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Gianfranco Angelino, Costante M. Invernizzi. Carbon dioxide power cycles using liquid natural gas as heat sink. Applied Thermal Engineering, 2009, 29 (14-15), pp.2935. 10.1016/j.applthermaleng.2009.03.003 . hal-00573462

HAL Id: hal-00573462

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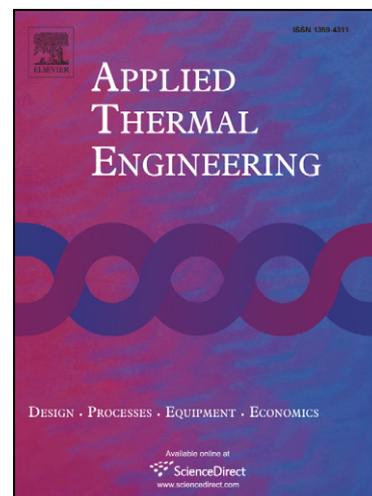
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PII: S1359-4311(09)00071-4
DOI: [10.1016/j.applthermaleng.2009.03.003](https://doi.org/10.1016/j.applthermaleng.2009.03.003)
Reference: ATE 2749

To appear in: *Applied Thermal Engineering*

Received Date: 21 November 2008
Revised Date: 23 January 2009
Accepted Date: 1 March 2009



Please cite this article as: G. Angelino, C.M. Invernizzi, Carbon dioxide power cycles using liquid natural gas as heat sink, *Applied Thermal Engineering* (2009), doi: [10.1016/j.applthermaleng.2009.03.003](https://doi.org/10.1016/j.applthermaleng.2009.03.003)

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Carbon dioxide power cycles using liquid
natural gas as heat sink

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March 14, 2009

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Abstract

Liquefied natural gas (LNG) is recognized as a source of usable cryogenic exergy for power cycles. The performance of conventional cycles are calculated. A binary steam–Organic Rankine Cycle (ORC) at 550 °C has an efficiency of about 52 %, somewhat higher than that of a nitrogen Brayton cycle (50.7 % at 700 °C). Carbon dioxide is recognized as an almost ideal medium for implementing single fluid condensation cycles. Its proven practical use both at low temperature (by the refrigeration industry) and at high temperature (by the nuclear reactor industry) makes it suitable for direct utilization without any extended preliminary research. A carbon dioxide cycle in its basic configuration featuring a pump, a regenerator, a heater and a condenser is much simpler than the binary steam–ORC cycle but has a lower efficiency (around 47 %). All condensing cycles (ORC, CO₂ ...) exhibit a limited capability of exploiting the whole cryogenic exergy of LNG in that they cannot heat the natural gas at temperatures above the condensation temperature. This drawback is fully overcome in nitrogen Brayton cycles which can heat LNG up to ambient temperature. Slightly modifying the basic CO₂ cycle so that it can partially use free thermal energy from sea water increases efficiency to 51 %. Multiple condensation cycles allow a better overall performance at the cost of a more complex layout. Compound CO₂ cycles, featur-

ing also a gas compressor, exhibit an improved thermodynamics by reducing the temperature difference within the regenerator, with the result of increasing the overall efficiency at values better than those of both binary and Brayton cycles. At 600 °C top temperature, for example, a compound cycle at 100 bar maximum pressure has an efficiency of 55.3 % (52.3 % for a binary steam–ORC cycle at 550 °C, 150 bar steam parameters; 46.5 % for the nitrogen cycle at 600 °C top temperature).

Keywords: Carbon Dioxide, Liquefied Natural Gas, Power Cycles, Supercritical Thermodynamic Cycles.

Nomenclature

Symbols and Acronyms

p	pressure, bar
t	temperature, °C
T	temperature, K
p_C	compressor discharge pressure in the LP compound cycle, bar
t_{cond}	cycle condensation temperature, °C
t_{max}	maximum temperature of the thermodynamic cycles, °C
p_{max}	maximum pressure of the thermodynamic cycles, bar
β_C	compressor pressure ratio
η	thermodynamic cycle efficiency
τ	(= T_{max}/T_{min}) overall maximum to minimum cycle temperatures ratio
AGR	Advanced Gas Cooled Reactor
HP	High Pressure
LP	Low Pressure
LNG	Liquefied Natural Gas
ORC	Organic Rankine Cycle
SPP	Specific Power Performance Parameter, MWe/(kg/s) of LNG

1 Introduction

All over the world natural gas transportation is increasingly relying on liquefaction and conveyance by specialized tankers. For structural reasons LNG is shipped basically at atmospheric pressure at a temperature of about -160°C . At the receiving terminal a large amount of cryogenic cold is available which, in general, is lost since LNG is directly vaporized using sea water as heat source.

Sometimes, mainly in Japan, LNG physical exergy is used for technological (air separation, CO_2 liquefaction, ...) or for power uses. In the latter case a propane or a refrigerant Rankine cycle using sea water as heat source and LNG as heat sink generates electricity without any fuel consumption. All existing power plants of this kind (around fifteen) employ a small fraction of the overall LNG cooling capability available at the site (net outputs limited to a maximum of 9400 kW). Often the net power obtained from the thermodynamic cycle is supplemented by an additional contribution due to the direct expansion of methane which is heated at a pressure greater than that requested by the final user and eventually expanded [1, 2].

Envisaging the use of a high temperature conversion cycle (for example a closed nitrogen Brayton cycle, ref. [3]) the amount of power which can be generated increases greatly. As a first approximation a typical LNG import

terminal of 8 Gm³/year gaseous methane capacity (which could feed at least 5000 MWe combined cycle plant) has the potential of generating about 150 MWe in a high efficiency cycle using the LNG as heat sink.

It is not surprising that most existing energy recovery plants employ an organic fluid Rankine cycle since, besides direct expansion, this is a practical, straightforward way of exploiting the LNG thermal exergy.

Owing to the cryogenic temperatures envisaged the direct use of steam is banned. With reference to condensing cryogenic cycles both hydrocarbons and refrigerants exhibit a limited thermal stability besides some technical and environmental drawback. Carbon dioxide, on the contrary, while lending itself to the implementation of a condensing cycles, is stable up to very high temperatures, is non flammable, non toxic, non corrosive, inexpensive and was used extensively in the past, for decades, by the power industry in the Advanced Gas Cooled Reactors (AGR) at a top temperature of 650 °C. In principle it could represent an ideal option for low temperature uses as water it is at ambient temperatures. A limit of CO₂ as a working fluid is its triple point at -57 °C which, in general, prevents its use at even lower temperatures. Carbon dioxide power cycles were extensively investigated in the past for applications where the final cooling of the working medium in a condenser takes place just below the critical temperature of the substance (31 °C, ref. [4, 5, 6, 7]). In these conditions waste heat rejection takes place near the

top of the saturation dome and the power cycle extends almost completely in the gas region. When LNG is used as a cooling medium condensation can be performed at a much lower temperature and pressure thus enhancing the condensing nature of the power cycle and dramatically reducing optimum operating pressures.

In recent years there was a revival in CO₂ cycles studies mainly connected with their perspective use in medium temperature advanced nuclear reactors ([8, 9, 10]). A detailed analysis shows that supercritical CO₂ cycles could be more efficient than steam or closed nitrogen cycles at temperatures above 600 °C and that turbomachinery design exhibits definitive advantages over both steam and nitrogen. As a consequence of the rising interest in CO₂ as a working medium some hardware development was performed relating to compressors, heat transfer equipment and plant layouts ([11, 12]).

Having made clear that carbon dioxide features a number of qualities to be proposed as working fluid at cryogenic temperatures the aim of this study is the evaluation in detail of its merits in the exploitation of the LNG cooling capability.

With reference to typical capacity of LNG import terminals a power plant output in the range 50–80 MW was considered in order to estimate the turbomachinery and other components performance both in conventional and in CO₂ conversion systems.

For the LNG to be heated two situations of practical interest are considered: vaporization at 70 bar i.e. at the supercritical pressure needed by long distance conveyance pipelines and vaporization at 30 bar as requested by power station supply lines.

2 Conventional power plants

In order to obtain a consistent picture of the merits of conventional and of new solutions a steam/ORC and a closed nitrogen gas cycle were firstly examined. Both these systems can be considered as "conventional" (even if none is in operation in connection with an LNG power plant), in that the related hardware was widely implemented in a number of power stations built in the past.

2.1 Steam–ORC system

With reference to the steam cycle in principle it would have been possible to assume a standard vacuum condensing cycle supplemented by a bottoming ORC unit. However since the low pressure turbine is the most expensive plant component it seemed more reasonable to "cut" the steam cycle at a pressure slightly above one atmosphere and to commit to an ORC bottoming cycle the task of exploiting the full thermal potential from about 100

°C to the cryogenic LNG temperatures. In this way reheating of partially expanded steam becomes unnecessary since the steam quality at the end of the expansion is sufficiently high (above 0.92 at 150 bar and 550 °C turbine inlet conditions). Cycle configuration, steam parameters and component performance were stipulated in agreement with a plant capacity in the range of 50–80 MWe and are reported in **Table 1**. Rankine cycle data were taken from existing plants of similar output. Brayton cycle parameters are consistent with the ORC current practice. Nitrogen cycle data were taken from the literature, [14]. An overall efficiency of 31.7 % was calculated resorting to AspenPlus computation program.

For the bottoming cycle n-butane ($n - C_4H_{10}$) was selected as the working fluid since its critical temperature of 152.0 °C approximately optimizes the cycle performance. A one stage extraction was assumed for the regeneration process. As it is typical when dealing with organics at the end of the expansion the vapour is slightly superheated which favours a good turbine efficiency. In principle a hydrocarbon with a more complex molecule such as n-pentane ($n - C_5H_{12}$) could be used. In this case however, at a condensation temperature of -50 °C vapour pressure at the turbine exhaust would be impractically low (0.013 bar against 0.096 bar for n-butane).

For a steam/n-butane binary plant configuration (see **Figure 1.a**) the computed overall efficiency, obtained by the use of the AspenPlus program,

is 52.3 %. A more straightforward solution consisting of a standard, re-heat steam cycle condensing at 30 °C followed by a n-butane cycle vaporizing at 20 °C and condensing at -50 °C according to the same computing program would exhibit a slightly better overall efficiency (53.7 %) at the cost of a complex vacuum section and of a second steam superheating (42.3 % steam, 19.7 % n-butane cycle efficiency).

2.2 Brayton cycle system

The successful more than decennial experience of closed cycle gas turbines ([13, 14]) represents a sound basis for the extension of this concept to cryogenic temperatures. A detailed solution for one such plant serving an LNG terminal is reported in [3]. Nitrogen with a very small oxygen addition is believed to be the best working fluid. Top temperatures are limited to 700–800 °C owing to material problems in the primary heater. The cooling characteristics of LNG which must be heated from -160 °C to ambient temperature suggest the adoption of a simple, non-intercooled, regenerative cycle. The overall maximum to minimum temperature ratio ($\tau = T_{max}/T_{min}$) which was around 3.4 in historical plants [13] in the peculiar LNG application rises to about 7* with a strong reduction in compressor work and a substantial gain in efficiency (from historical values of about 30 % to well above 50 %). The

*In the case of an open combined cycle with an LNG heat sink τ exceeds 11.0.

ability of closed gas cycle to burn various coal grades and other low quality fuels represents a distinctive feature in comparison with open cycle gas turbines. With reference to a supercritical LNG vaporization the heat rejection mode of a closed gas cycle matches almost perfectly with the LNG heat absorption law. This character is almost unique and makes the gas cycle a privileged conversion system.

As it is clear from **Figure 1.a** and **1.b**, in contrast with the ORC cycle behaviour, the closed gas cycle is capable of heating the LNG up to the ambient temperature thus exploiting the entire cooling capability of the liquefied natural gas. Condensing cycles can reach a similar target only with a complex configuration featuring a plurality of condensation levels. As a consequence a typical Specific Power Performance Parameter (*SPP*) defined as the power output to the LNG mass rate ratio, (MW/(kg/s)) reaches for the closed gas cycles unrivalled values.

All the assumptions made to evaluate the performance of closed gas (nitrogen) cycles are given in **Table 1**.

System efficiency and parameter *SPP* for super- and sub-critical LNG are given in **Figure 2a** and **2b** respectively at various compressor pressure ratio and turbine inlet temperatures. For supercritical LNG at 700 °C optimum efficiency is 50.7 % ($SPP = 0.85$ MW/(kg/s)) i.e. slightly better than that of steam/ORC binary cycle at 550 °C ($\eta = 0.523$). At 800 °C $\eta = 0.543$, *SPP*

= 1.0. In the case of subcritical LNG optimum efficiencies remain unchanged while the specific power parameter increases significantly (0.94 at 700 °C).

3 Simple CO₂ cycles

For basic theoretical reasons no saturated Rankine cycle can handle the large temperature span available when dealing with LNG which extends, for example, between a top temperature of 600 °C and a minimum temperature of -50 °C ($\tau = 3.9$). A large portion of the cycle must be placed necessarily in the super-heated region. On the other hand a similar situation is encountered also in usual steam cycles even if, in this case, the temperature ratio is of only 2.7. Making reference to the actual CO₂ physical properties the simplest configuration for a power cycle extended to cryogenic temperatures turns out to be the one shown in **Figure 3**. The working fluid is condensed at -50 °C from point 5 to point 1, pressurized by pump to point 2, regeneratively pre-heated to point 6, super-heated to point 3, expanded in a turbine to point 4 and regeneratively pre-cooled to point 5 in a counterflow heat exchanger. With reference to **Figure 3.c** even if the minimum temperature difference within the regenerator is small (for example 10 °C) the computed hot-end temperature difference is rather large (212 °C) owing the strong difference in the heat capacity of the high and of the low pressure fluid i.e.

of the supercritical CO₂ (average heat capacity of 2.64 kJ/kg K) and of the low pressure gas after the turbine expansion (average heat capacity of 0.96 kJ/kg K). This substantially irreversible heat transfer represents the main efficiency limit of the simple CO₂ conversion cycle.

Assuming the components characteristics reported in **Table 2** (which are consistent with those stipulated for the steam and for the Brayton cycles), **Figure 4** reports the overall cycle efficiency as a function of turbine inlet pressure, turbine inlet temperature and of condensation temperature. Using once more the AspenPlus computation program, at $t_{max} = 600$ °C, $p_{max} = 150$ bar, $t_{cond} = -50$ °C efficiency turns out to be 46.3 % which compares with a 46.5 % efficiency of the nitrogen cycle (50 % against 50.7 % for nitrogen at 700 °C). In the case of CO₂, however all the compression is performed by a pump and the total power involved in both compression and expansion is only slightly larger than the plant net output (1.12 power ratio against 2.22 for Brayton cycle).

As it happens in all condensing cycles a compromise must be found between cycle efficiency (which request the minimum achievable condensation temperature) and the LNG exergy utilization which requires a high LNG final temperature which can only be obtained with a high condensation temperature and a reduced cycle efficiency.

Owing to these conflicting requirements the specific power output SPP

reaches a maximum at a definite condensation temperature while cycle efficiency increase regularly at decreasing t_{cond} . Such a behaviour is illustrated in **Figure 5**. With reference to a LNG mass rate of 1 k/s at 150 bar top pressure, for example, an optimum SPP value of 0.43 MWe is achieved at $t_{cond} = -10\text{ }^{\circ}\text{C}$ with a cycle efficiency of 38 % (bold lines).

3.1 Two condensation levels

The situation outlined above can be improved with a cycle featuring two condensation levels (see **Figure 6**) in which the LNG heating is performed in two steps with a reduction of the average temperature difference between condensing CO_2 and LNG (**Figures 6.b and 6.c**). The definite gain in SPP parameter is clear from **Figure 5**, dotted lines. For a fixed lower condensation temperature of $-50\text{ }^{\circ}\text{C}$ at an upper condensation temperature of $-10\text{ }^{\circ}\text{C}$, at 150 bar, SPP rises to 0.53 MWe/(kg/s) (from 0.43) while cycle efficiency reaches 43 %. Obviously multiple condensation cycles could also be considered with a further benefit in system performance.

3.2 One condensation level with a partial input of free energy extracted from sea water

As it is clear from **Figure 3** the pre-heating of CO₂ after condensation starts at a very low temperature, for example around -50 °C. Such pre-heating could be performed partially using sea water heat at about ambient temperature. Obviously in this case the regenerative pre-heating is shifted toward higher temperatures as shown in **Figure 7**. As a consequence the whole thermodynamic cycle is modified with a substantial gain in efficiency[†] while parameter *SPP* turns out to be practically constant (**Figure 5**, light lines). For example at $t_{cond} = -50$ °C, $p_{max} = 150$ bar $t_{max} = 600$ °C cycle efficiency increases from 46.3 % to 51.0 %. Owing to the simplicity of the technical intervention involved the partial use of the free heat from sea water seems, in general, advisable.

[†]Free energy transferred from sea water is not considered as part of the cycle heat consumption.

4 Compound CO₂ cycles

4.1 High pressure compound cycles

With reference to **Figure 3.c**, as previously observed, the large temperature difference in the simple cycle regeneration process ($t_4 - t_6 = 212$ °C in the diagram reported) is responsible of a severe entropy production and efficiency loss. To get around this drawback in the regeneration process heat capacities of the cold and of the hot medium can be balanced reducing the mass rate of the condensed gas by compressing, in a mechanical device (without condensation), a fraction of the pre-cooled gas. **Figure 8** reports the new cycle configuration and component layout as originally suggested in [7]. As shown in **Figure 8.b** heat transfer in the cold regenerator becomes much less irreversible (average temperature difference 17 °C) with an important gain both in efficiency and in the *SPP* parameter. As shown in **Figure 9** at a top temperature of 600 °C, for example, maximum efficiency is 53.3 % (47.5 % for the simple cycle, 46.5 % for the nitrogen Brayton cycle) and maximum *SPP* is 0.69 MWe/(kg/s) (0.50 for the simple cycle, 0.72 for the nitrogen Brayton cycle). It must be observed, however, that for CO₂ in opposition to what happens in the nitrogen cycle, top efficiency and top *SPP* are reached at different condensation temperatures (-50 °C for the best efficiency, 0 °C for the best *SPP*).

An advantage of the high pressure compound cycle with respect to the nitrogen cycle is that the power input for the mechanical compressor is comparatively small since it handles only about half of the total mass rate. At a top temperature of 700 °C a remarkable efficiency of 57.2 % is achieved (50.7 % for the nitrogen cycle).

4.2 Low pressure compound cycles

In the high pressure compound cycle both pump and compressor pressurize the working fluid to the same pressure which, aiming to a good overall efficiency, must be comparatively large (for example 150 bar). Since compressor losses are more severe than pump losses such a high pressure generates within the compressor a larger than desirable fluid-dynamic penalty which could be reduced if compressor discharge pressure is substantially lowered. For the pump, on the contrary, discharge pressure can be kept high. In other words the gas cycle and the condensation cycle can be partially de-coupled generating a new cycle configuration, see ref. [15], which is illustrated in the T-S plane in **Figure 10**. As in the cycle of **Figure 8** in the low temperature regenerator the low pressure CO₂ of both the gas and the condensation cycle pre-heat jointly the condensed liquid from point 1 to point 15, while cooled from point 20 to point 23. Two independent high temperature regenerators

preheat the high pressure gas from point 15 to point 16 in the condensation cycle and from point 2 to point 4 in the gas cycle.

As a consequence of this de-coupling optimum top pressures fall to about 100 bar for the condensation cycle and to about 40 bar for the gas cycle which implies obvious structural benefits and a simplified design for the compressor and the turbine of the gas cycle. **Figure 11** reports cycle efficiency as a function of compressor discharge pressure. At 600 °C top temperature optimum efficiency of the low pressure compound cycle is 55.3 % at 100 bar pump, 40 bar compressor discharge pressure which compares with 53.3 % of the high pressure compound cycle. At 700 °C top temperature efficiency reaches a remarkable 59.0 %.

5 Conclusions

The thermodynamic analysis reported in the previous sections, with reference to Figure 12 too, suggests the following conclusions:

1. conventional power cycles, both Rankine and Brayton, can be adapted to the exploitation of the LNG thermal exergy with conversion efficiencies in the range 50 to 54 %. Rankine cycles must have a binary (steam–organic fluid) configuration. Nitrogen Brayton cycle, although less efficient than Rankine cycle at a fixed top temperature, has the

capability of heating the LNG up to ambient temperature.

2. Carbon dioxide exhibits several merits to be considered as a working fluid at cryogenic temperatures: it is inert, stable, non toxic, inexpensive, readily available and was extensively used in the past both at low temperatures (by the refrigeration industry) and at high temperatures (mainly in the AGR nuclear reactors). Furthermore carbon dioxide cycles in the condensing mode, have the potential of covering, with a single fluid, the whole available temperature range, for example from -50 to 600 °C. The recent revival of the studies related to fast nuclear reactors using CO₂ as working fluid for the power cycle produced a wide literature covering thermodynamics, material science and hardware development.
3. Simple CO₂ condensing cycles, featuring a gas turbine, a regenerator, a condenser and a pump, offer a moderately good efficiency (for example, 46.3 % at 600 °C, 150 bar) combined with a very simple layout. The simple cycle can be adapted to partially use free heat extracted from sea water with a significant efficiency gain (from 46.3 % to 51.0 %). More complex compound cycles, which require also a compressor, allow much higher efficiencies at moderate top pressures. The low pressure compound cycle for example at a maximum pressure of 100 bar exhibit

an efficiency of 55.3 % at 600 °C and of 59.0 % at 700 °C

4. With respect to a binary steam–ORC solution, CO₂ cycles are characterized by a much simpler layout, higher efficiencies and smaller and cheaper turbomachinery. With respect to a nitrogen Brayton cycle at a given top temperature, CO₂ cycles exhibit a better fuel economy but with the drawback of not being able to heat the LNG up to ambient temperature.

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Figure Captions

Figure 1 Binary steam–butane cycle (Figure 1.a) and nitrogen closed gas cycle (Figure 1.b) in the T–S plane using LNG as heat sink.

Figure 2 Efficiency and specific power output of Brayton nitrogen cycles for supercritical (Figure 2.a) and for subcritical (Figure 2.b) LNG heat sink.

Figure 3 Carbon dioxide simple cycle at $-50\text{ }^{\circ}\text{C}$ condensation temperature.

Figure 4 Carbon dioxide cycle efficiency at $600\text{ }^{\circ}\text{C}$ (Figure 4.a) and $700\text{ }^{\circ}\text{C}$ (Figure 4.b) top temperature at various condensation temperatures.

Figure 5 Carbon dioxide simple cycle efficiency and specific power output for one and for two condensation levels and for a partial use of sea water heat.

Figure 6 Temperature–entropy diagram and heat rejection mode of a two condensation level CO_2 cycle (Figure 6a and Figure 6b). A one condensation level heat rejection diagram is also reported for comparison (Figure 6c).

Figure 7 Carbon dioxide simple cycle with partial use of sea water heat: T–S diagram and component layout.

Figure 8 High pressure compound cycle with improved low temperature regeneration.

Figure 9 Efficiency and specific power output for high pressure compound cycle at various top temperatures.

Figure 10 Low pressure compound cycle: T-S diagram and component layout.

Figure 11 Comparison of cycle efficiency of low- and high-pressure compound cycles at different top temperatures.

Figure 12 Cycle efficiency for the various options considered as a function of top temperature.

Table 1: Characteristics of steam, n-butane and nitrogen cycles.

<u>a) steam cycle</u>	
type of cycle	superheated with four regenerative extractions
turbine inlet temperature	550 °C
turbine inlet pressure	150 bar
condensation temperature	111.4 °C
condensation pressure	1.5 bar
turbine adiabatic efficiency	0.85
pump hydraulic efficiency	0.70
mechanical efficiency	0.985
alternator efficiency	0.98
overall fractional pressure drop	20 %
computed cycle efficiency	0.317
<u>b) n-butane bottoming cycle</u>	
type of cycle	saturated with one regenerative extraction
turbine inlet temperature	103 °C
turbine inlet pressure	16 bar
condensation temperature	-50 °C
condensation pressure	0.096 bar
turbine adiabatic efficiency	0.90
pump hydraulic efficiency	0.70
mechanical efficiency	0.985
alternator efficiency	0.98
overall fractional pressure drop	20 %
computed cycle efficiency	0.301
<u>c) binary cycle</u>	
overall efficiency	0.523
specific power output, <i>SPP</i>	0.44 MW/(kg/s) of LNG
<u>d) nitrogen closed gas cycle</u>	
turbine inlet temperature	600 to 800 °C
compression inlet temperature	-120 °C
turbine adiabatic efficiency	0.90
compressor adiabatic efficiency	0.85
regenerator effectiveness	0.90
mechanical efficiency	0.985
alternator efficiency	0.98
overall fractional pressure drop	13 %

Table 2: Assumed component efficiencies for CO₂ cycles.

turbine inlet temperature	550 to 700 °C
condensation temperature	-50 to 10 °C
turbine adiabatic efficiency	0.90
compressor adiabatic efficiency	0.85
pump efficiency	0.70
minimum temperature difference in the low temperature regenerator	10 °C
minimum temperature difference in the high temperature regenerator	25 °C
mechanical efficiency	0.985
alternator efficiency	0.98
overall fractional pressure drop	13 %

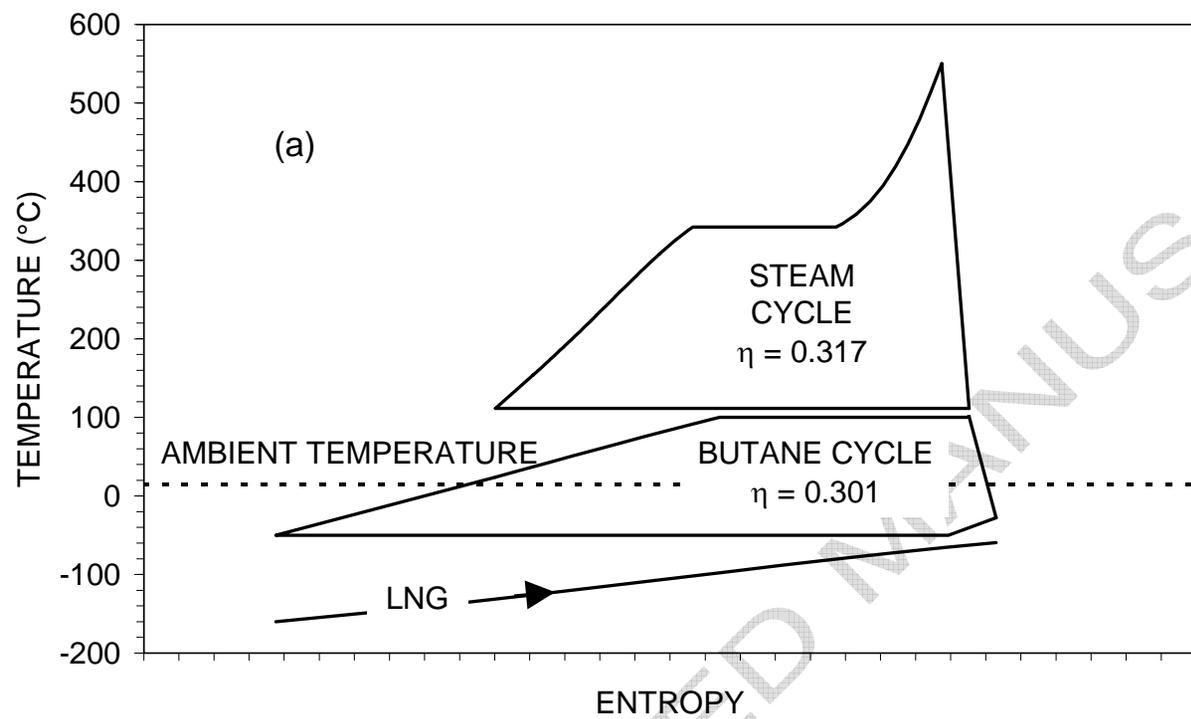


Figure 1.a

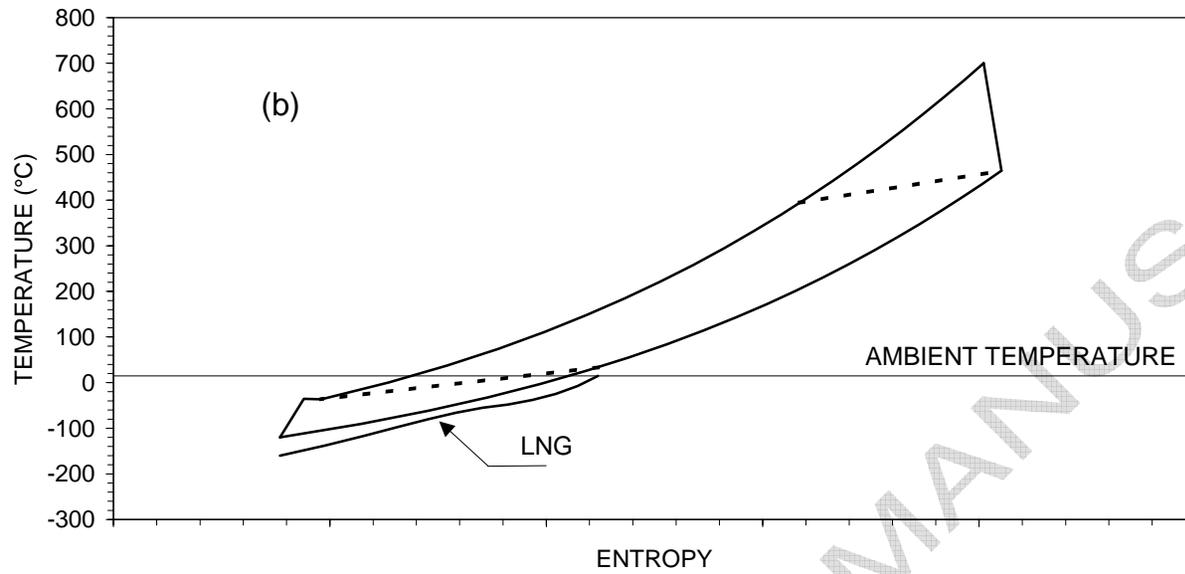


Figure 1.b

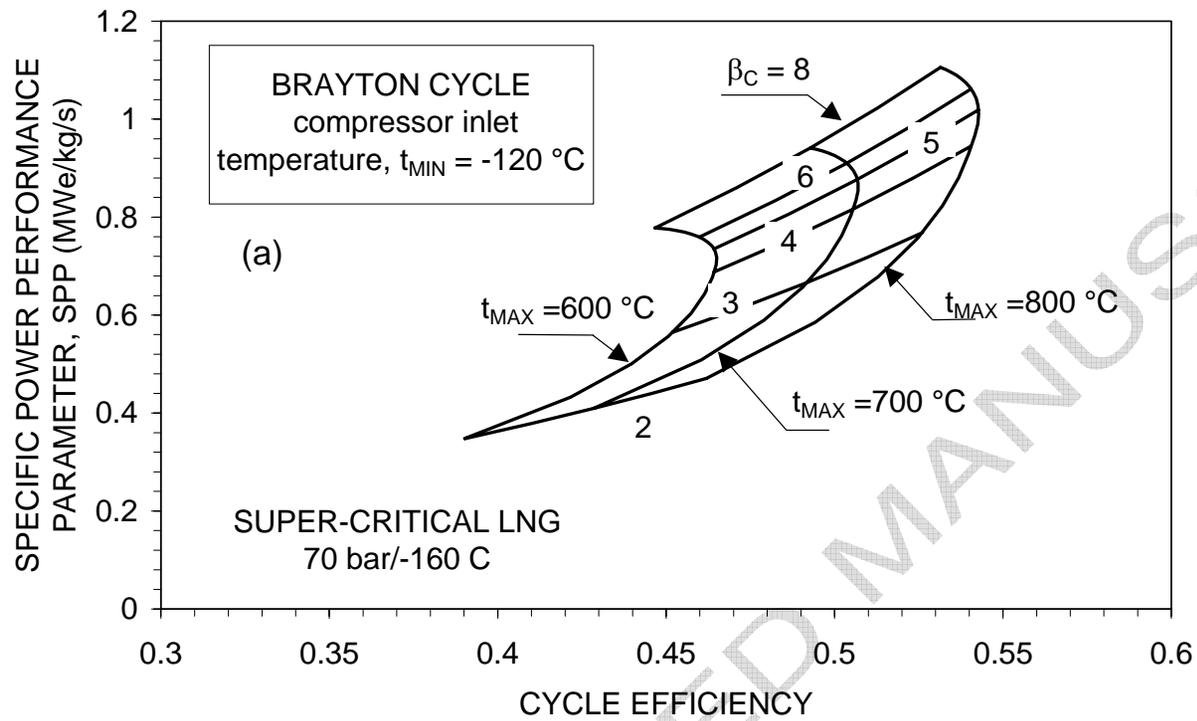


Figure 2.a

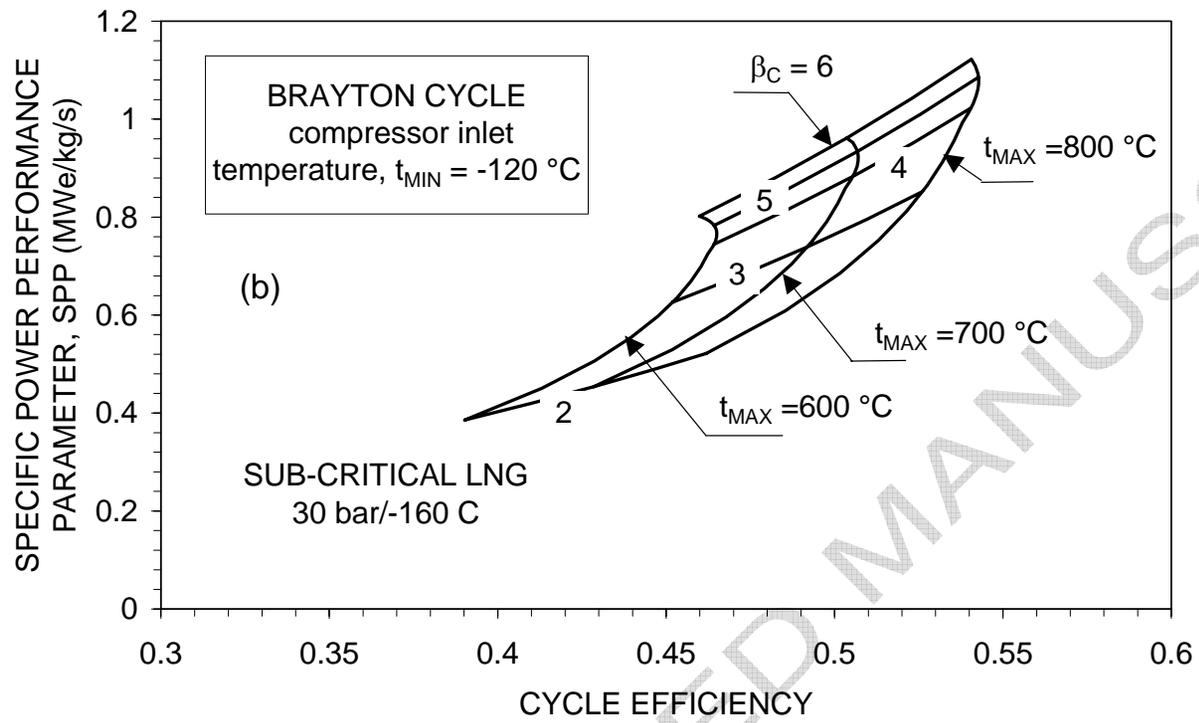


Figure 2.b

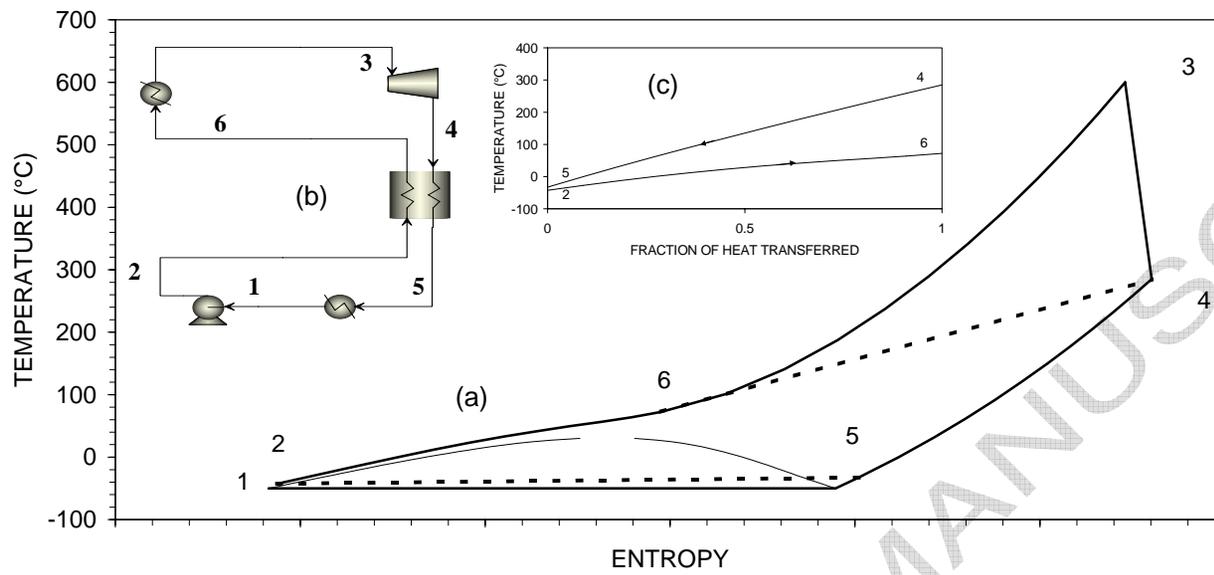


Figure 3

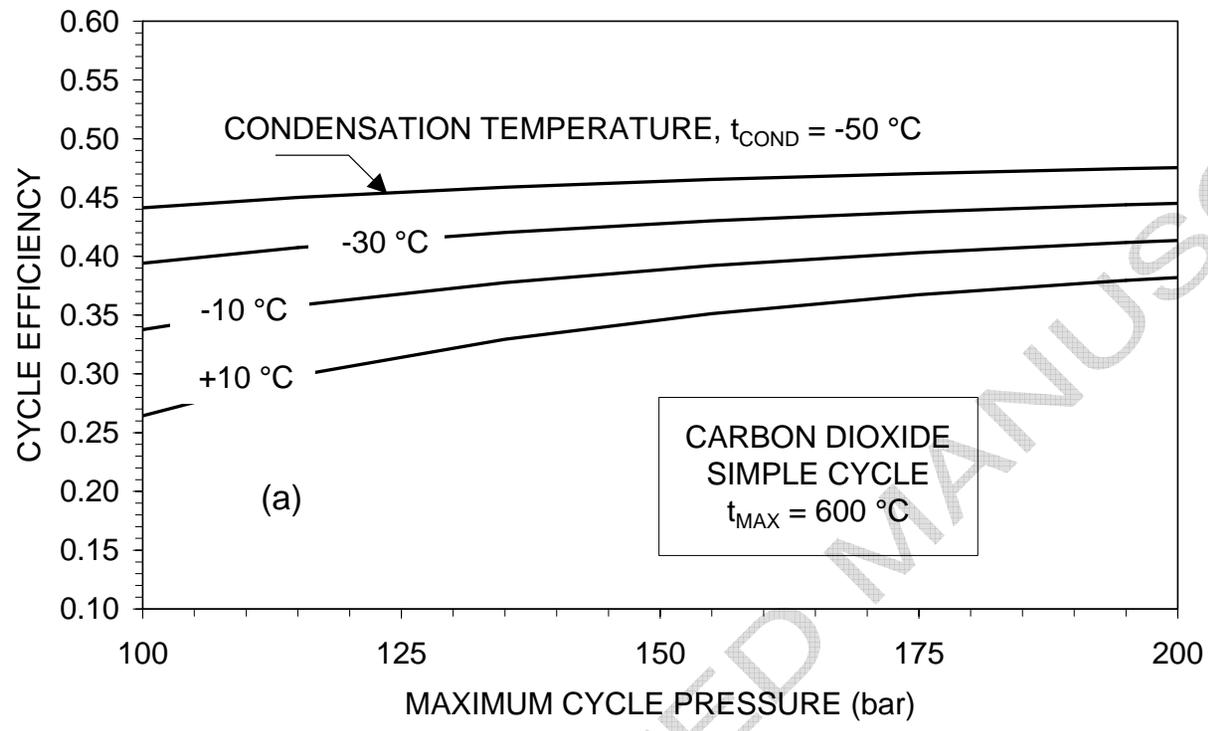


Figure 4.a

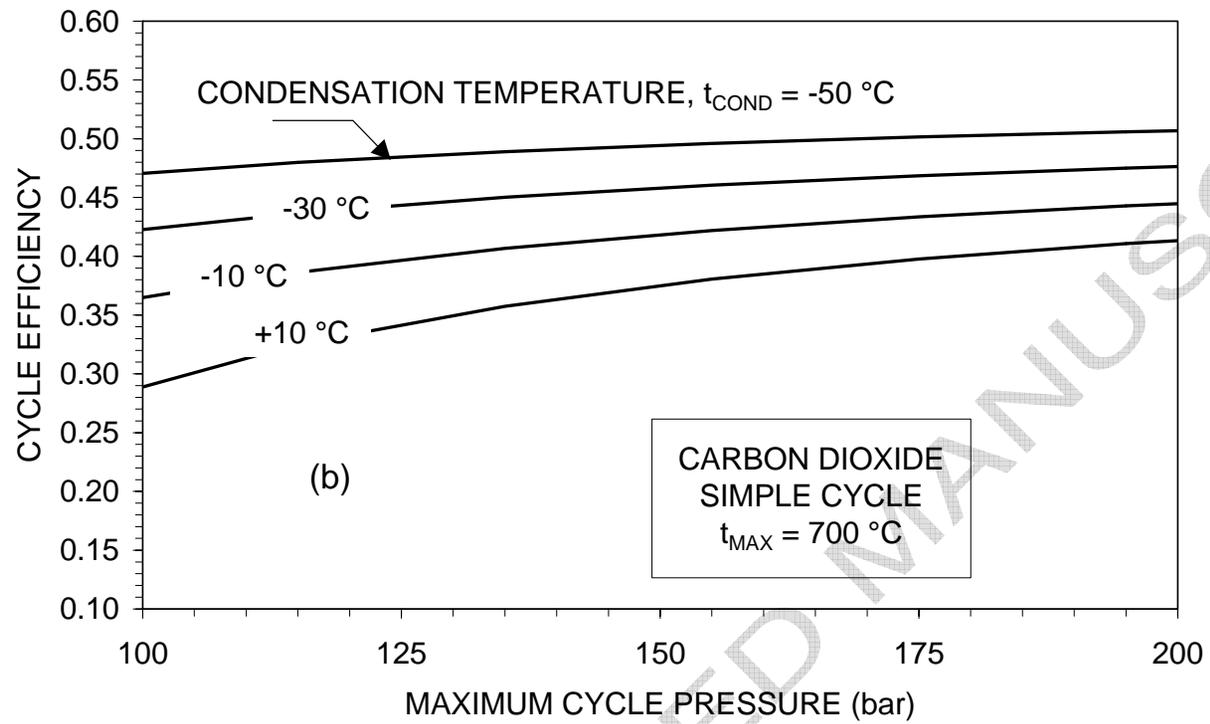


Figure 4.b

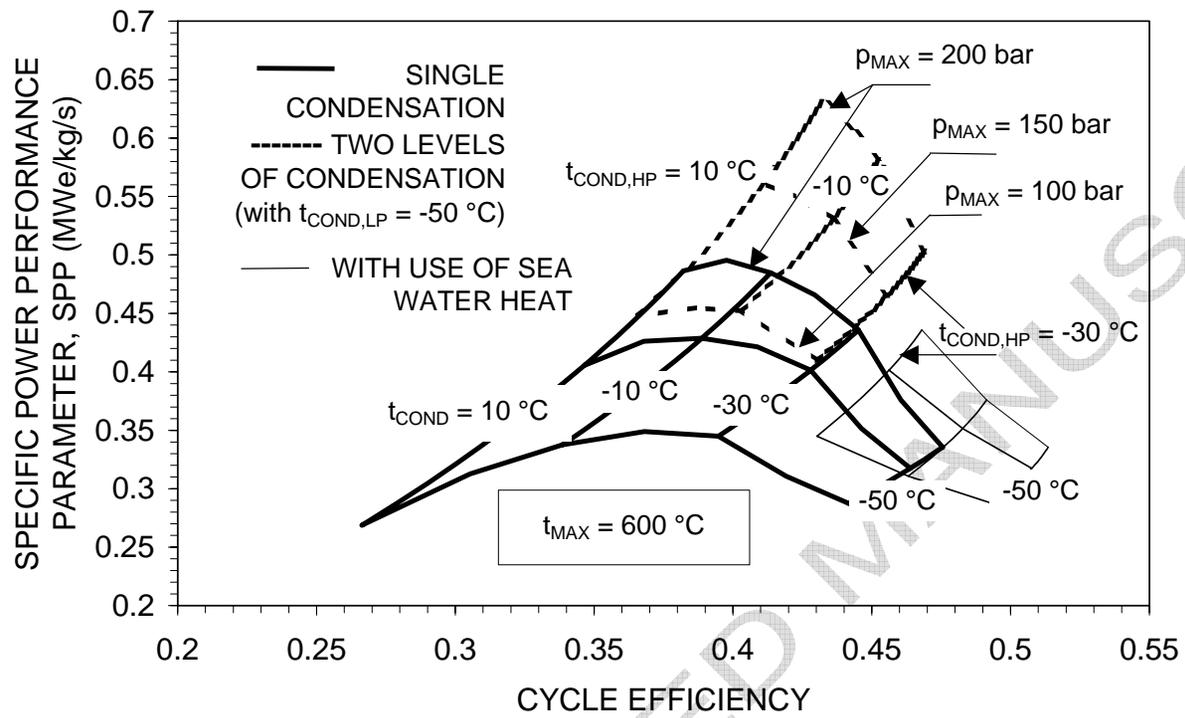


Figure 5

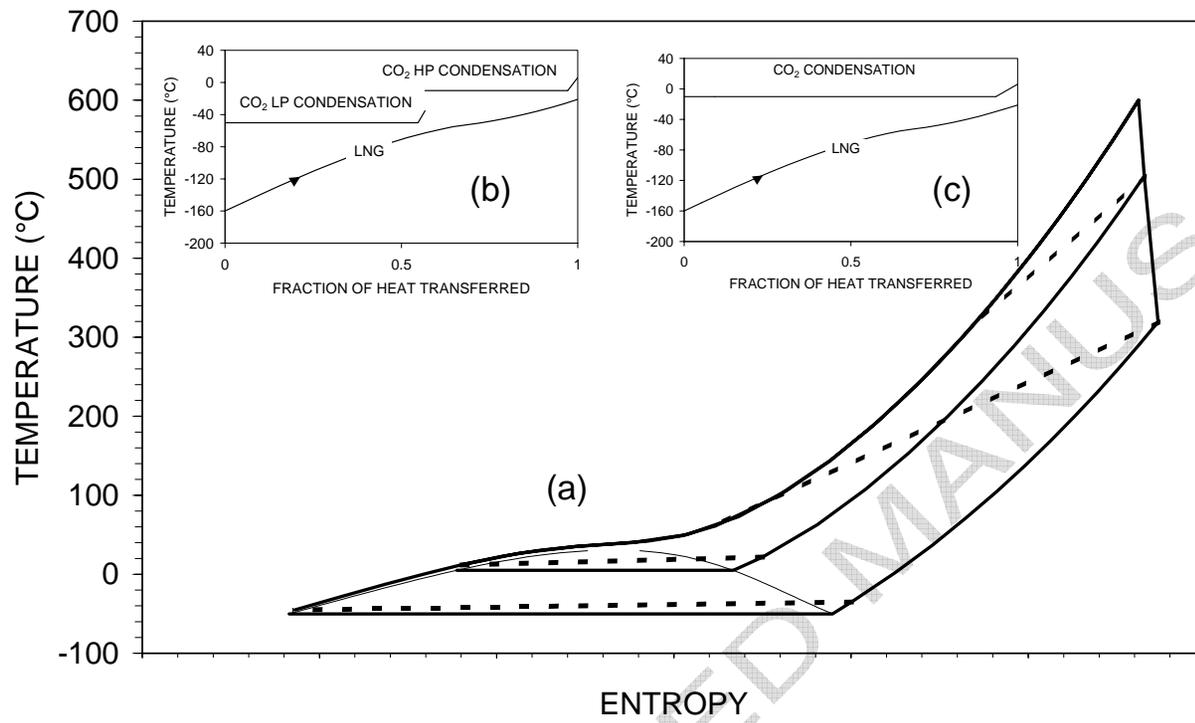


Figure 6

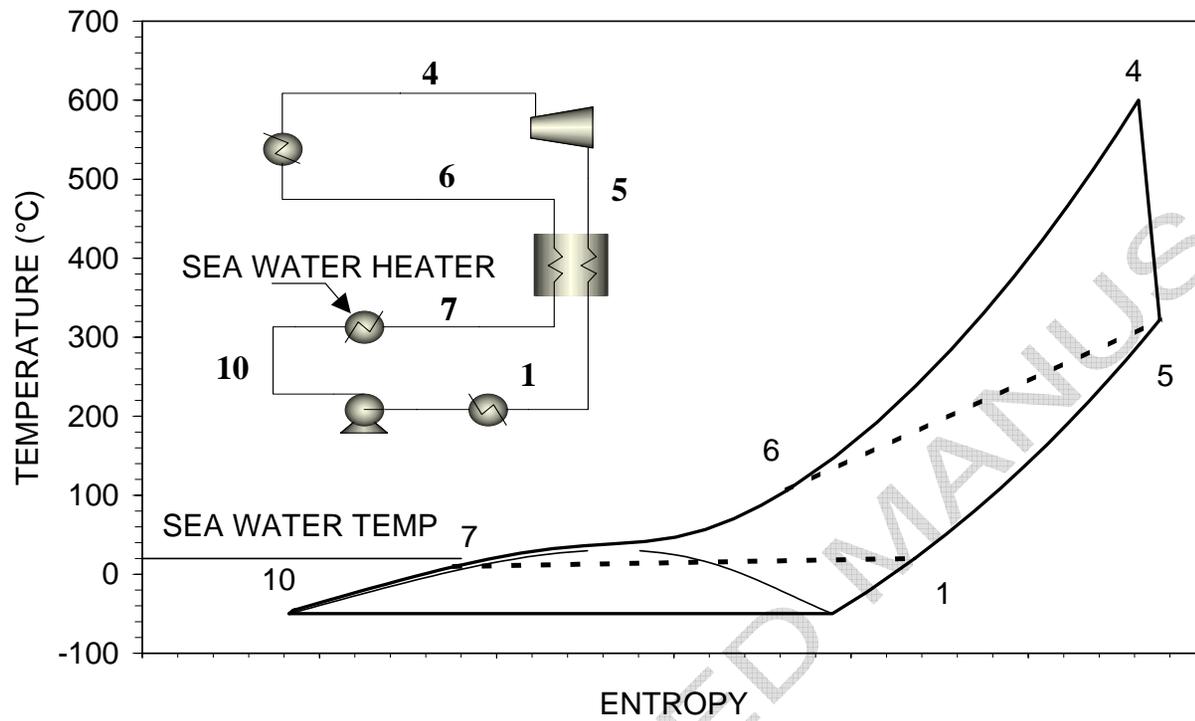


Figure 7

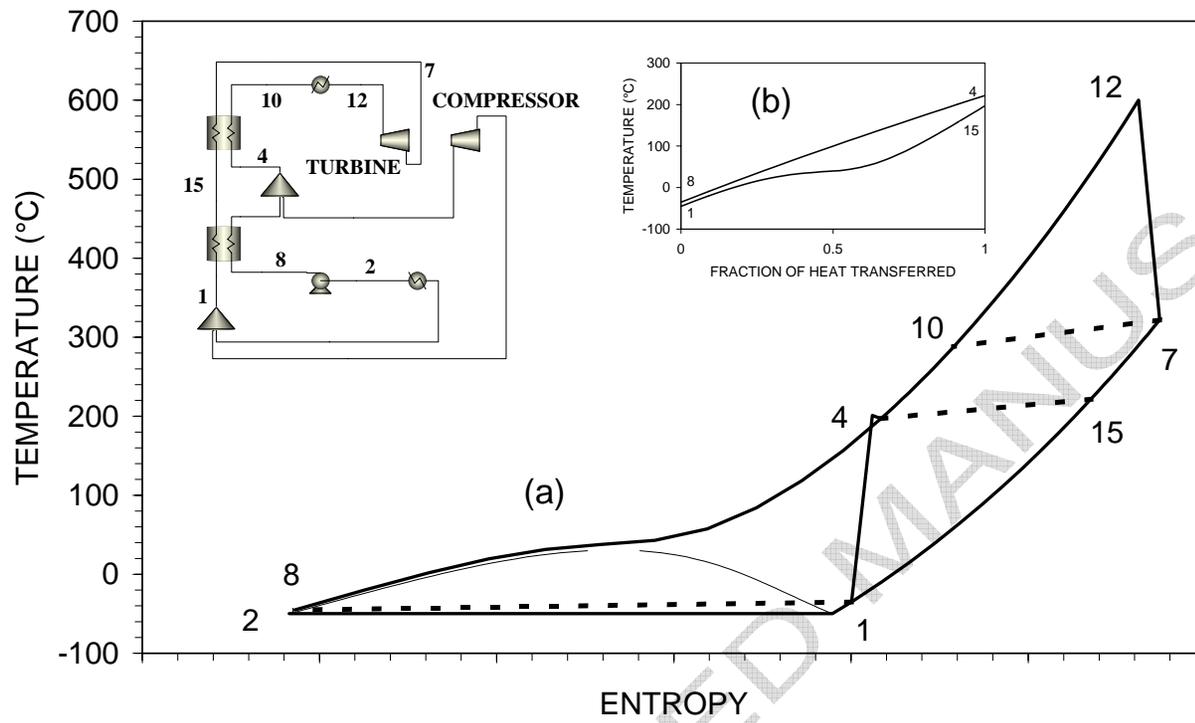


Figure 8

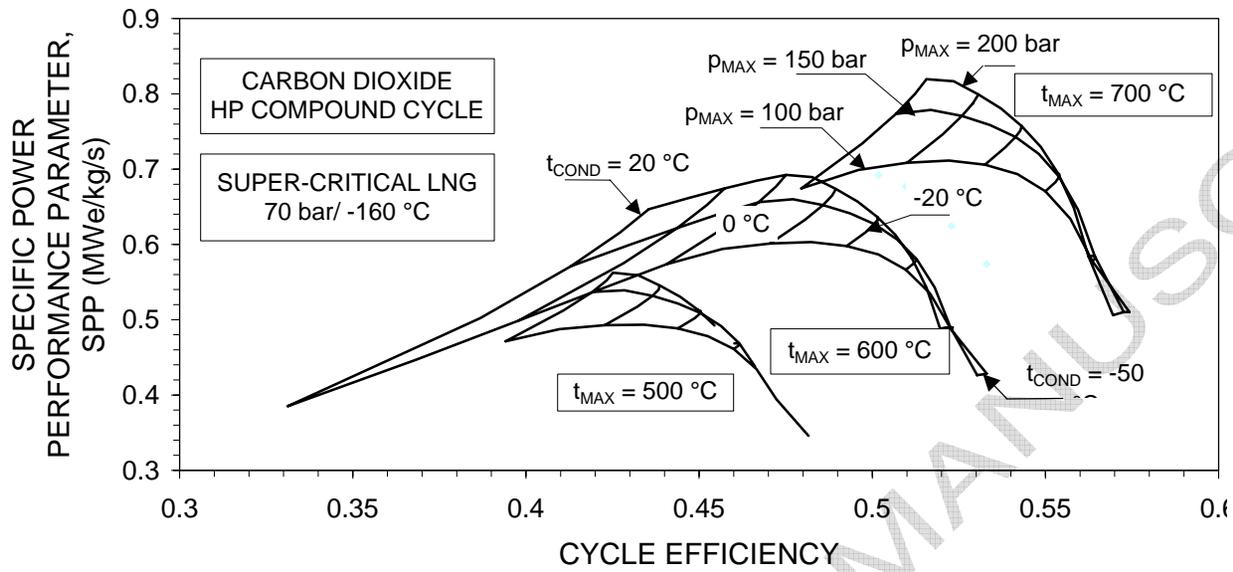


Figure 9

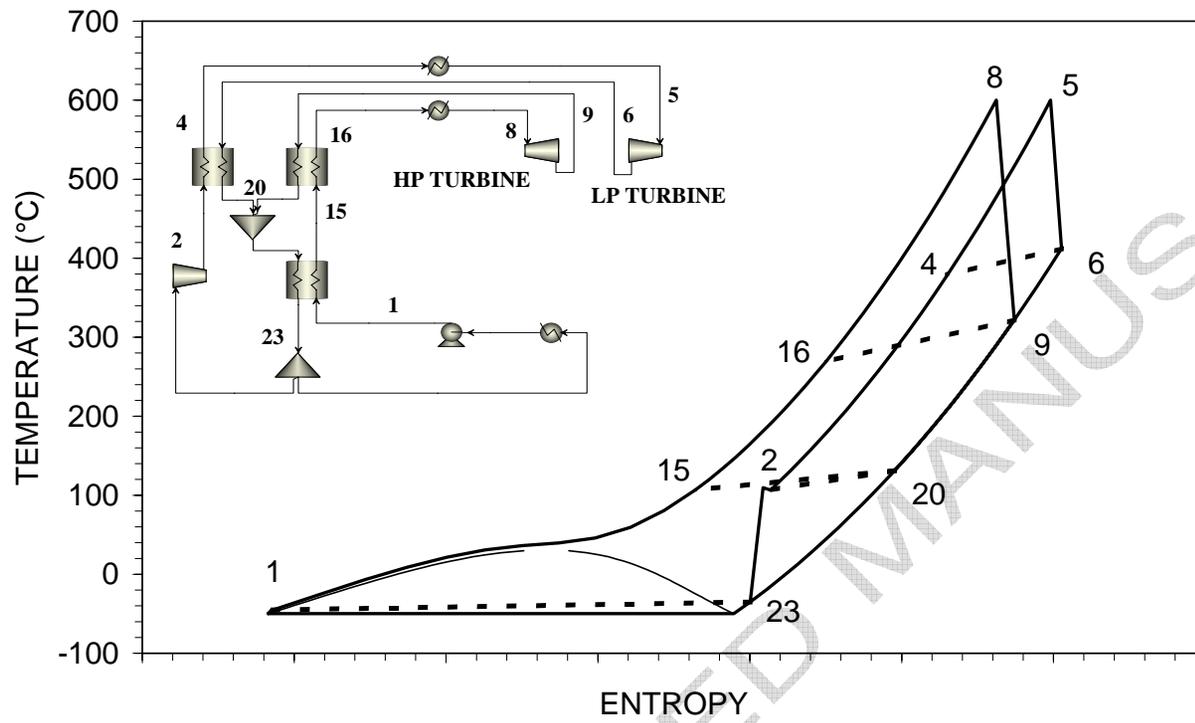


Figure 10

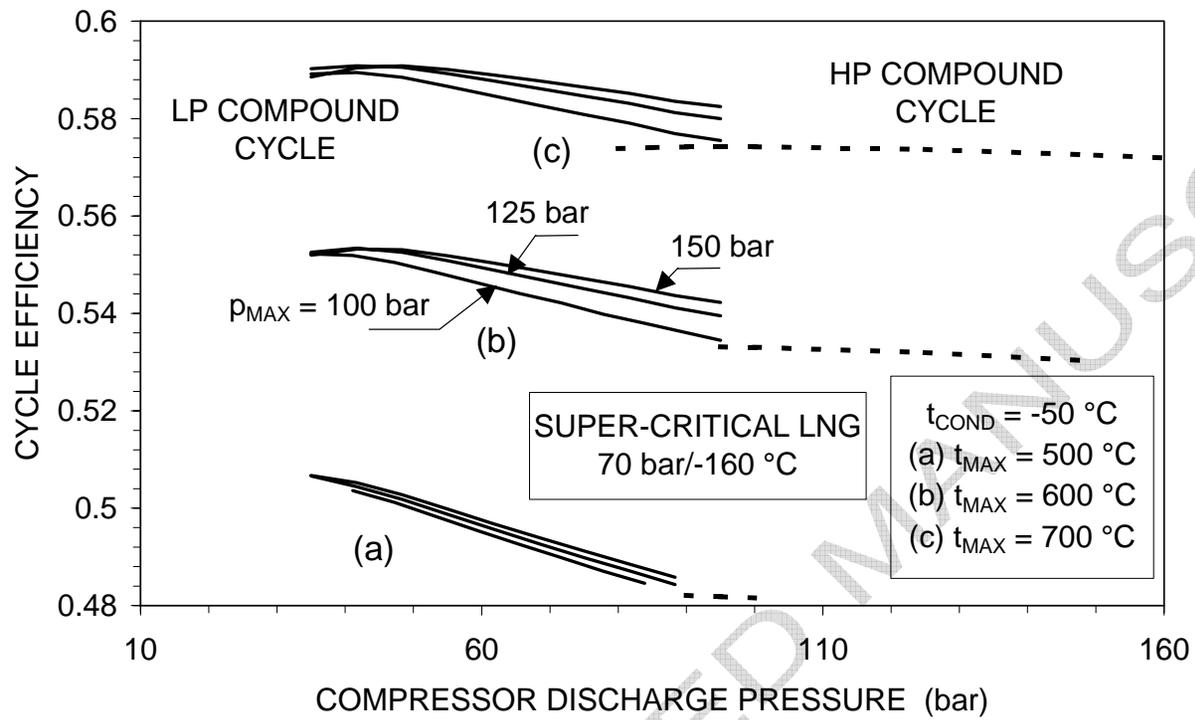


Figure 11

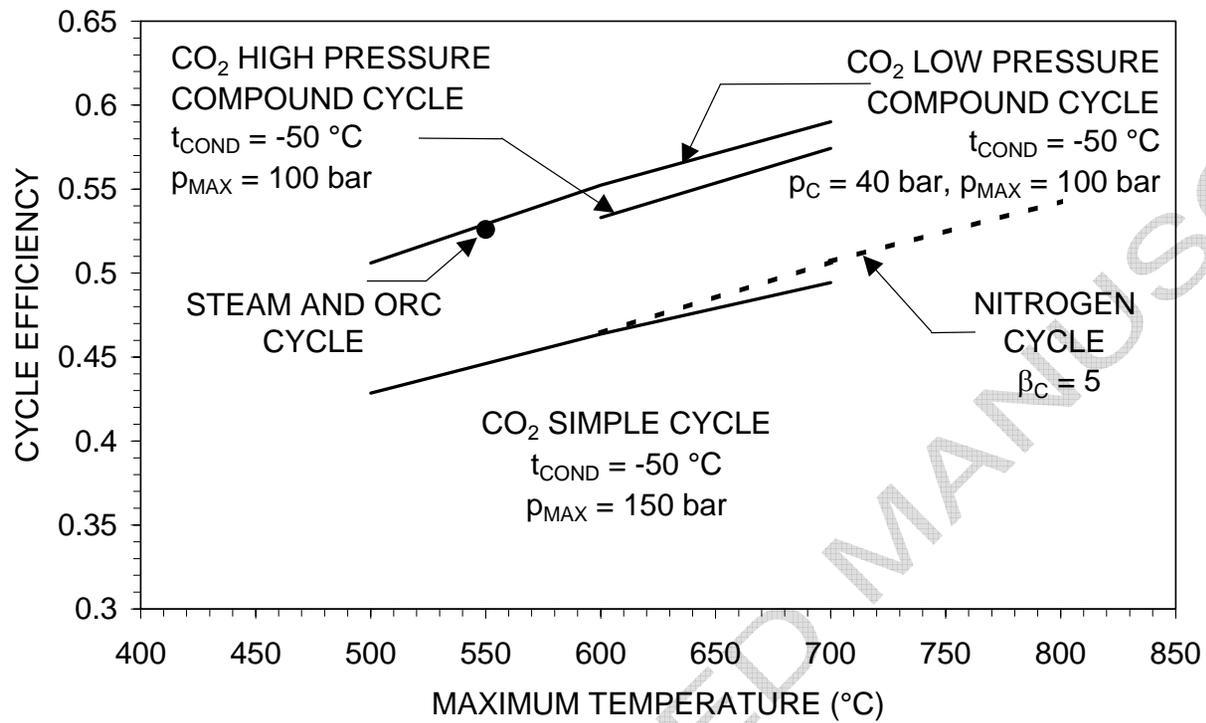


Figure 12