The assessment of alternative refrigeration solutions

John Buckingham*, CEng FIMechE

** BMT, UK*

* Corresponding Author. Email: John.Buckingham@BMTglobal.com

Synopsis

The provision of chilled water is a critical service in any warship as it ensures mission systems can operate reliably within suitably air conditioned spaces. A Chilled Water Plant (CWP) cools circulating clean filtered water through a refrigeration system which passes the heat to sea water (SW). Historically, the refrigerant employed has been damaging to the Earth's ozone layer but these were outlawed by the UN Montreal Protocol. Since then new refrigerants such as R134a have provided much improved ozone depletion but they have a high Global Warming Potential (GWP) if leaked to the atmosphere.

To address the GWP issue, the European directive 2006/40/EC, which went into effect in 2011, required all new cars on sale in Europe to use a refrigerant in its air-conditioning system with a GWP below 150. One group of the new refrigerants to address this required change are called hydrofluoroolefins (HFO) with the R1234yf being a leading candidate for general use with a GWP of 4. However, R1234yf is currently manufactured in limited quantities and so is quite costly. It also has mild flammability with toxic products when combusted. Although it is a close drop-in for R134a, it requires changes to the lubricating oil.

Paradoxically, carbon dioxide (CO_2) has long been used as a refrigerant and is often used for cold and freezer displays in shops. It is not flammable and has a GWP of 1, much lower than many alternative refrigerants. However with a critical point of 31°C and 71 bar, a CWP cooled by SW the refrigeration cycle would need to be transcritical, i.e. the use of CO_2 as a supercritical fluid.

To obtain a rounded assessment of the benefits of CO_2, versus R134a, the performance of two indicative CWP designs has been analysed. The Coefficient of Performance (COP) and other design and performance issues are identified.

The system arrangement and equipment selection issues are addressed, together with a high level consideration of the GWP in-service for different refrigerant leak/loss rates.

The ability to drive an absorption chiller plant (ACP) with the heat rejected from the CO_2 CWP is assessed and the energy efficiency and ship cooling benefits are assessed.

Keywords: Refrigeration; Global Warming Potential; Carbon Dioxide

1. Introduction

More than ever before, the Chilled Water System (CWS) plays an increasingly critical role in a warship. Mission systems, both sensors and weapons, [Buckingham, 2021] increasingly have higher power demands and with this comes a greater rejection of waste heat often on an *ad hoc* basis when the mission equipment is used at short notice and always with a need for instant availability.

Additionally, warships are operating worldwide on a range of tasking roles each with its own specific operational duties, but all with a need for a high availability of power and cooling. Worldwide operations bring a range of environmental conditions with SW temperatures from as a low as -2°C to local harbour temperatures over 40°C. Likewise ambient air temperatures can range from -20° and below through to 50°C. When air and sea temperatures are in these upper values finding reliable ways to sustain the full CWS capability whilst being able to dump heat to the heat sink that is the sea can be very challenging. To achieve an efficient cooling system capable of operation across such temperature ranges requires designs which have adequate margin and good low load performance.

Current CWS designs are still based on the same fundamental refrigeration cycle and design approach that has been employed since the 19th century. In a warship, chilled water is supplied to heat loads at 6-7°C and at full heat load, leaves at 12-13°C. Using the different cycle stages as shown in the sub-critical temperature – entropy chart in Figure 1, the CW heats and boils the refrigerant in the CWP (5-1) which is then compressed to a higher temperature and pressure (1-2). After the compressor, the superheated refrigerant is cooled to the saturated vapour pressure point (3) and then further condensed by the heat sink which is nominally four or more °Celsius below the refrigerant temperature (3-4) to the saturated liquid point (4).

Author's Biography

John Buckingham is the Chief Mechanical Engineer at BMT. A Fellow of the IMechE, he has over 38 years' experience in marine engineering systems design. He has designed and analysed hybrid power and propulsion systems for naval and commercial vessels and is currently involved with the modelling and analysis of energy saving technologies. He has been the technical lead for a wide range of technology studies and concept development work, specifically on studies relating to hydraulic fluid power and heat management systems.

The cooler refrigerant is still pressurised but is usually in liquid form before it then passes through an expander valve where it cools to a saturated gas (4-5) and is then ready to cool the incoming CW in the evaporator to repeat the cycle. Some transcritical cycles do not condense the refrigerant and it remains gaseous and its expansion is then managed by a throttle.

The refrigerant cycle, temperature-entropy chart for the current standard refrigerant, R134a, is shown in Figure 1. The two sets of R134a and CO_2 thermal properties have been digitised to allow the enthalpy and entropy to be identified for given values of temperature and pressure, and any other parameter value from the two other parameter values. Values are therefore sensibly accurate for the purposes of this study.

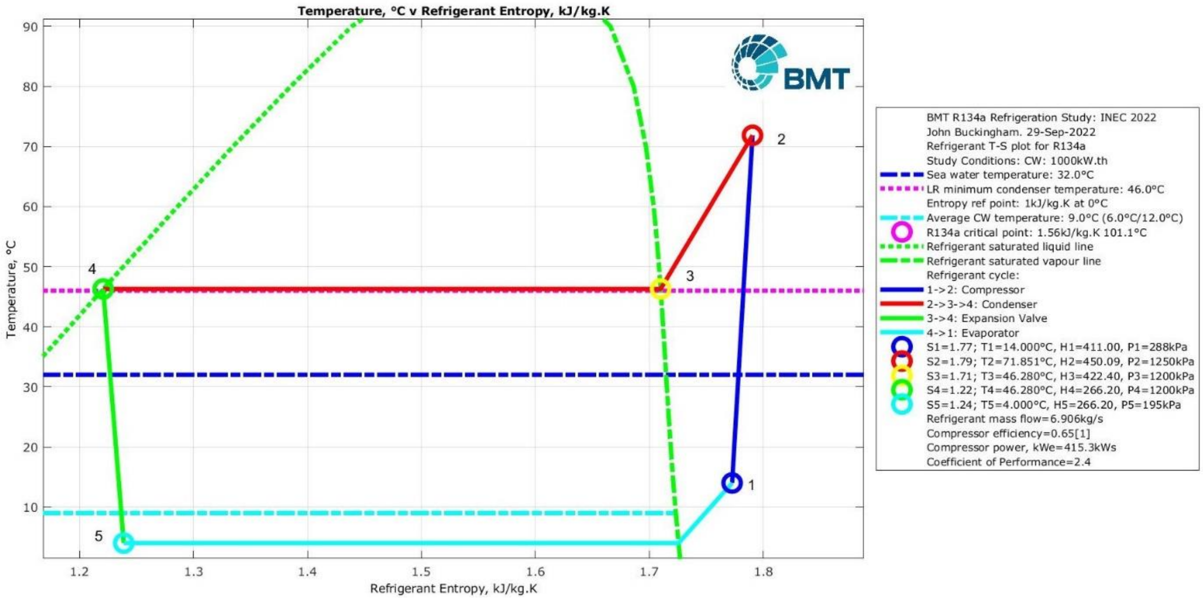

Figure 1. R134a Temperature – Entropy Chart

Figure 1 shows the simplified refrigeration cycle design for a standard maximum SW temperature of 32°C.

The minimum pressure at the refrigerant condenser is set at the saturated vapour pressure for a temperature of 46°C to meet the requirements of Lloyds Naval Rules (Lloyds 2022, Section 2, 2.7). This ensures there is always adequate heat rejection to SW for a range of anticipated excessive SW temperatures.

The refrigerant evaporator design temperature is set to 5°C below the CW average temperature of 9°C (i.e. at 4°C). Figure 1 shows how entropy efficiency affects the expansion valve transition (4-5) and the compressor (1-2) stages. The isentropic efficiency of the compressor is assumed to be 99%.

The maximum R134a temperature in this indicative cycle is 71.85°C. The critical point of R134a is 101°C, 4,060.3 kPa. Although, this study did not consider transcritical operations for R134a, it has been shown that R134a does not degrade at temperatures up 368°C [Calderazzi, 1997].

Figure 1 also shows 10°C superheat at the pre-compressor stage (1) to ensure the supply to the compressor inlet comprises dry gas.

The pressure-enthalpy chart for this cycle is shown in Figure 2. Nominal pressure drops values (<100kPa) are assigned for the flow through the evaporator and the condenser.

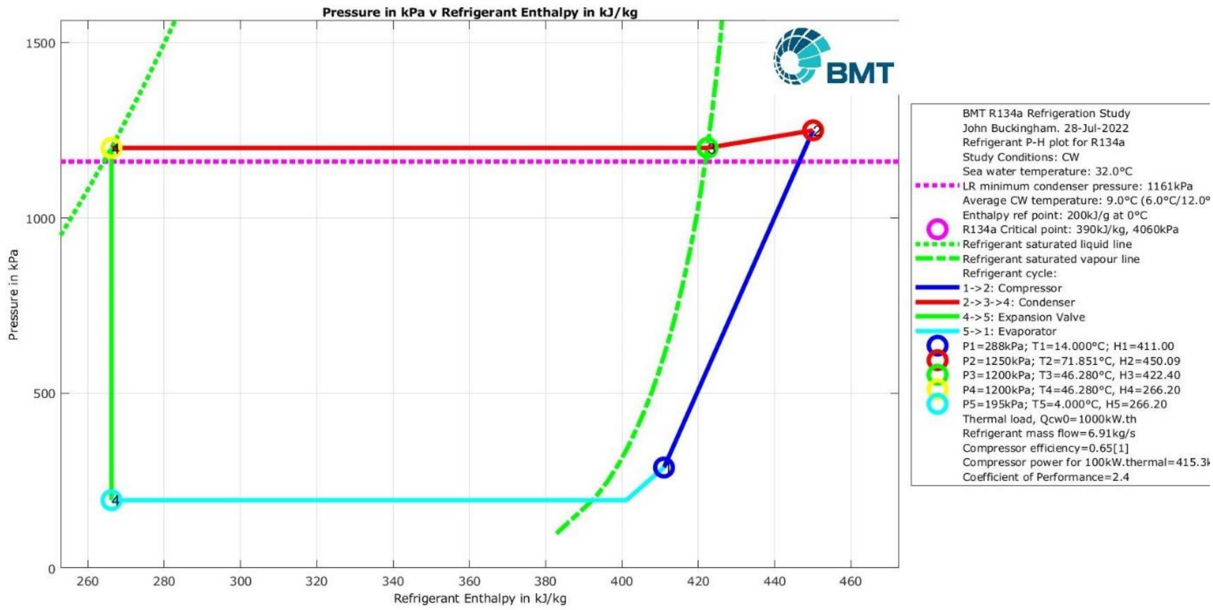

Figure 2. R134a Pressure-Enthalpy Chart

2. Environmental Considerations

2.1 Ozone Depletion

It was in 1985 that UK scientists from the British Antarctic Survey [Farman, 1985] alerted the world to the ozone hole over Antarctica. It was established that the worldwide use of refrigerants, largely based on chlorofluorocarbons or CFCs, leach chlorine and bromine which react with ozone in the upper atmosphere. The damage to the world's ozone layer was identified and addressed in remarkably quick time, by making the use of CFC outlawed by the UN Montreal Protocol in 1989. However as Figure 3 [NOAA, 2021] shows, the situation has stabilised but is hardly much better than 1995.

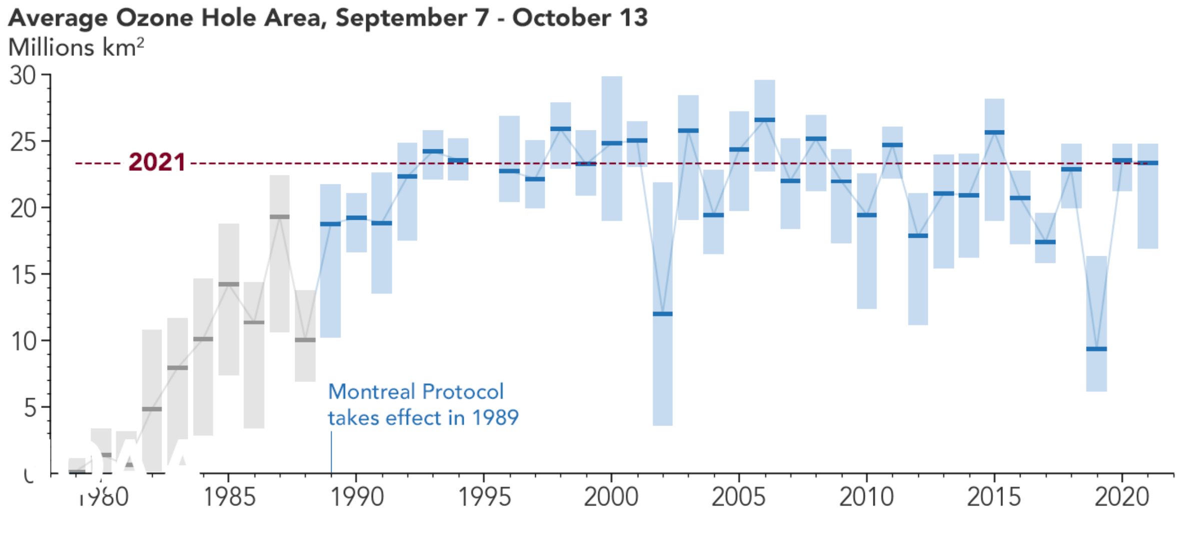

Figure 3. Antarctic Average Ozone Hole Size, 1980-2020

Since the Montreal Protocol was implemented, alternative new refrigerants such as the hydroflurocarbon (HFC), R134a (1,1,1 tetrafluoroethane) have provided much improved ozone depletion performance (ODP), with R134a having an ODP score of zero.

2.2 Global Warming Potential

However the range of HFC refrigerants often have a high Global Warming Potential (GWP) if leaked to the atmosphere. R134a has a 100-year GWP of over 1,300.

To address the GWP issue, the European Union directive 2006/40/EC, which went into effect in 2011, requires all new cars on sale in Europe to use a refrigerant in its air-conditioning system with a GWP below 150. One group of the new refrigerants to address this required change are called hydrofluoro-olefins (HFO) with the R1234yf being a leading candidate for general use with a GWP of 4. However, R1234yf is currently manufactured in limited quantities and so is quite costly. It also has mild flammability with toxic products when combusted, thus it is in the ASHRAE safety group of A2, [ASHRAE, 2021], where A is a designation for low toxicity. Designation 2 is for lower flammability refrigerants with a maximum burning velocity less than 10cm/s

Although R1234yf is a close drop-in for R134a, it may also require changes to the CWP's lubricating oil.

Paradoxically, carbon dioxide (CO_2) has long been used as a refrigerant and is increasingly being used for cold and freezer displays in shops. It is not flammable and has a GWP of 1, much lower than many alternative refrigerants. However with a critical point of 31°C and 71 bar, for a CWP the refrigeration cycle would need to be transcritical for SW cooling at condensing temperatures (Tsw) over 35°C.

Table 1 shows a set of refrigerants for medium temperature applications (i.e. CW) with their principal characteristics [BOC, 2012]. The ASHRAE safety group designation provides an indication of the relatively safety of the refrigerant.

Ammonia is always spoken of as a candidate refrigerant but it has a safety group of B2 due to its toxicity and is unlikely to be acceptably safe if it leaks in the event of battle damage.

Table 1. Product Data Summary for Key Refrigerants

Refrigerant Description	**Critical Temperature /Pressure[3] (°C/bara)**	**ODP UNEP (2006 (R11=1)**	**GWP =(100 year), [IPCC, 2007]**	**Flammability ASHRAE safety group**	**Comments**
R22	96/50	0.055	1,810	A1	HCFC. Previously used in RN before R134a
R134a	101 / 41	0	1,300/ 1,430	A1	HFC
R1234yf	95/34	0	4	A2	HFO, Opteon® YF, Solstice™ yf
R1234ze	109/36	0	6 (<1?)	A2/A2L	Solstice™ ze trans-1,3,3,3-tetrafluoro-1-propene
R744	31 / 74	0	1	A1	CO_2
R445A	85.6/46.5	0	146	A2L	R744/134a/1234ze(E) (6.0/9.0/85.0)

In Table 1, A2L designates a lower flammability refrigerant with a maximum burning velocity of <=10cm/s.

There are no suitable HFC with a GWP below 150, so the focus is now on the uptake of HFO, or refrigerant blends such as R445A and natural substitutes such as CO_2 (R744).

3. Comparative Assessment

To obtain a rounded assessment of the benefit of the refrigerants, R134a and CO_2, the performance of two indicative CWP designs has been analysed at the standard high Tsw of 32°C. The COP and other design and performance issues are identified.

The system arrangement and equipment selection issues are addressed, together with a high level consideration of the GWP in-service for different refrigerant leak/loss rates.

The ability to drive an absorption chiller plant (ACP) with the heat rejected from the CO_2 CWP is assessed and the energy efficiency and ship cooling benefits are assessed.

To allow a comparison of the total GWP of each installation, it is assumed the refrigeration plant has a refrigerant capacity of four minutes maximum refrigerant flow.

4. R134a Refrigerant

As Figure 1 shows for a 1,000kW.th CW duty heat load, there is a required refrigerant flow of 6.91kg/s, giving an assumed total refrigerant capacity of 1,656kg. With a GWP of 1,300, this is equivalent to the release to atmosphere of 2,152,800kg.CO_2, or 2,152 tonnes.CO_2.

5. Carbon Dioxide Refrigerant

To permit a ready comparison with the R134a design, a transcritical refrigerant cycle using CO_2 as the refrigerant was defined, for a 1,000kW CW heat load and a Tsw of 32°C. Figure 4 provides a direct comparison between the use of R134a and CO_2 refrigerants by showing the cycles on a pressure-enthalpy chart.

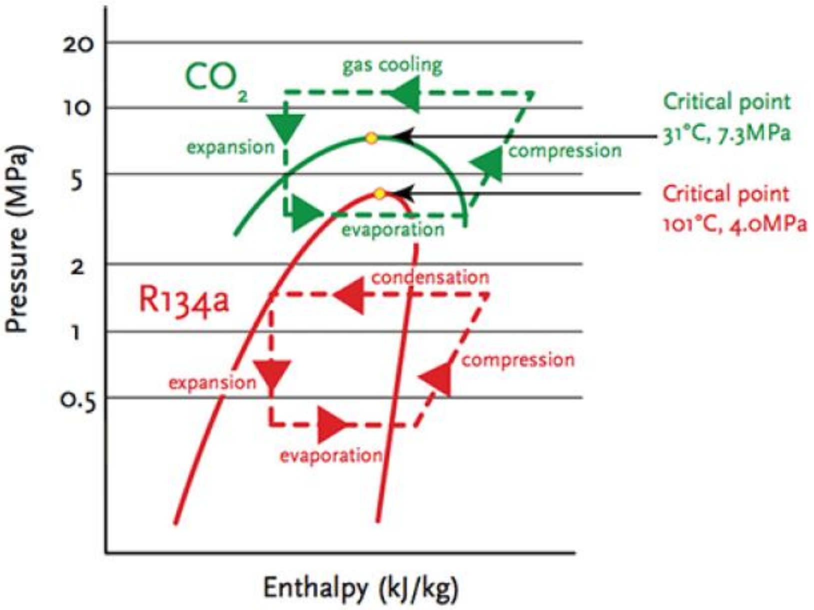

Figure 4. A basic comparison of subcritical R134a refrigeration and a transcritical CO_2 cycle [CIBSE , 2021])

Figure 4 shows that the CO_2 cycle will need to operate in a transcritical mode at pressures above 7.3MPa (73bar) whereas the R134a can operate at pressures below 4.0MPa (40 bar) thus reducing the required compressor power demand associated with the latter system.

In the example in Figure 4, the CO_2 is permitted to liquify at the end of the condenser as it passes to the left of the critical point. To meet Lloyds Register (LR) requirements for reliable heat transfer to warm SW, the condenser/gas-cooler temperature at exit is to be maintained above the LR requirement of 46°C and its equivalent pressure.

As a temperature of 46°C is above the critical point for CO_2, the system will be designated “transcritical”. The cooling gas will therefore not condense as it loses heat to the SW. Between the compressor exit and a point in the throttle/expander valve, the fluid is to be considered as a homogenous ‘super-critical’ fluid.

At temperatures above the critical point, the latent heat of vaporisation is zero between liquid and the super-critical’ fluid as shown in Figure 5. As the CO_2 refrigerant is to stay above 46°C when in the condenser, or gas cooler as it does not condense, the indicative CO_2 cycle shown here stays to the right of the critical point and the CO_2 is a supercritical fluid in the gas-cooler.

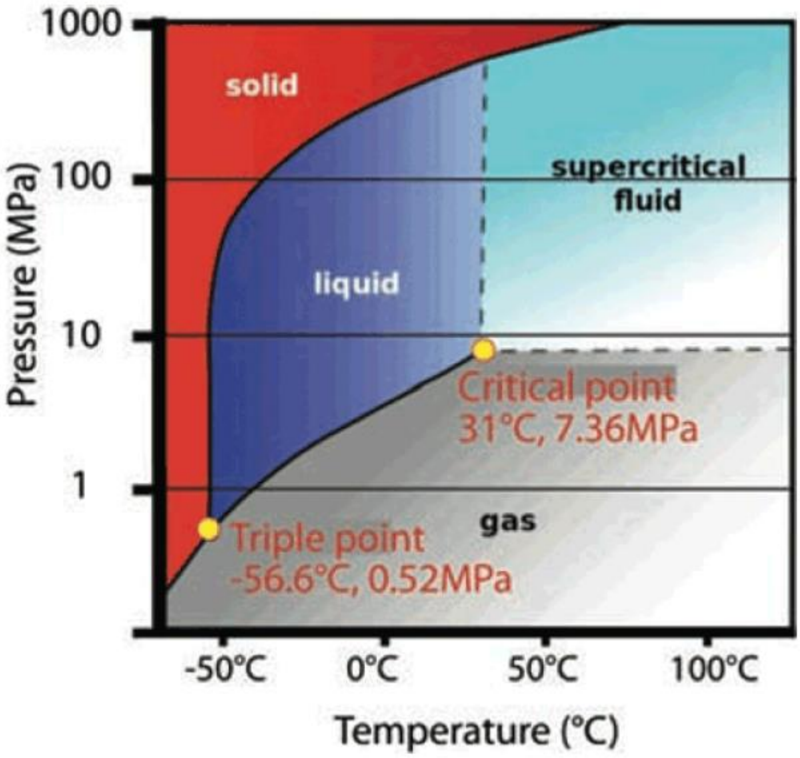

Figure 5. The phases of Carbon dioxide, [CIBSE, 2021]

The T-S chart in Figure 6 shows how the condenser temperature stays above the LR requirement of 46°C between 2-3, and then cools through the saturated state boiling curve as it passes through the throttle/expander valve (3-4).

The compressor operates at 11.0MPa (110 bar) at point 2 so there is a much greater requirement for high pressure machinery, which has implications for increased pressure integrity and the consequential weight and size of the equipment.

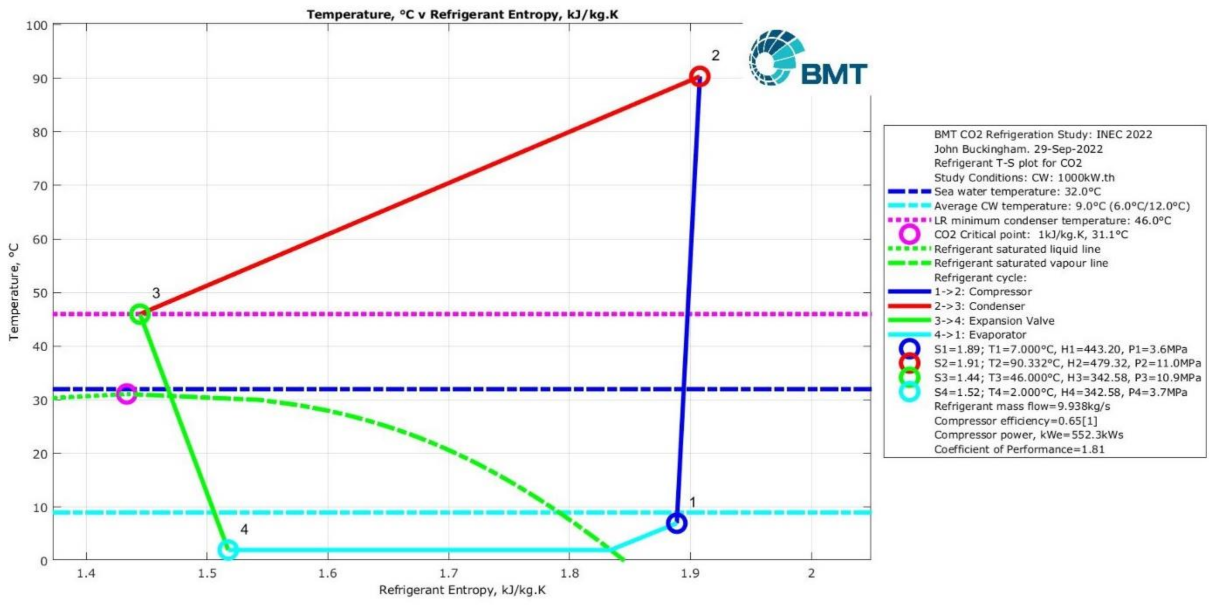

Figure 6. CO_2 Temperature – Entropy Chart

To transfer 1,000kW.th from the CW to the refrigerant, there is a CO_2 flow requirement of 9.94kg/s. With a reservoir sized to four minute refrigerant flow, this makes 2,386kg.CO_2, or an extra 44% of the R134a installation.

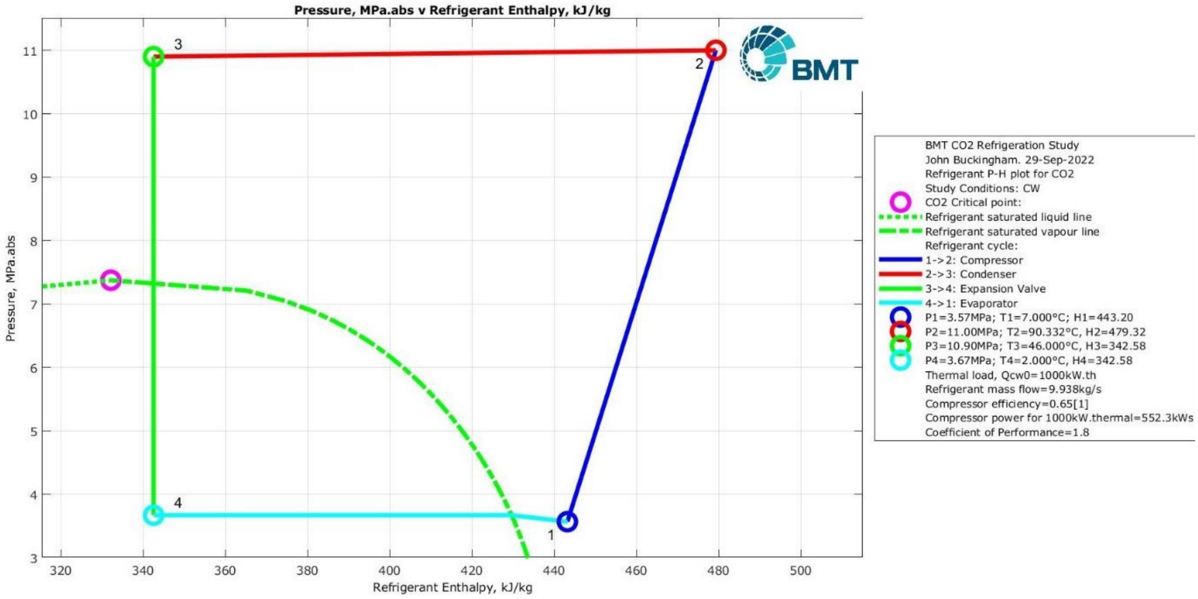

Figure 7. CO_2 Pressure-Enthalpy Chart

Figure 7 shows the pressure–enthalpy plot with the indicative pressure drops across the evaporator and condenser shown. The key parameters for this indicative cycle are shown in Table 2.

Table 2. CO_2 Cycle Design Parameters

Parameter	Value
1. <u>Post evaporator</u>	<u>Pre-compressor</u>
Pressure, P, MPa	3.57
Temperature, T, °C	7.0
Enthalpy, H, kJ/kg	443.20
Entropy, S, kJ/kg.K	1.88
2. <u>Post-compressor</u>	<u>Pre-gas cooler</u>
Pressure, P, MPa	11.00
Temperature, T, °C	90.33
Enthalpy, H, kJ/kg	479.32
Entropy, S, kJ/kg.K	1.90
3. <u>Post gas-cooler</u>	<u>Pre-throttle/expander valve</u>
Pressure, P, MPa	10.90
Temperature, T, °C	46.00
Enthalpy, H, kJ/kg	342.00
Entropy, S, kJ/kg.K	1.44
4. <u>Post-throttle/expander valve</u>	<u>Pre-Evaporator</u>
Pressure, P, MPa	3.67
Temperature, T, °C	2.00

Parameter	Value
Enthalpy, H, kJ/kg	342.58
Entropy, S, kJ/kg.K	1.52
Refrigerant mass flow,kg/s For 1,000 kW.th CW duty load	9.94
Compressor efficiency, BMT assumption	0.65
Compressor power demand, kWe	552.30
Coefficient of Performance,	1.81

The pressure-enthalpy chart in Figure 7 shows the slight pressure drops down the condenser and evaporator lines as well as the adiabatic expansion of the gas through the throttle valve (3-4).

As the mass flow is 9.938kg/s, the CO_2 design will have a heavier weight of refrigerant than the R134a.

The study results show that when used alone as the source of CW cooling, the CO_2-based CWP has a poor COP of 1.81 due to the classification society requirement to have the gas-cooler temperature at 46°C or above. This is to ensure the refrigerant can be cooled at all foreseeable SW temperatures.

The combination of this limitation and the low critical point for CO_2 means that a transcritical system is required which is wholly gaseous (i.e. supercritical fluid) in the gas cooler and operates to the right of the critical point. As a result the COP of 1.81 is worse than the 2.4 achieved with the R134a refrigerants and due to the high cooling temperature required, it results in a solution which can be considered to be impractical when used alone.

6. Absorption-based Chiller Plant

An Absorption-based Chiller Plant (ACP), [Buckingham 2021], uses a 90°C heat source to drive internal evaporation of the working fluid and thus when also combined with SW cooling create a capability to cool chilled water.

In more detail, an ACP uses waste heat to pressurise a refrigerant and absorber mixture so that the refrigerant is pressurised and evaporates away from the absorber (lithium bromide). The refrigerant is cooled by a heat sink (SW) then allowed to expand through an expansion valve where it cools due to adiabatic expansion. The much cooler refrigerant at a low pressure is then used to cool the CW from 14°C to 7°C before it enters the tank where it recombines with the absorber. The diluted absorber is pumped to the generator tank where the refrigerant is evaporated off to re-start the cycle. This process is shown in Figure 8 which shows how the ACP is configured. The ability to use water as the refrigerant requires a low-pressure side so that the water is vapour at low temperatures.

Figure 8 shows how an arrangement of the flows within an ACP. The ACP takes a waste heat source (Qgen) to evaporate the refrigerant so that it is disassociated from the absorber. This together with the cooling stream (Qc & Qa out), (SW for a ship) then allows a chilled fluid stream (negative Qe) to be generated which can be used as a supplement to the supply from the ship's CWP.

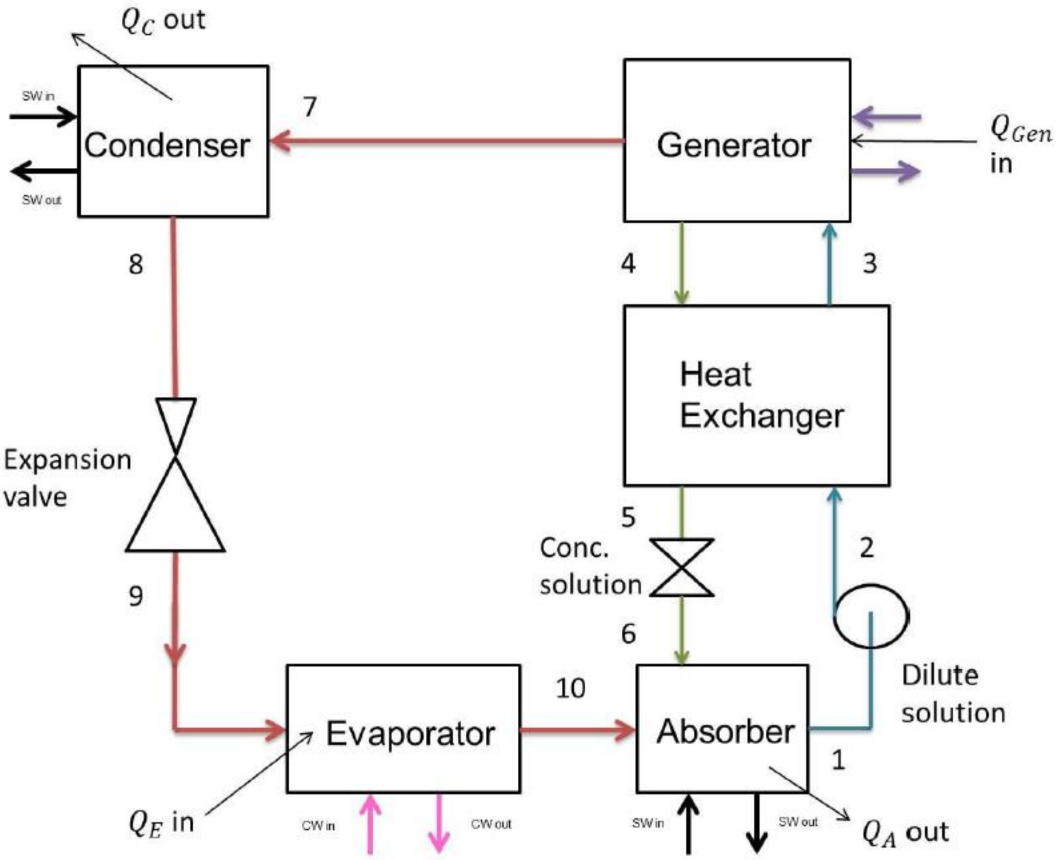

Figure 8. Schematic of an Absorption Chiller Plant Cycle

The total heat rejection by the CO_2 gas cooler is 1,359kW.th of which 308kW.th (pro rata by temperature differentials) can be used in the range of 90-80°C to drive the hot water supply side of an ACP. With a COP of 0.71, the ACP can then provide an additional 219kW.th CW cooling capacity. When combined with the 1,000kW.th capacity of the CO2 CWP, the combined CO_2 refrigeration plant with an ACP then has an overall COP of 2.21, thus approaches to the 2.4 of the R134a CWP.

7. Comparison

7.1 Performance

The CO_2 design has a compressor power requirement of 552kWe, versus 415kWe (See Figure 1) for the R134a design. Assuming a DG power supply with a specific fuel consumption of 0.22kg/kWh, 200 days at sea per year on full load, the CO_2 based refrigerant system would emit an extra 662 tonnes CO_2 pa due to fuel combustion compared to the R134a-based system (the ACP pumping loads being very small in comparison).

When the CO_2 CWP design is combined with an ACP, the benefit of the additional CW from the ACP, increases the COP from 1.81 to 2.21. The excess CO_2 emissions due to fuel consumption to make 1,000kW.th CW, reduce to 543 tonnes.CO_2 pa. This amount of CO_2 is ~25% of the CO_2 locked into the R134a tank capacity of 2,153 tonnes.CO_2.eq.

7.2 Ship Impact

The volume of the individual equipment onboard a warship can affect the ease of access for inspection and upkeep, and for the ability to replacement defect equipment with ease. The weight of equipment when accumulated with the weight of other naval engineering equipment can also affect the ship's stability and its payload, i.e. the scope for addition fit of mission systems towards its mid-life update.

Table 3 shows the weight, volume and power demand comparison for the baseline R134a CWP and the combined CO_2 CWP and its supporting ACP. These values have been estimated from a set of equipment information and will vary with different equipment suppliers. Power demand is estimated for the equipment operating at its full load condition.

Table 3. Comparison of equipment weight, volume & power demand

Parameter	**Equipment Volume, m³**	**Equipment Weight, tonnes**	**Power Demand, kWe**
Baseline R134a CWP	33.81	14.25	351
CO_2 CWP	190.51	60.24	552
ACP	13.50	6.70	10 (nominal)
CO_2 CWP & ACP combined	204.00	67.00	562
Differentials to baseline	+170	+53	211

Table 3 clearly shows that the adoption of a transcritical CO_2-based CWP with an ACP would lead to a much heavier and bulkier ship installation. The equipment volume is six times the R134a design and it is over four times heavier.

8. Conclusions

A CO_2-based CWP operating with an ACP provides a thermodynamic cooling performance which is comparable to that of an R134a CWP but with a higher power demand and extra, high-pressure rated equipment leading to a heavier and larger installation.

Due to the additional power demand of CO_2-based CWP , the extra operating CO_2 emissions due to the DG sets will offset any benefits associated with the use of a refrigerant with a low GWP, providing there are not repeated, accidental leaks of the R134a to atmosphere.

The adoption of more advanced refrigerant such as R1234yf with a much lower GWP will further reduce the risk of damage to the Earth's ozone layer through accidental leakage whilst providing an acceptable coefficient of performance.

9. Acknowledgements

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The views expressed in this paper are that of the author and do not necessarily represent the views and opinions of BMT. The author is grateful to BMT for the time and resources made available to allow this study to be undertaken.

10. References

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