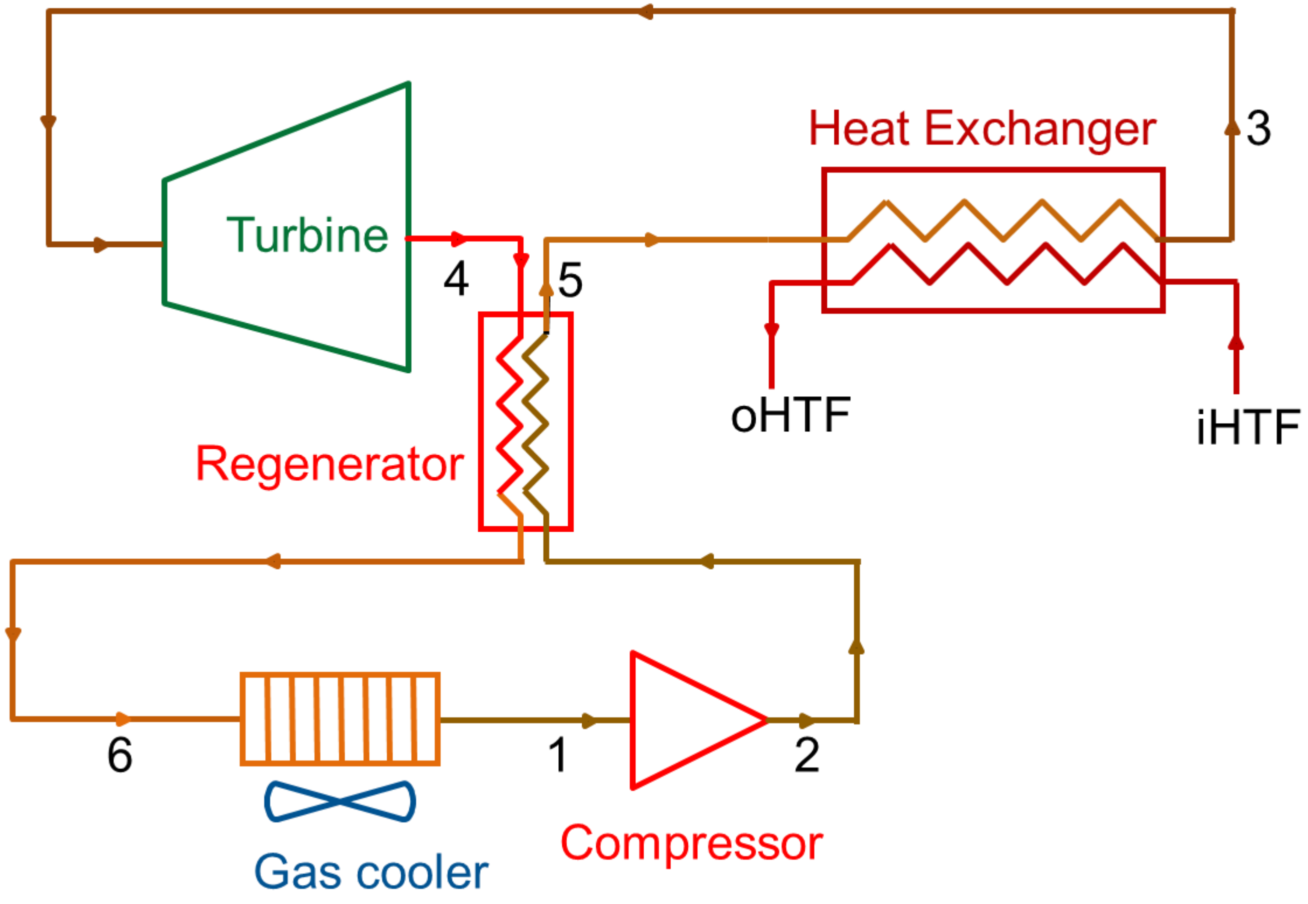
Pressure-Enthalpy Diagram of CO_2



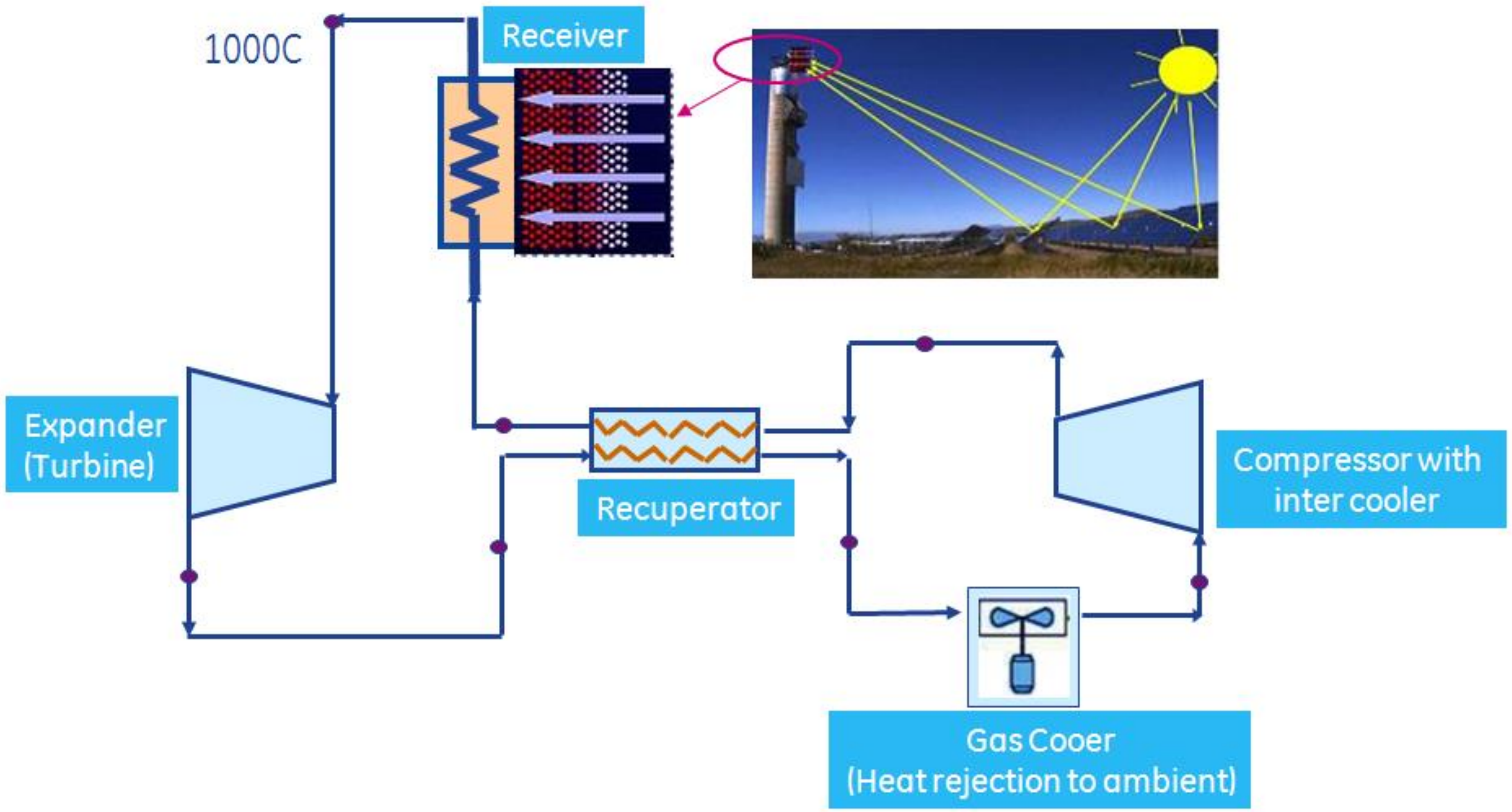
From brochure of Semi-hermetic compressors for R744 (CO2), trans-critical applications of Frascold

CO_2 as a Working Fluid in Closed Cycle Power Generation

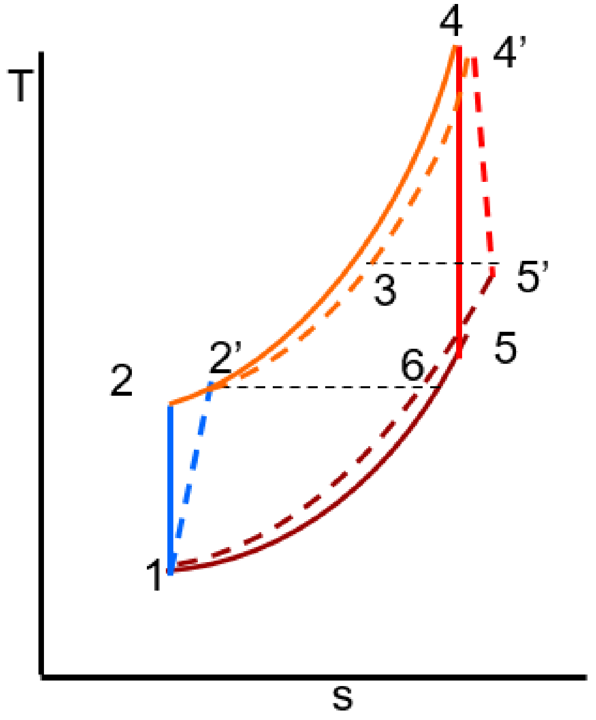
Simple Brayton Cycle with Regenerator



Closed loop CO_2 Brayton Cycle

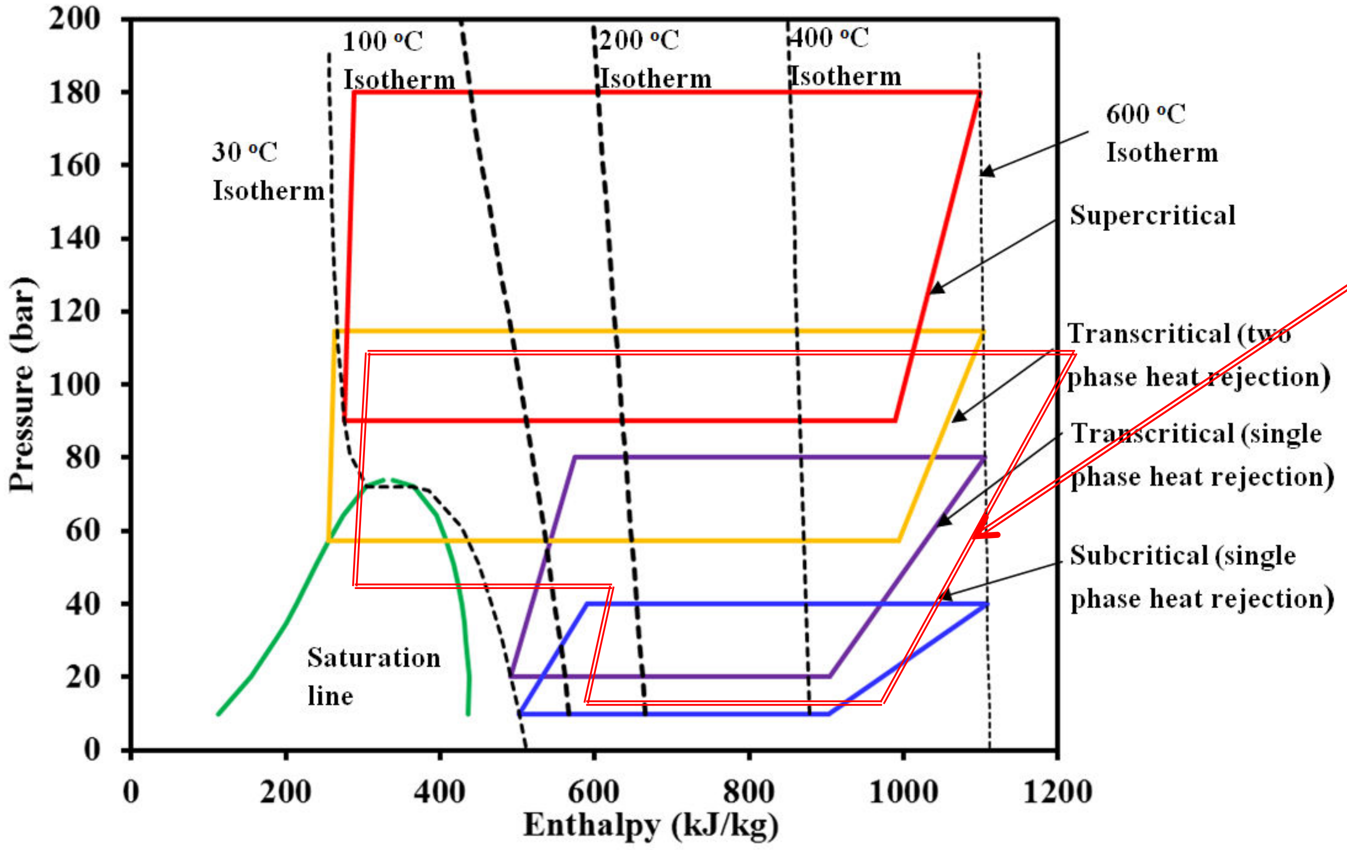


T-s diagram of Brayton Cycle



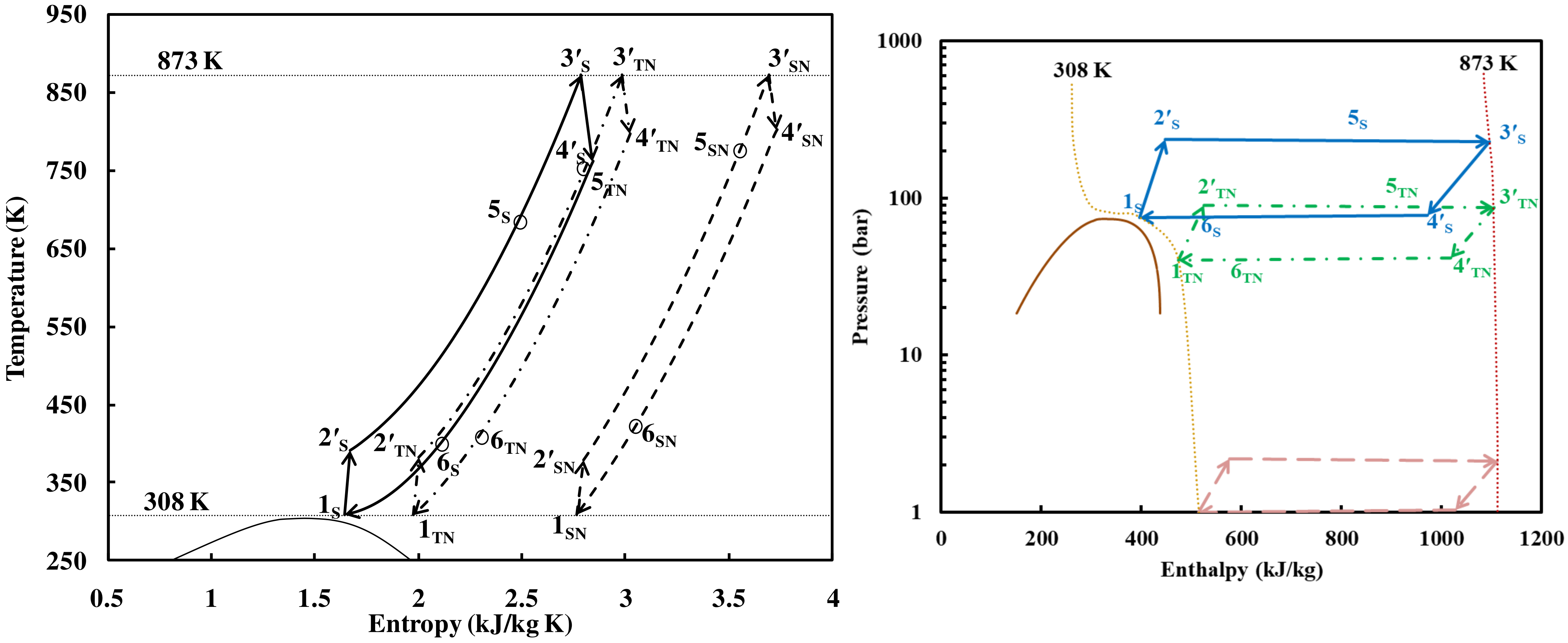
Full line – ideal cycle
Dashed line- actual cycle

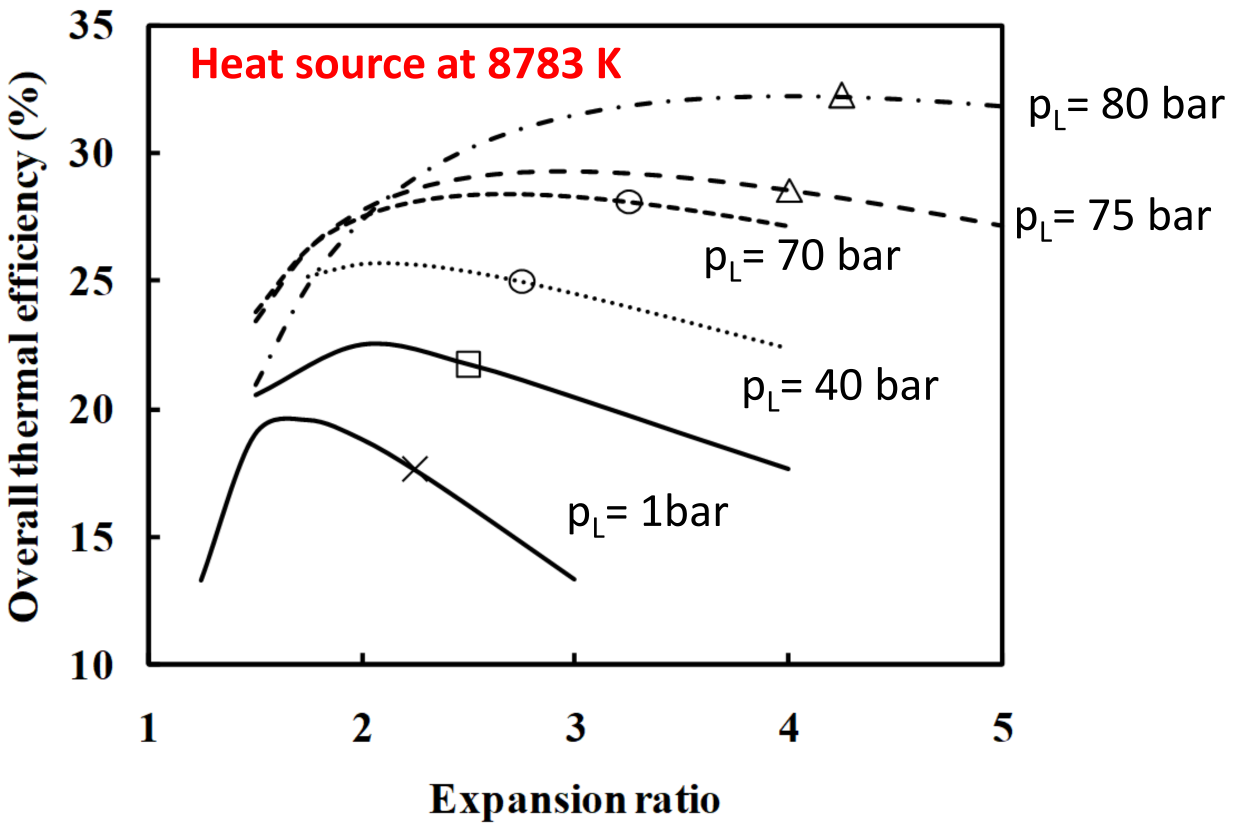
Options available with CO_2



Combination of trans-criticals

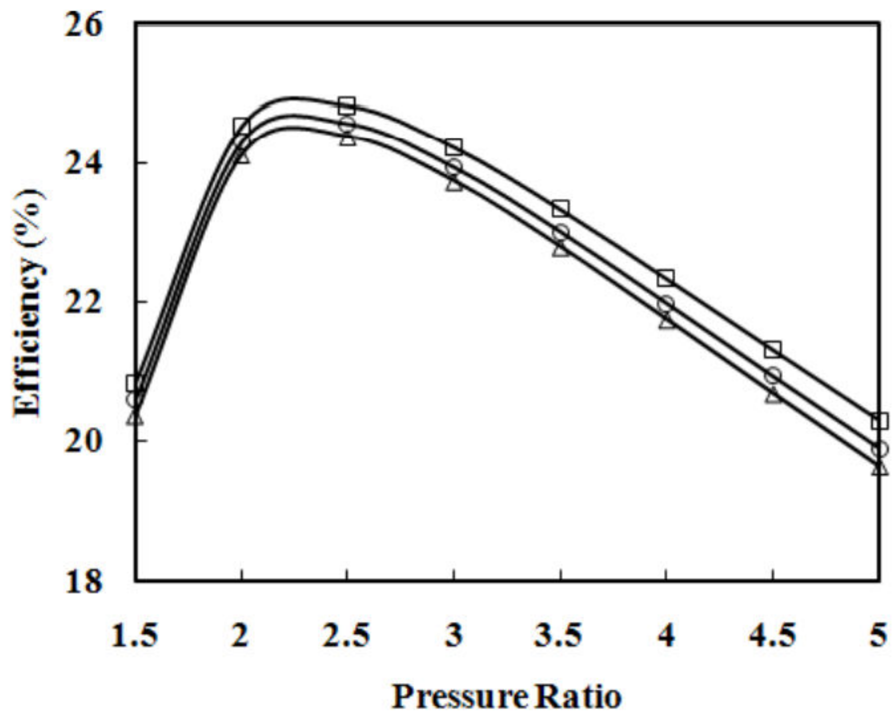
Simple supercritical CO_2 Brayton Cycle



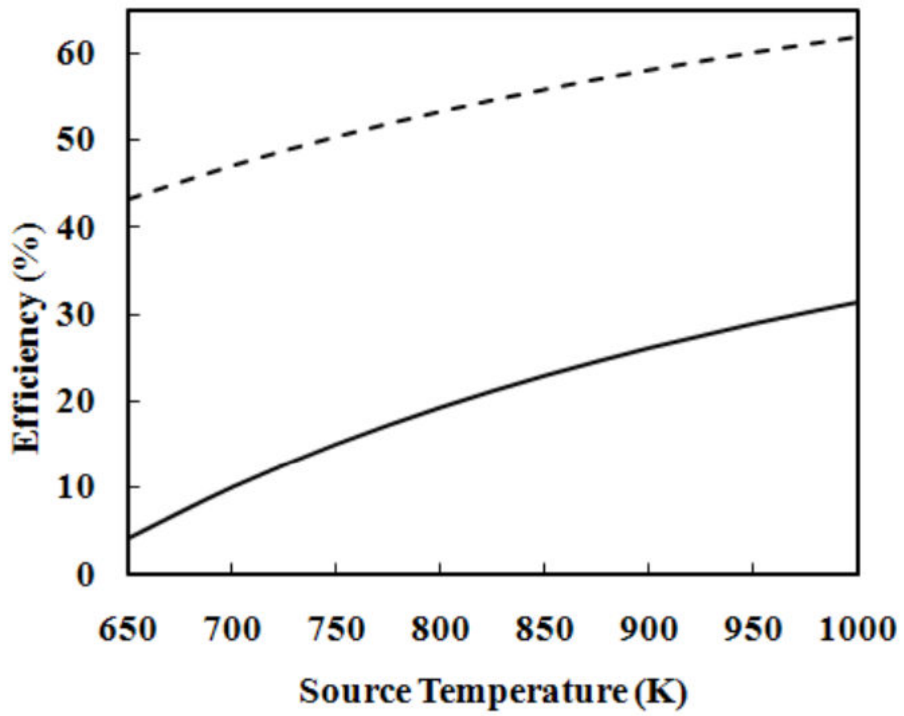


Heat source at 8783 K
Overall thermal efficiency (%)
35
30
25
20
15
10
1
2
3
4
5
Expansion ratio
p_L= 80 bar
p_L= 75 bar
p_L= 70 bar
p_L= 40 bar
p_L= 1bar

Some performance indicators (sub-critical)

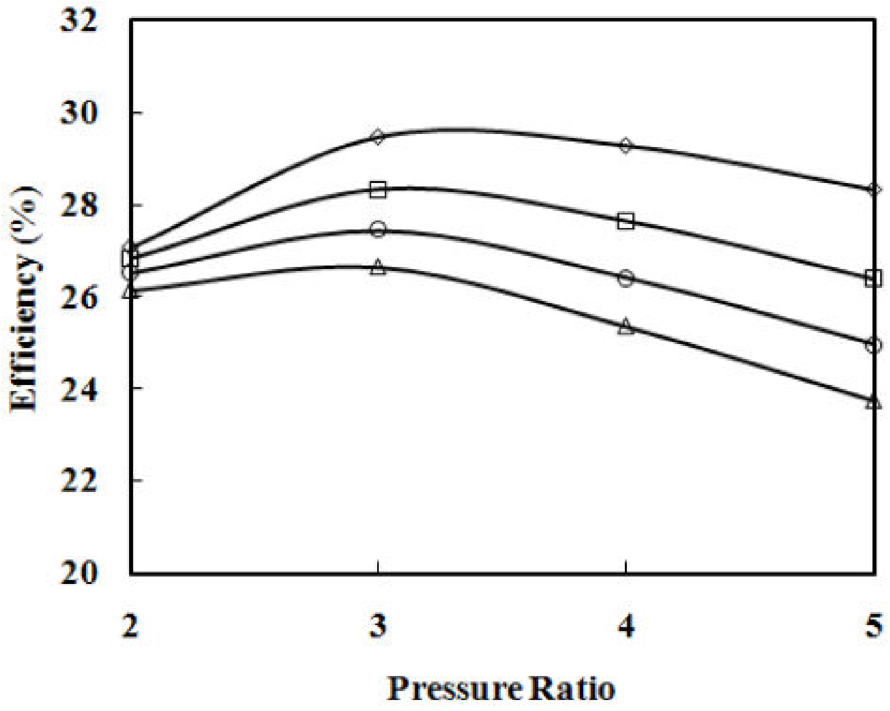


Real cycle efficiency v/s pressure ratio at fixed source temperature of 873 K Δ =1bar, ○ =5bar, □ =9bar

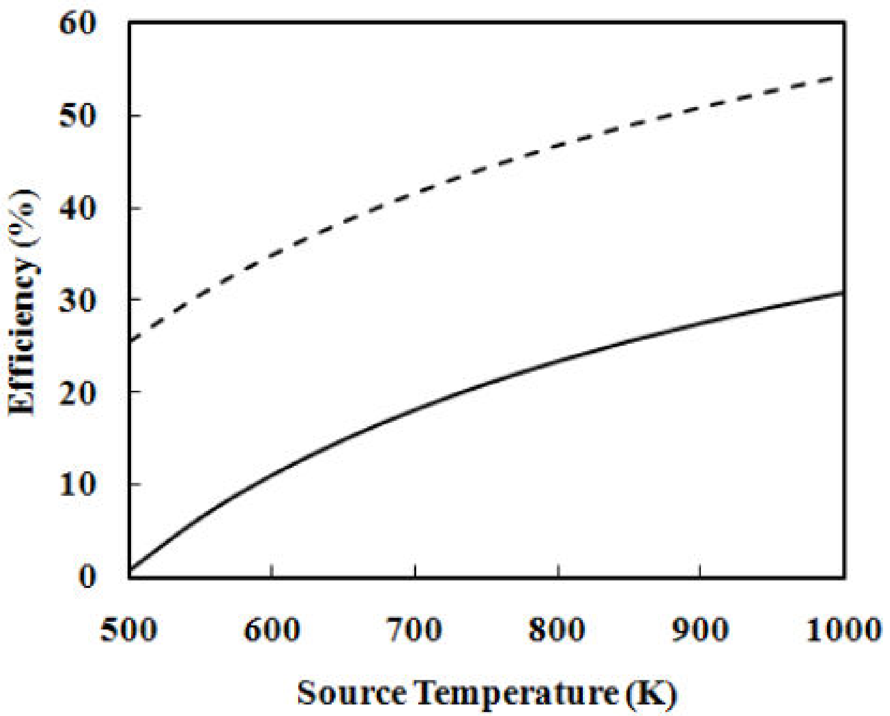


Comparison between ideal and real cycle efficiency for pressure ratio of 2.3 and low side pressure of 1bar at different source temperatures.

Some performance indicators (trans-critical)

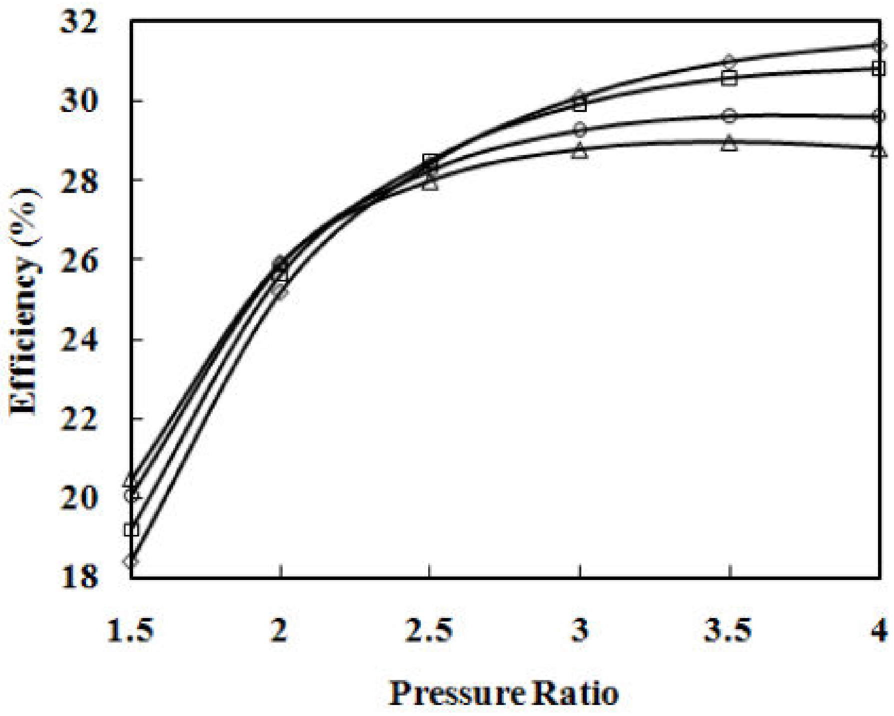


Real cycle efficiency v/s pressure ratio at fixed source temperature of 873 K. Δ = 40 bar, ○ = 50 bar, □ = 60 bar, ◊ = 70 bar

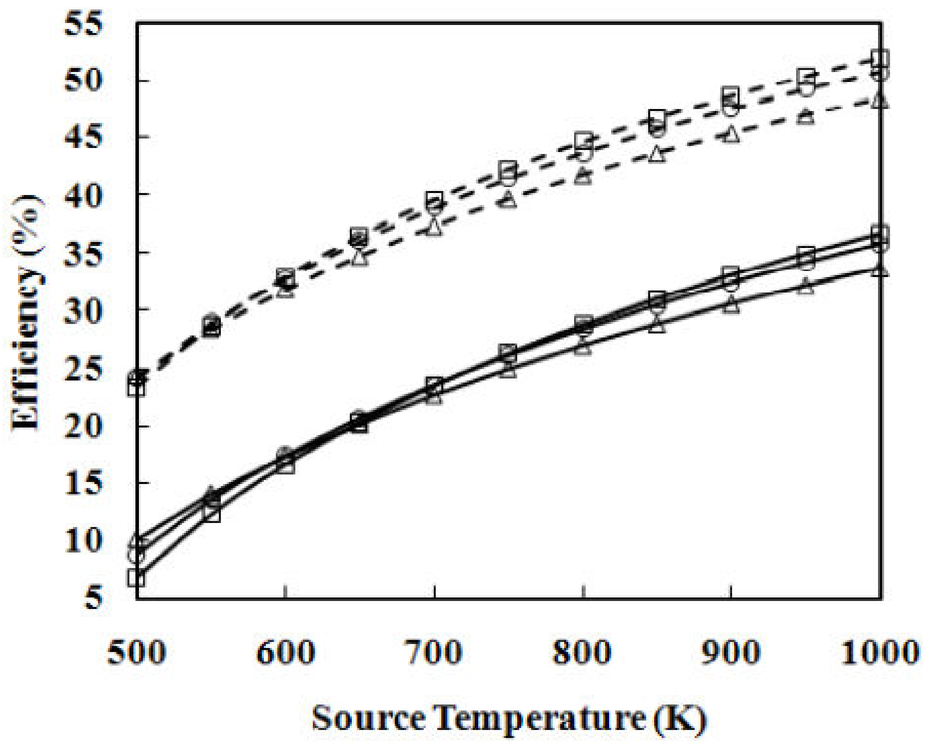


Comparison between ideal and real cycle efficiency for pressure ratio of 2.3 and condenser pressure of 50 bar at different source temperatures

Some performance indicators (supercritical)

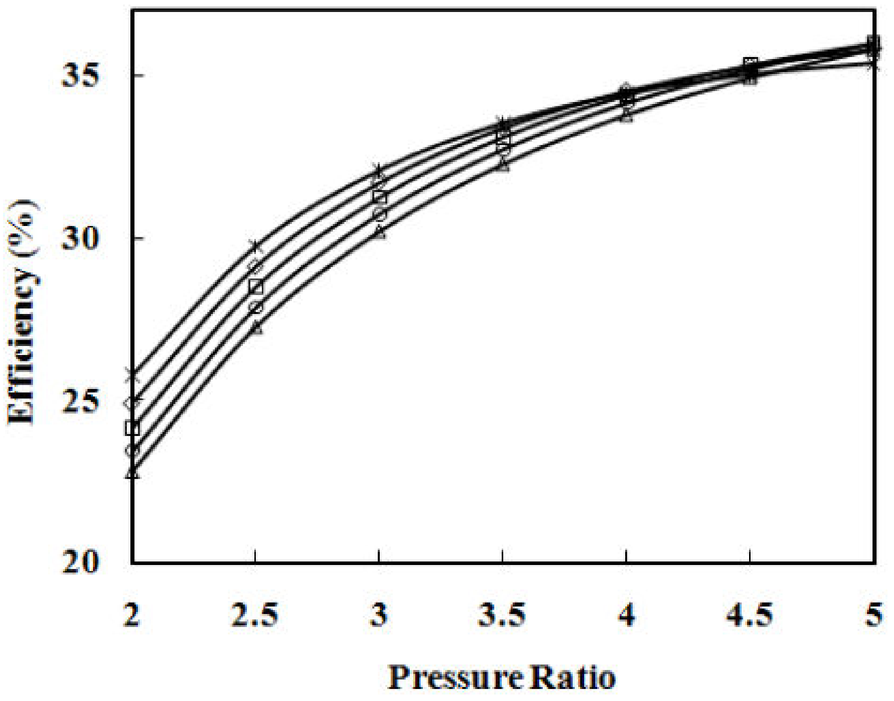


Real cycle efficiency v/s pressure ratio at fixed source temperature of 873 K. Δ = 75 bar, ○ = 77 bar, □ = 79 bar, ◊ = 81 bar

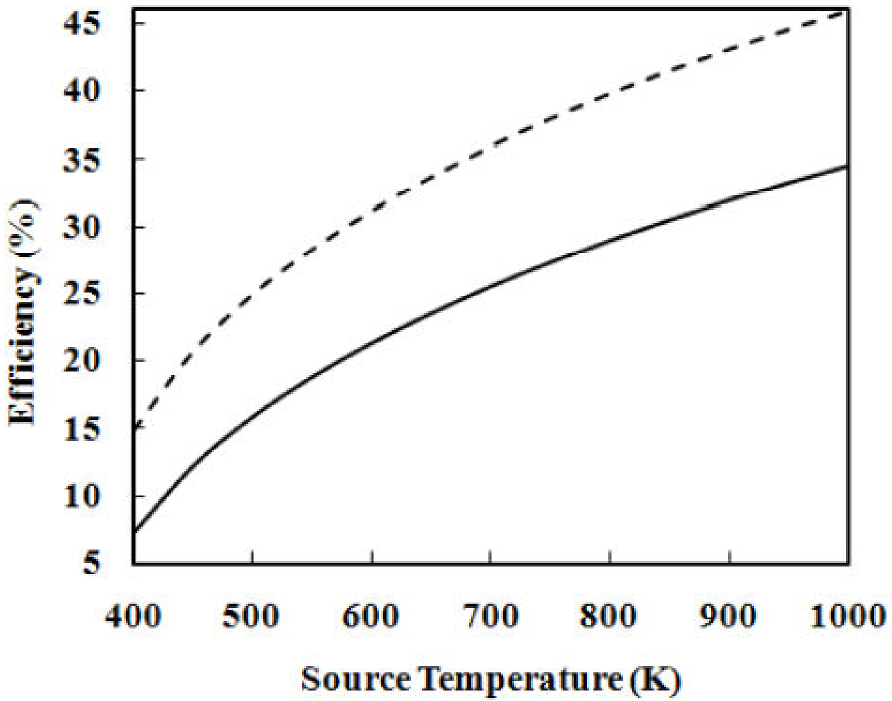


Comparison between ideal and real cycle efficiency for pressure ratio of 2.3 and condenser pressure of 50 bar at different source temperatures

Some performance indicators (Brayton-Rankine)

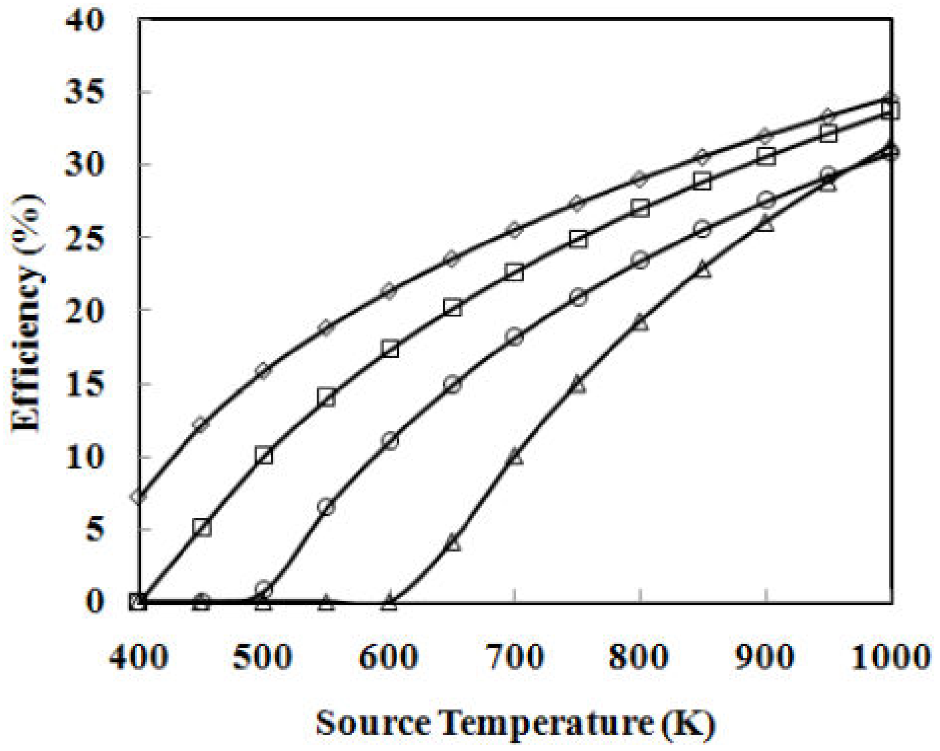


Real cycle efficiency v/s pressure ratio at fixed source temperature of 873 K for different condenser temperature Δ = 280 K, ○ = 285 K, □ = 290 K, ◊ = 295 K, × = 300 K

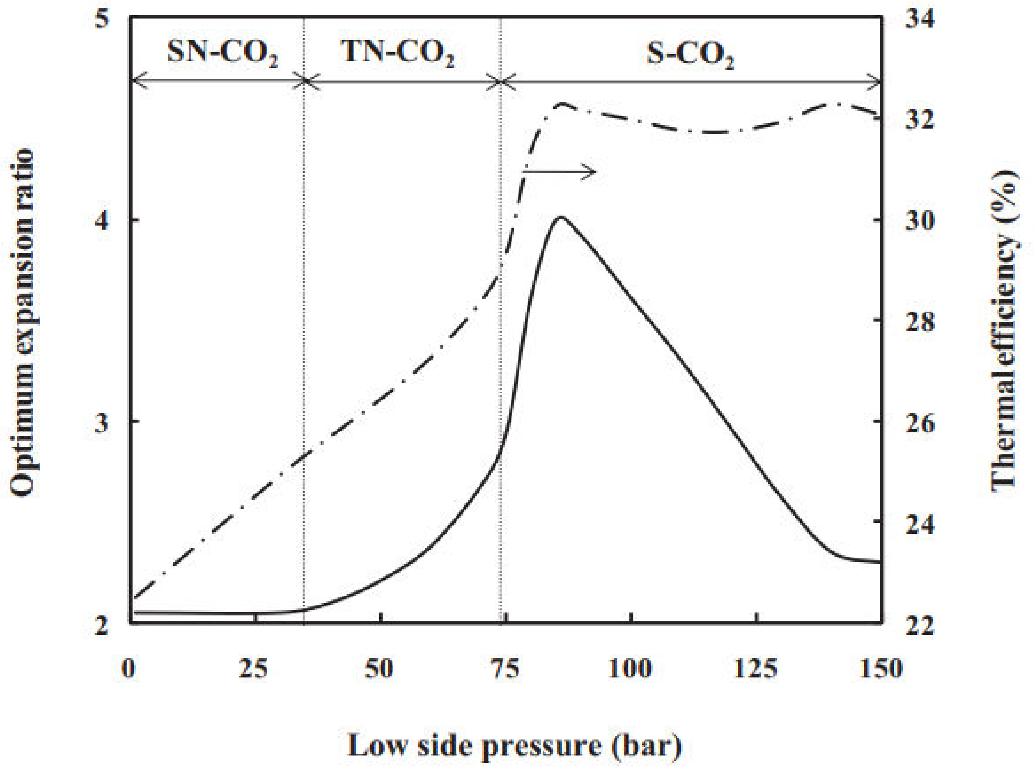


Comparison between ideal and real cycle efficiency for pressure ratio of 2.5 and condenser pressure of 67.13bar at different source temperatures.

Comparison of all the four possibilities under their optimal conditions

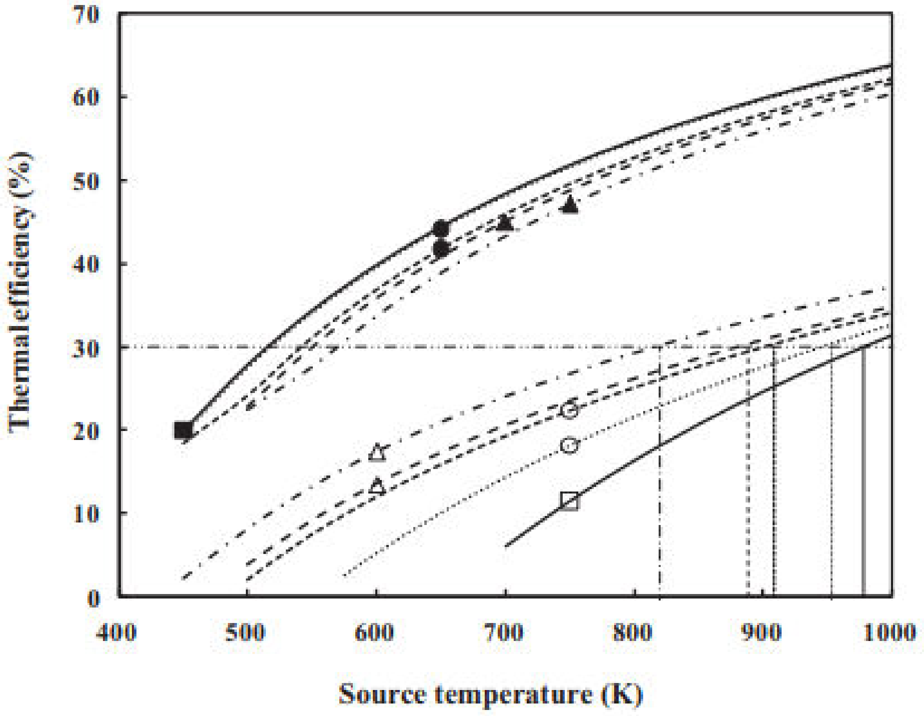


Real cycle efficiency *v/s* at fixed source temperature of 873 K for the optimized configurations of different possibilities of CO2 cycle Δ subcritical, ○ trans-critical, □ supercritical, ◊ trans-critical condensing

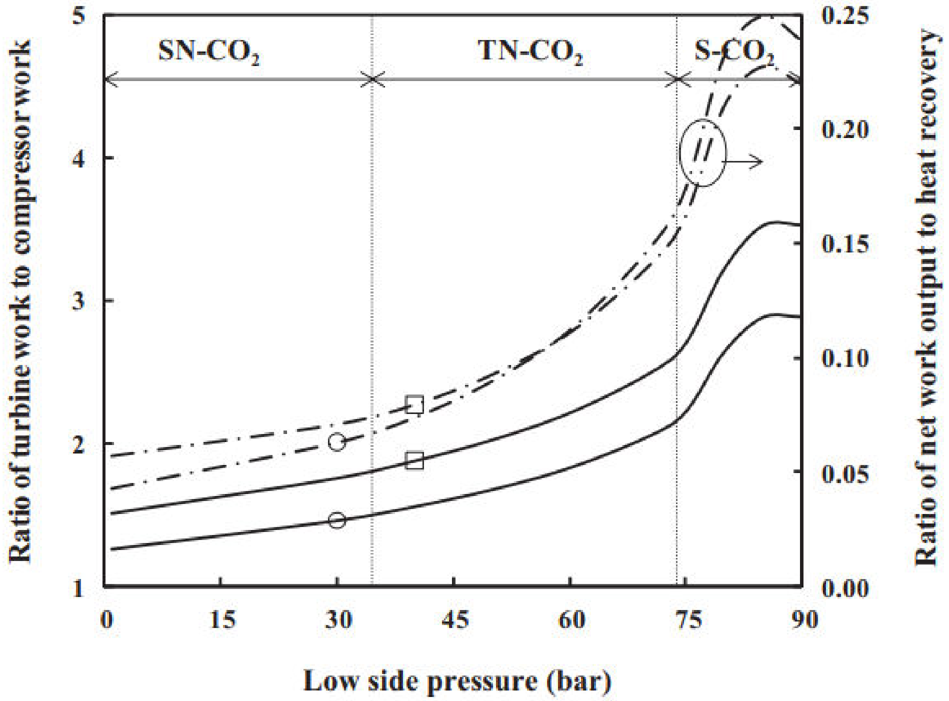


Optimum expansion ratio and corresponding thermal efficiency vs low side pressure for 873 K.

Further comparison of all the four possibilities under their optimal conditions

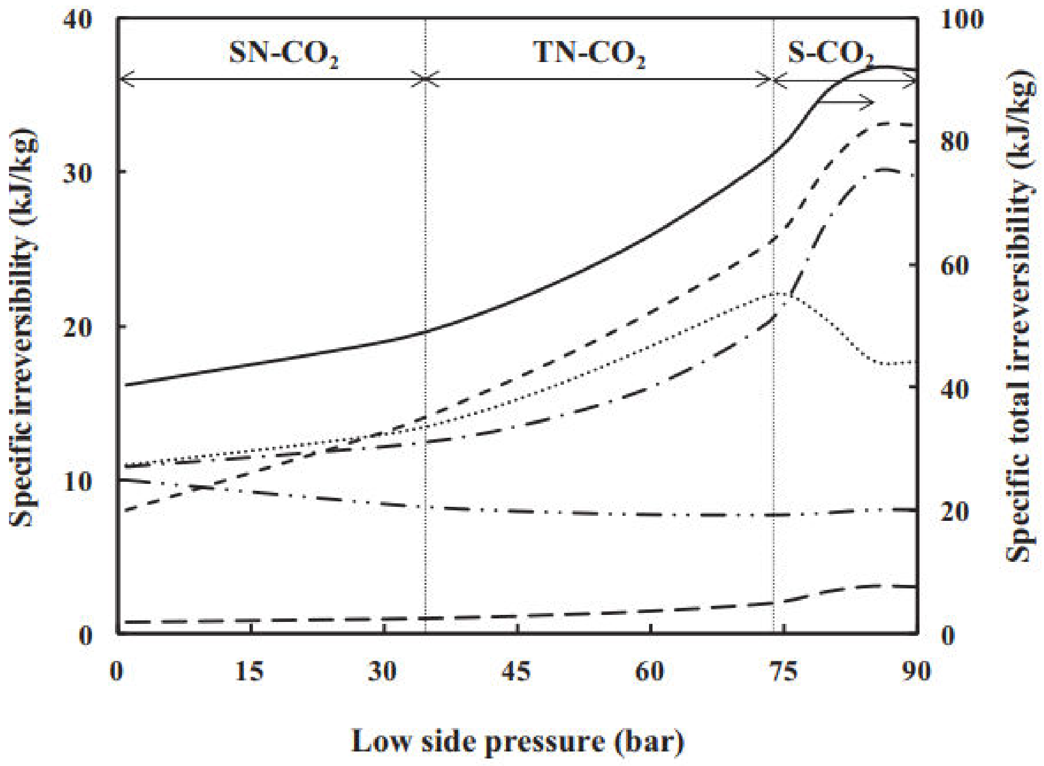


Ideal and real cycle efficiency vs turbine inlet temperature for different low side pressures at their optimum expansion ratios. p1 = 1 bar, ········ p1 = 40 bar, ------ p1 = 70 bar, - - - - p1 = 75 bar, − · − · − p1 = 85 bar, SN-CO2 cycle, TN-CO2 cycle, S-CO2 cycle, ideal SN-CO2 cycle, ideal TN-CO2 cycle, and ideal S-CO2 cycle



Ratio of turbine work to compressor work and net work output to heat recovery vs low side pressure at source temperature of 820 and 978K. ratio of turbine work to compressor work (left ordinate), − · − · − ratio of net work output to heat recovery (right ordinate), T3 = 820K, and T3· = 978K

Reasons for low efficiencies despite high pressures and temperatures



Component wise and total specific irreversibility vs low side pressure at source temperature of 899K. Sub-systems (left ordinate); boiler, · · · · compressor, - - - - regenerator, – · – · – turbine, ········ gas cooler; system (right ordinate), and total.

- Regenerator and gas cooler are the biggest sources of irreversibility. Both are heat exchangers.
- At low pressures and non condensing subcritical cycles the compressor takes away most of turbine work.
- Even in supercritical operation almost a third of work is taken away by the compressor

MAIN DISADVANTAGES

- The compressor work is a sizeable fraction of expander output
- kW or even low MW level power generation will be impractical because the mass flow rates are small and so are volumetric flow rates.
- This causes partial admission around the expander rotor or requires extremely high speeds of rotor (~30000 rpm).
- High rpm necessitates a gear box if connected to an alternator
- Reciprocating expanders are impractical because of high power output needs and low isentropic efficiencies.
- To derive decent work output high-temperature source will be needed (such as concentrated solar, deep earth geothermal) if renewable sources are to be used

ADVANTAGES

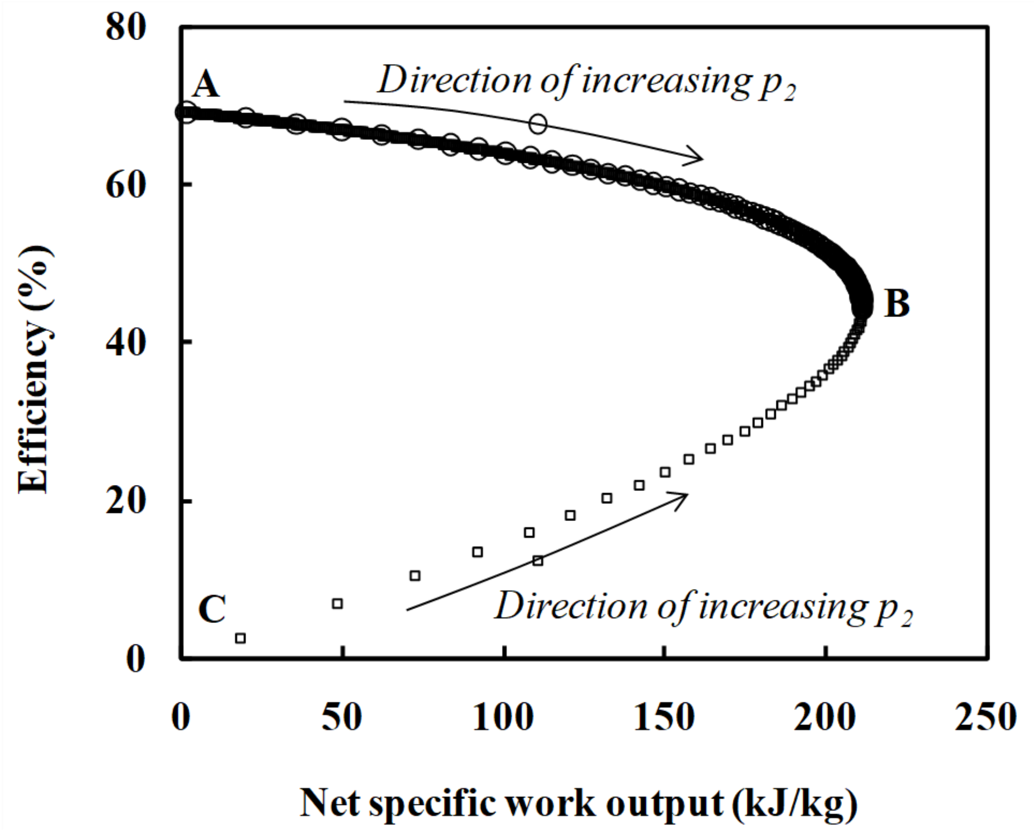
- Because low specific volumes piping can be smaller and compact than in the case of air
- Because of excellent heat transfer properties heat exchangers will be compact.
- High temperatures can be obtained just like in gas turbine power plants.
- No corrosion problems as in gas turbines and moisture is always present in combustion products

All other out put and efficiency improvement processes such as multistage compression, reheat during expansion, regeneration are as applicable to CO2 based Brayton cycles as conventional ones.

Eg: Multistage compression reduces compressor work; Reheat increases work output

Both can be used to increase the expansion ratio (as well as compression ratio)

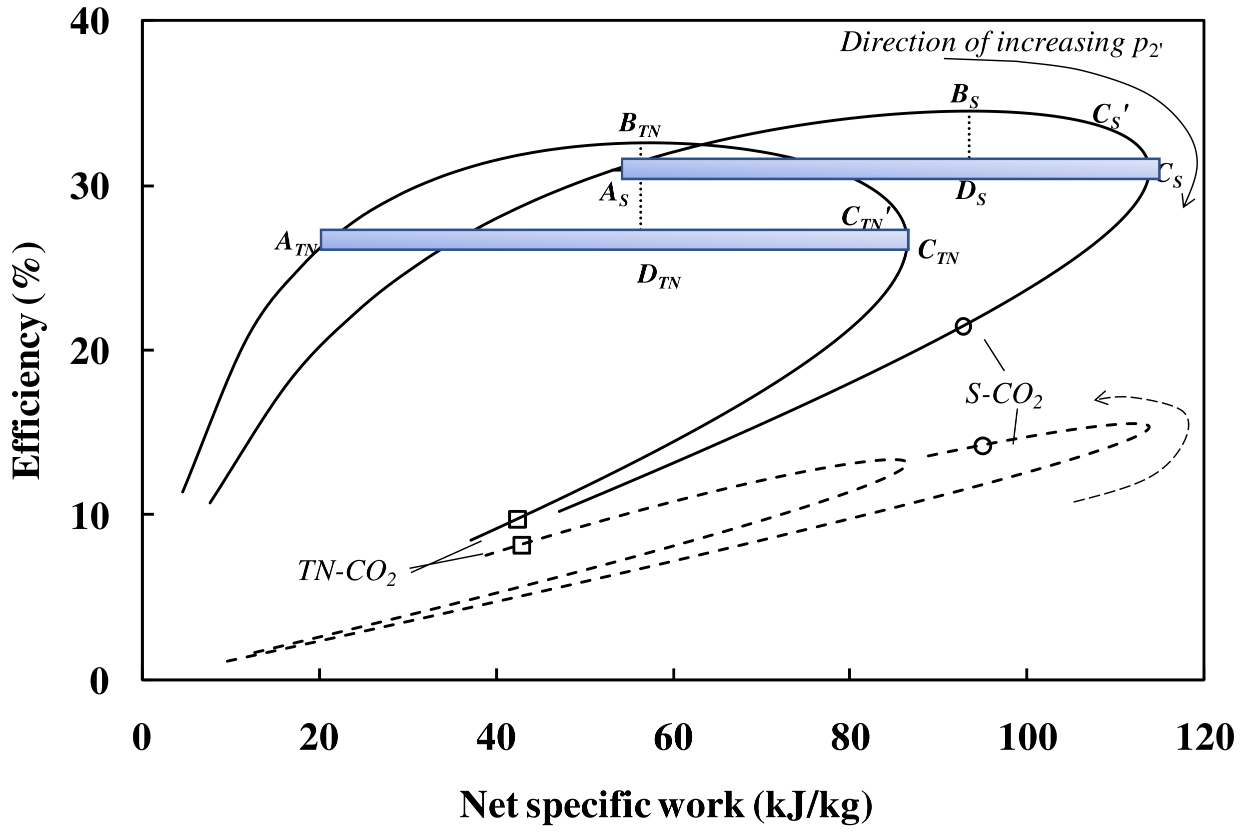
Performance issues of CO_2 Brayton Cycles



Efficiency *vs* net specific work for air Brayton cycle with p_1 = 1 bar, T_1 = 308 K, T_3 = 1000 K. Legend: □ without regeneration, ○ with regeneration

- Interestingly, two parameters we are interested are the efficiency and power output. Although, one likes to have both of them to be maximum, they never occur at the same operating conditions.
- Most of academia emphasize on efficiency (or COP in the case of refrigeration)
- Industry is invariably concentrates on maximising power output (or cooling capacity in the case of refrigeration)

Nature of efficiency-specific work relations in general



A-B efficiency and specific work increase
B - the point of maximum thermal efficiency
C –point of maximum specific work
B-C -the direction of increasing expansion ratio,
Below C - both efficiency and specific work output decrease , a region is not expected to be of interest.

Zone of regulation without sacrificing efficiency

Efficiency vs net specific work for TN-CO2 and S-CO2 cycle at T3' = 1000 K. Legend: —— with regeneration, – – – without regeneration, □ TN-CO2 cycle (p1= 50 bar), ○ S-CO2 cycle (p1= 75 bar). Direction of increase in p2' is clockwise for a cycle with regeneration and anti-clockwise when the cycle is without regeneration

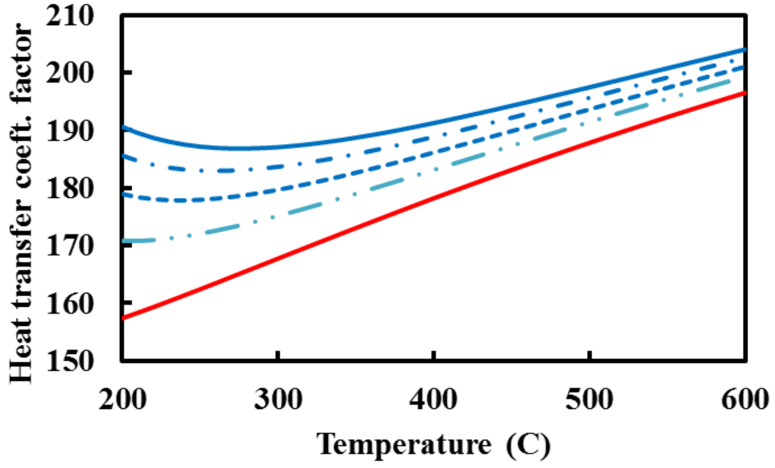
- With receiver temperature in the range of 800-1000 °C, these cycles can have efficiency as high as 50%.
- In closed loop Brayton cycles are more efficient in multistage compression with intercoolers and regeneration.
- These cycles do not need a bottoming ORC cycle.
- Energy storage is possible using a thermal energy storage like pebble bed or silica based rock
- Safe operation from hazardous substance and flammability point of view but need high pressures

Heat transfer and pressure drop issues

$$h \propto k\mu^{-0.8}\mathrm{Pr}^{0.33}\, d^{-1.8}$$

Virtually no difference between

- the two streams for same d at high temperature end.
- 75 and 80 bar warm streams

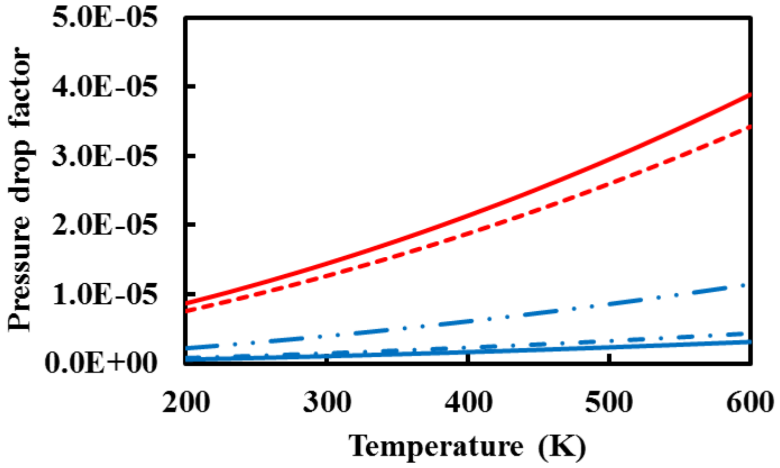


Cooler stream

300 bar — 200 bar

250 bar — 150 bar

warmer stream — 75 bar — 80 bar



$$\Delta p \propto \mu^{0.25}\,\rho^{-2}\, d^{-4.75}$$

Low pressure stream has a substantially larger pressure drop, mainly at the point of entry to regenerator for same d.

Critical science and engineering challenges

- Development of high temperature receivers for Brayton cycle, with CO_2 as the working fluid.
- High temperature thermal energy storage and low cost reflectors for power tower systems, if solar power is used
- Development of high efficiency compressors and turbo-expanders
- Biggest liability is the compressor power. If this can be taken out, there will be no limitation of pressure ratios nor other limitations imposed by it.

CO_2 as a Working Fluid in Vapour Compression Refrigeration

History of CO_2 as a Refrigerant

1850	First patented
1860	First refrigeration system built
1920	CO_2 became more widely used
1950s	The last CO_2 systems were installed in marine applications
1993	The revival of CO_2 refrigeration technology with the first subcritical systems being installed again
2000s	First trans-critical system in a supermarket
2008	The introduction of parallel compression and subsequently ejectors led to a much higher adaptability of trans-critical CO_2 in regions with high ambient temperatures

Refrigerants used for cooling

Basic requirement	- non-ozone depleting - non- greenhouse gas
Pre-Montreal Protocol era	*- R-22 and R-502*
Post Montreal Protocol options	- R-507a, 404a, 407c, 410a (all are blends of zero ODP refrigerants)
Post Kyoto Protocol options	“good old” ammonia (not widely accepted yet in supermarkets because of smell)
Flammable refrigerants	-Propane/Propylene (not fully accepted because of safety reasons)
Natural refrigerant	- CO_2 High operating pressures (~30 bar in evaporator)

Why CO_2 is coming back as a refrigerant

Refrigeration and air conditioning is deemed to be responsible for nearly 40% of greenhouse gas emissions.

Global Warming Potential (GWP)

It is a measure of how much a given mass of Greenhouse gas is estimated to contribute to global warming. It is a relative scale which compares the gas in question to that of the same mass of Carbon Dioxide.

Refrigerant	R12	R22	R502	R404a	R507	R407c	R134a	R744	R717
GWP	10600	1700	4500	3800	3900	1700	1300	1	0

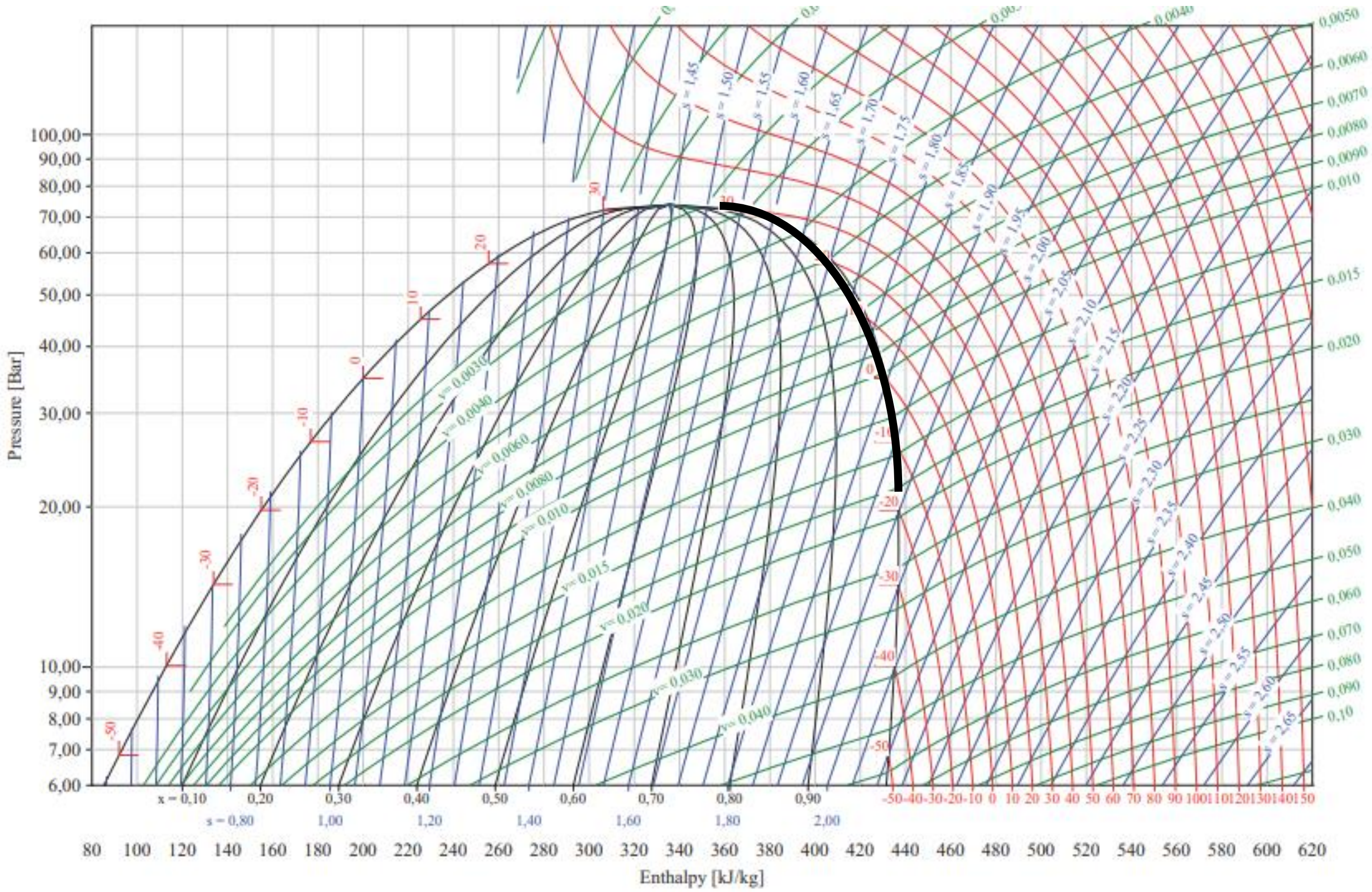
Total Equivalent Warming Impact. (TEWI)

It is the total global warming impact resulting from direct emissions of refrigerant leaking into the atmosphere plus the total indirect global warming impact arising from the indirect emissions as a result of the production of electricity that will be consumed by the plant during operation.

$$TEWI = GWP_{Direct} + GWP_{Indirect.}$$

For CO2 direct emission contribution will be nearly zero

Pressure-Enthalpy Diagram of CO_2

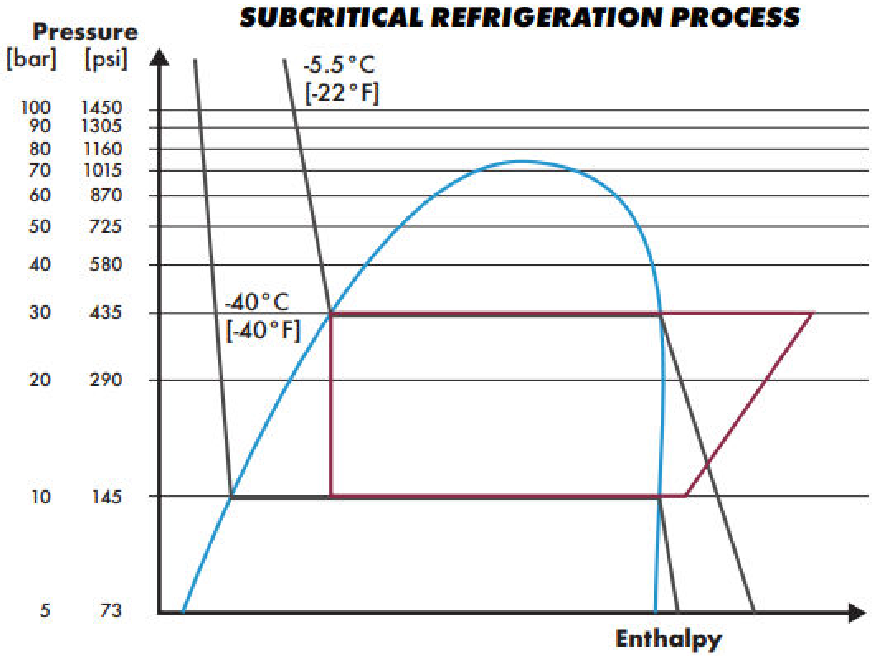


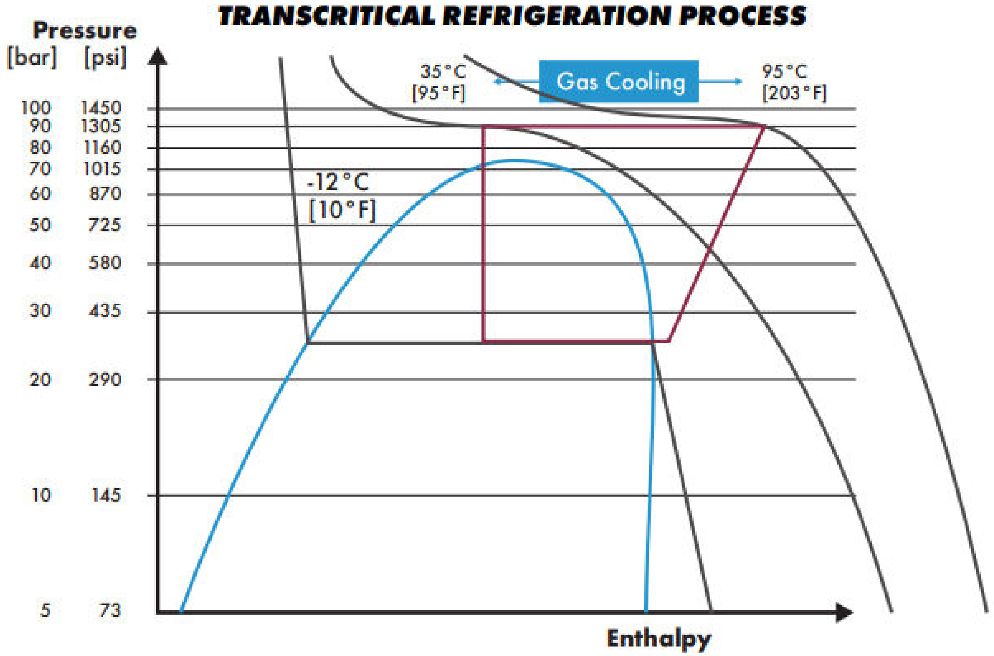
Zone in which most refrigeration systems work.

Note the –ve slope region of saturated vapour curve in p-h plane.

This is a special feature of CO2

From brochure of Semi-hermetic compressors for R744 (CO2), trans-critical applications of Frascold

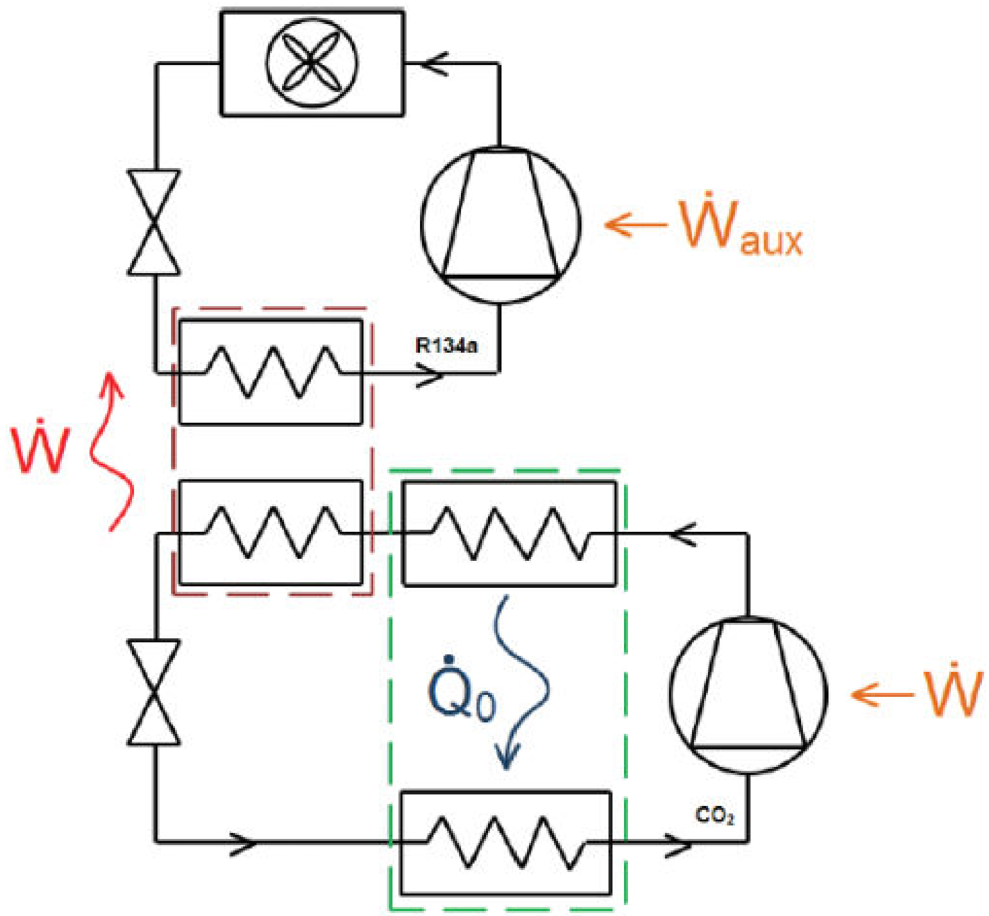




This is exactly like a standard vapour compression cycle except that the condensing temperature is well below ambient and not suitable for conditions in India

Unlike in standard vapour compression, there is no condenser. It is replaced by a gas cooler. Note that now that it is in supercritical region, gas cooler exit pressure and temperature are not related and each has to be chosen.

Cascaded CO_2 refrigeration system with upper cycle operated with HFC

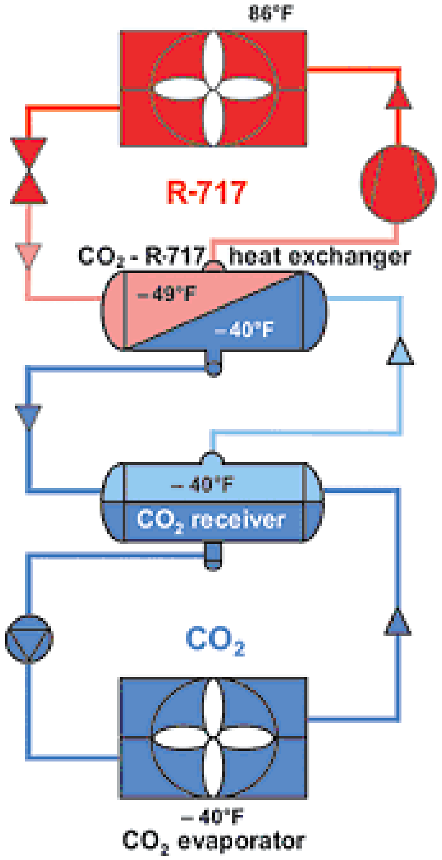


- There are still some problems with trans-critical CO2 refrigeration cycles (pressures, controls etc.)
- Subcritical CO2 is more or less standardized
- The MT compressors are replaced by a conventional HFC operated system in a cascaded mode.
- The condenser load of CO2 cycle is dumped on to evaporator of the upper stage

Liquid CO_2 Circulation Systems

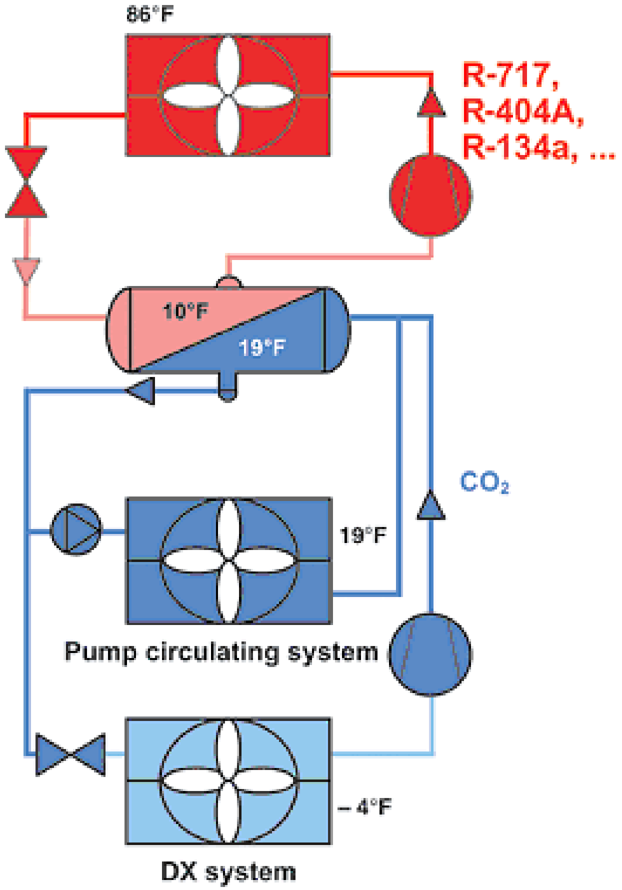
- CO2 is used as a secondary refrigerant (brine).
- Fully liquid recirculated CO2 systems would need a large volumetric flow rate at about temperatures where refrigeration is required because of its low specific heat (2.23 kJ.kgK). This will necessitate large pumps.
- CO2 must be at a pressure where required subcooling is required.
  - Eg: for -10^{o}C refrigeration, CO2 temperature my be allowed to change from -20 to -15^{o}C. At -15^{o}C, saturation pressure is about 23 bar. The liquid CO2 pressure should be ≥25 bar.
- To reduce volumetric flow rate a small amount of CO2 is allowed to evaporate, because of its large latent heat (~271 kJ/kg)
- The vapour generated is condensed using the main refrigeration plant that could be operated with HFC’s or ammonia.

Some configurations of liquid recirculation systems

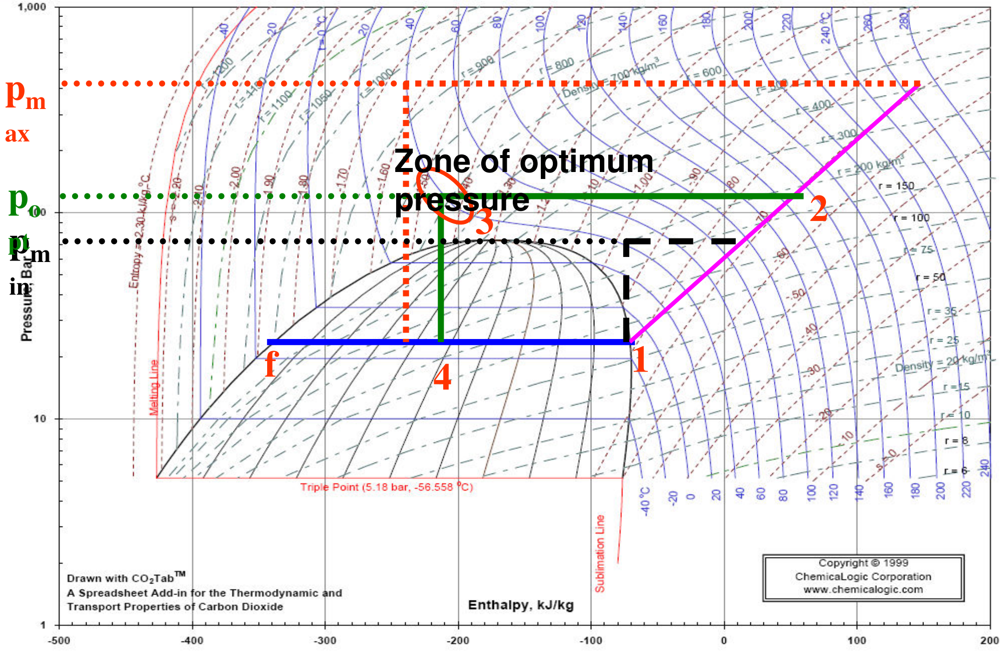


A typical CO2 cascade system with liquid recirculation. CO2 system rejects heat to upper stage cascade system operated with any other refrigerant (ammonia in this figure)

A typical CO2 cascade system with two temperature levels
Direct expansion for low temperature and liquid recirculation for medium temperature



How to determine the optimum gas cooler pressure



We can choose the upper cycle operating pressure (*an option we did not have in condensing vapour cycles*)

We can maximise

either the cooling capacity (at the cost of very high pressure)

or

COP (with a sacrifice in cooling capacity)

Minimise the entropy generation due to throttling

$$COP = \frac{h_1 - h_4}{h_2 - h_1} \qquad \frac{dCOP}{dp_3} = 0$$

$$-\left.\frac{\partial h_3}{\partial p_3}\right|_{T_3}(h_2 - h_1) - \left.\frac{\partial h_2}{\partial p_2}\right|_{s_1}(h_1 - h_4) = 0$$

$$COP = \frac{h_1 - h_4}{h_2 - h_1} = -\left.\frac{\partial h_3}{\partial p_3}\right|_{T_3} \Big/ \left.\frac{\partial h_2}{\partial p_2}\right|_{s_1}$$

$$\left.\frac{\partial h_2}{\partial p_2}\right|_{s_1} = v_2 \qquad \beta_3 = \frac{1}{v_3}\left.\frac{\partial v_3}{\partial T}\right|_{p_3}$$

$$\left.\frac{\partial h_3}{\partial p_3}\right|_{T_3} = T_3 \left.\frac{\partial s_3}{\partial p_3}\right|_{T_3} + v_3$$

$$COP = \frac{h_1 - h_4}{h_2 - h_1} = \frac{v_3(\beta_3 T_3 - 1)}{v_2}$$

s_3-s_4 must be minimised

$s_4 = s_f + (h_3 - h_f)/T_1$

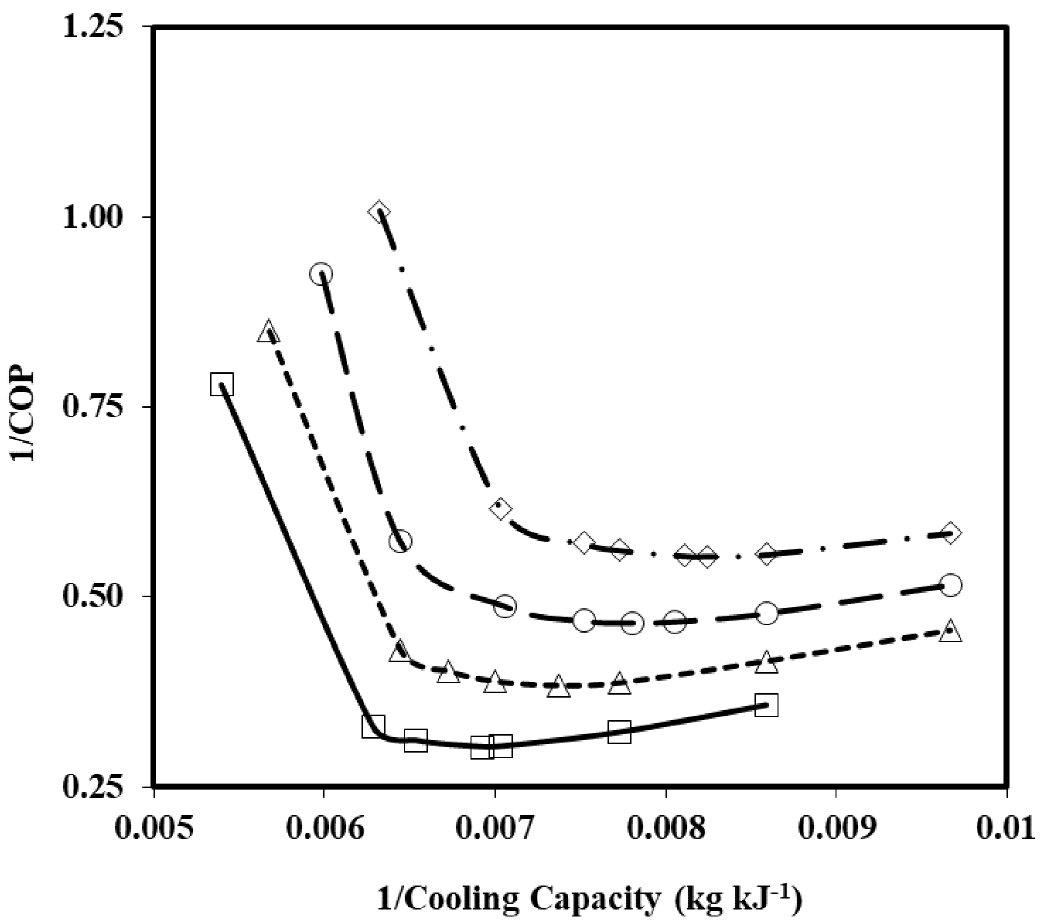
$$\left.\frac{\partial s_3}{\partial p_3}\right|_{T_3} - \frac{\left.\frac{\partial h_3}{\partial p_3}\right|_{T_3}}{T_1} = 0$$

$-\beta_3 T_1 = 1 - \beta_3 T_3$

$$\beta_3 = \frac{1}{T_3 - T_1}$$

p (bar) for -10/40 °C
min =74.4
for Q_{max}=436
Δs_{thmin} = 110.2
Ideal COP =2.1

- In any refrigeration system we can have either maximum COP or maximum cooling capacity.
- They do not occur at the same operating condition.
- Industry looks for maximising cooling capacity. This is because one has already invested in equipment and plant installation. Better to sacrifice some COP to get maximum cooling capacity.
- However, in the case of CO2 it is impossible to maximise cooling capacity due to requirement of very high pressures.



Positive aspects of CO_2 refrigeration

- Because of low specific volume of CO2 at operating conditions, the pipe line sizes will be small. This offsets the high pressure liability to an extent because thinner thicknesses may be used.
- High discharge temperatures from compressor provide an opportunity for heat recovery. This will be advantageous in dairy industry where heating and cooling are required.
- Because of good heat transfer properties overall size of heat transfer equipment can be reduced
- Because of various configurations available (discussed earlier) enormous flexibility is available to operate it as a stand alone system or in conjunction with other normally used refrigerants

Problems associated with CO_2 refrigeration

- Pressure ratios (p_g/p_e) will be large. Also pressure differential in the compressor will be large causing significant leakage from compressed gas to crank case.
- Isentropic index of CO2 is large
- Both of them together cause high discharge temperatures (>120^{o}C)
- Polyolester lubricating oils have to be used which are highly hygroscopic. They cannot be exposed to ambient humidity
- Because of high discharge temperatures oil cooler must be a part of the system. Also oil management is an issue
- Although high discharge temperatures provide an opportunity for heat recovery, it should not be a criterion for choosing CO2.
- Trans-critical gas cooler design correlations are still evolving and design procedures need standardization. Thermophysical properties change over a wide range in the critical region.
- Supercritical CO2 is difficult to store. CO2 must be expanded and stored in liquid form. Adequate additional refrigeration is needed to cool the receiver.
- The quality of CO2 admitted to evaporator will be quite large, hence only a small part of latent heat can be used.

Two stage compression

- Most of the problems are due to high temperatures and high pressures together
- If the high temperatures can be managed to be at the same level as with other refrigeration systems some problems can be alleviated.
- Two stage compression is worth exploring
- Intercooling will be essential between lower and upper stages.

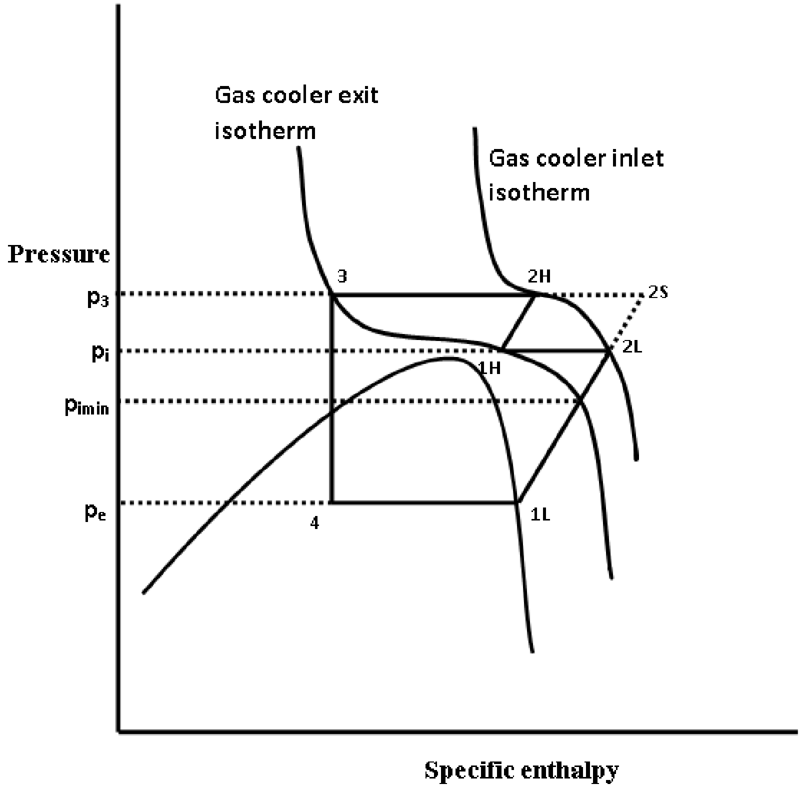
The following criteria have to be considered for determining the inter-stage pressure:

i) The minimum exit temperature from the low stage should be at least the gas cooler exit temperature for any heat transfer to occur in the intercooler,

ii) the exit temperature of the gas from low and high stage compressors should be equal,

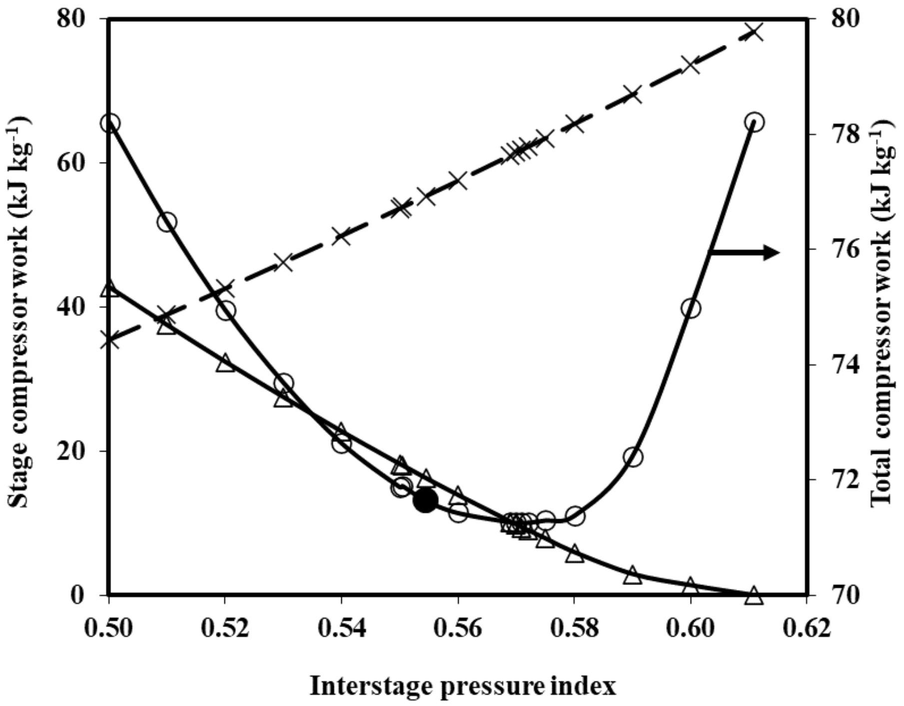
iii) the sum of work of compression in the two stages should be minimum, which is the conventional criterion and

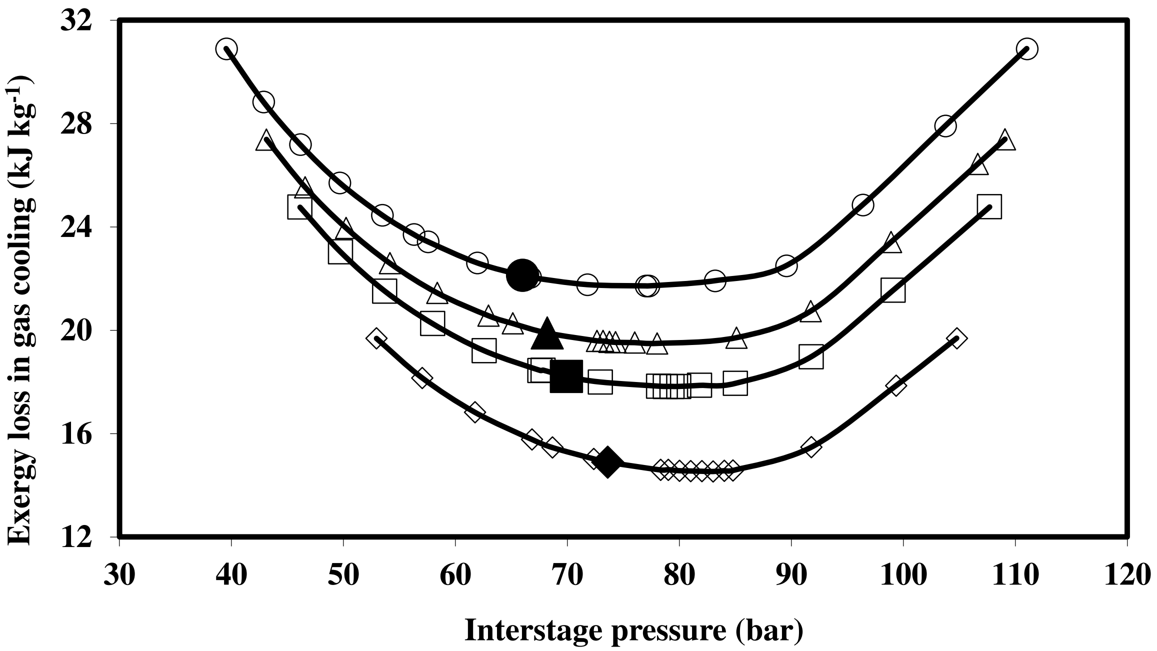
iv) the exergy loss during heat rejection (in the gas cooler plus the intercooler) should be the least.



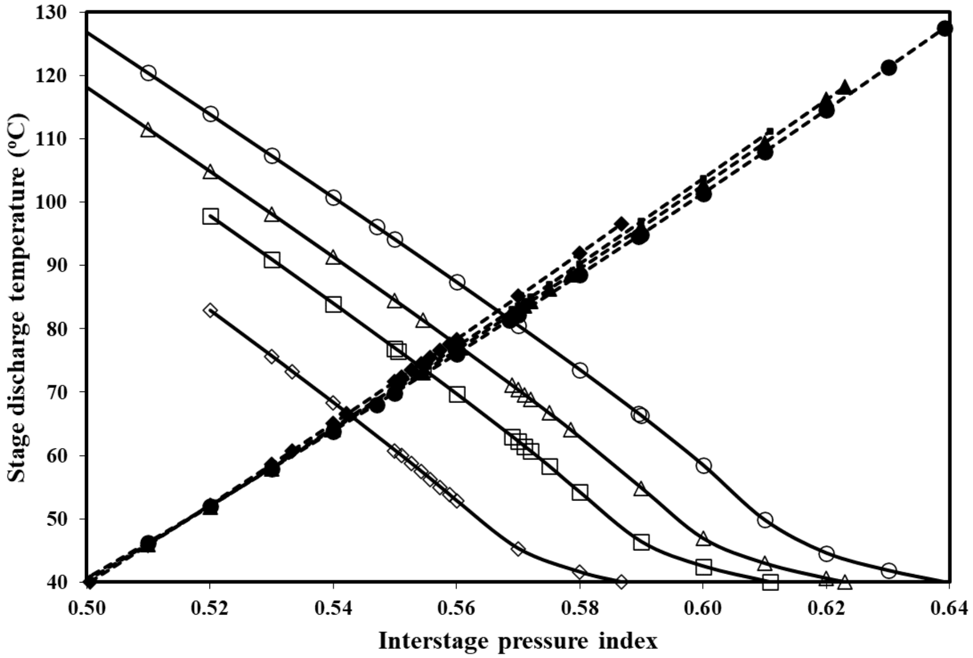
A schematic representation of CO2 trans-critical two-stage compression refrigeration cycle on the p-h co-ordinate system

Interstage pressure index $n = \dfrac{\ln p_i}{\ln(p_{1L} p_{2H})}$





Variation of exergy loss in gas cooling with inter-stage pressure for t_3=40 °C. Legend: ◇-t_e=-10 °C, ☐-t_e= -20 °C,D-t_e= -24.5 °C, O- t_e= -30 °C. Darkened points correspond to the state of equality of exit temperatures of each compressor stage



Stage discharge temperatures for various inter-stage pressure indices for t_3=40 °C. Legend: High stage - open symbols, low stage - filled symbols. ◇-t_e=-10 °C, ☐-t_e=-20 °C, D-t_e=-24.5 °C, O- t_e=-30 °C

Main advantage of 2-stage compression will be

- Large volumetric efficiencies of each stage
- Lower leakage across the piston
- Lower discharge temperatures from compressor- easy oil management

Conclusion

CO_2

- IS A NATURAL ENVIRONMENTALLY FRIENDLY WORKING FLUID
- IT HAS EXCELLENT THRMODYNAMIC AND THERMOPHYSICAL PROPERTIES
- IT HAS A WIDE RANGE OF OPERATION FOR POWER GENERATION AND REFRIGERATION
- IT ENALES COMPACT SYSTEMS, REDUCING THE COST

CHALLENGES

- High pressure management.

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