國立交通大學
National Chiao Tung University

CO_2冷媒技術與應用發展

王啟川, PhD, 國立交通大學機械工程系教授

Fellow ASME, Fellow ASHRAE

***e*-Mail ： ccwang@mail.nctu.edu.tw**

Tel: 03-5712121 ext. 55105

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致謝

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報告內容

- CO_2 系統背景與現階段發展現況
- CO_2冷媒之物性
- 天然冷媒之理論 COP
- CO_2冷媒之熱傳特性
- 潤滑油對CO_2熱傳特性之影響
- CO_2 系統特性
- 提昇CO_2 系統性能之一些做法
- 簡單總結

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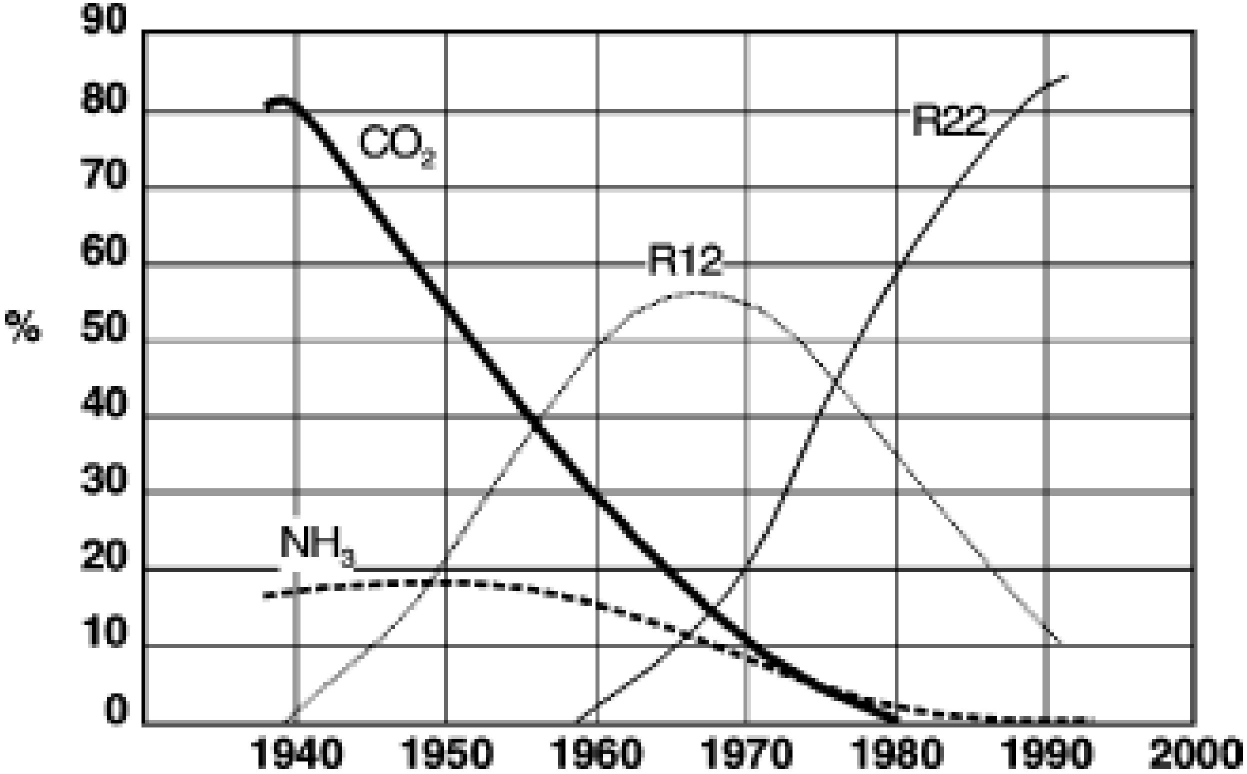
20世紀之 CO_2 系統應用



Stera A. Ammonia refrigerating plant on reefer ships. Introduction to ammonia as a marine refrigerant. Lloyd's Register Technical Seminar, London; 1992.

目前CO_2系統市場現況－日本

二氧化碳熱泵熱水器自2001年進入市場以來，銷售量持續上升。受經濟危機影響，2008年產品的銷售量雖與預期有些差距，但仍超過51萬台。按市場保有量計算，2008年家用二氧化碳熱泵熱水器在日本市場累計售出約180萬台；預計2010年銷售量約為80萬台，累計將達到320萬台。而原先預測的2010年產量達到100萬台、累計銷售量達520萬台的目標，估計需要推遲2年左右才能實現。

目前CO_2系統市場現況－歐洲

除了二氧化碳汽車空調之外，採用二氧化碳跨臨界迴圈技術的家用冰箱、商用冷藏冷凍設備已經投入市場多年。同時，一些歐洲企業目前正在進行二氧化碳跨臨界技術在熱泵應用領域的研究開發工作。2008年10月在德國紐倫堡舉行的Chillventa展覽會上，德國Thermea公司發佈了3個商用型二氧化碳熱泵產品的資訊，部分型號既可以供熱也可以製冷，用於供熱時額定COP值高達5.5W/W。在ECOBUILD 2009展覽會上，兩家英國企業分別展出了二氧化碳熱泵熱水器。英國ICS公司與義大利DeLonghi公司合作，展出了商用型產品，最高出水溫度可達90℃；而英國Stroma LZC公司展出的商用型產品出水溫度可超過80℃，COP值高達5.8W/W。

目前CO_2系統市場現況－美國與中國

近期美國採暖、製冷和空調工程師協會(ASHRAE)發佈了關於二氧化碳系統資料，認為高壓的二氧化碳系統會使得工質密度更高，有利於取得更好的運行效果。美國開利公司(Carrier)採用二氧化碳亞臨界迴圈技術的熱泵型空調機組已在歐洲地區進行了多年的場地試驗運行，美國民間環保機構綠色建築聯盟(GBA)也在積極推進二氧化碳系統在採暖熱泵領域的應用。

中國現已有3家壓縮機公司正在進行二氧化碳跨臨界迴圈熱泵系統專用壓縮機的開發工作。上海日立、西安慶安和廣東美芝目前均具備提供用戶樣品機的能力。GB/T 23137-2008《家用和類似用途熱泵熱水器》已經發佈，將於2009年9月1日起開始實施，該標準以附錄形式對二氧化碳熱泵熱水器提出了要求，主要參照了日本製冷空調協會發佈的JRA 4050《家用熱泵熱水器》的內容；同時，用於二氧化碳跨臨界迴圈熱泵裝置專用壓縮機的國家標準也在制定之中。與此同時，中國企業生產的採用二氧化碳跨臨界迴圈技術的家用電冰箱等產品已經開始進入市場。

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主要CO_2發展國家

國家	CO2系統發展情況
Norway	持續穩定朝向CO2 製冷系統發展，目前至少有14台CO2 製冷系統裝設於市場。
Sweden	瑞典是第二大裝設CO2製冷系統的國家，目前至少共43台CO2系統裝設在大賣場。CO2製冷系統製造商為Green & Cool與Epta。
Denmark	Denmark是裝設CO2製冷系統最多的國家。目前有超過100台的CO2製冷系統裝設於市場。
UK	目前積極朝向CO2製冷系統發展。
Germany	Epta與Carrier為德國主要CO2製冷系統製造商，此兩家的系統均已裝設在德國超級市場。 *高力於7/31寄K095C*72C-GB8M一台給德國Carrier測試，Carrier預計於9月進行測試。
Switzerland	至少有30台CO2 製冷系統安裝在超級市場，製造商分別為Carrier、Green & Cool、Goetz、enEX。
France	至少有6台CO2 製冷系統安裝在France，正持續在成長中。
Spain	CO2技術發展的進度緩慢，其當地製造商同時表示對CO2系統設計發展並不感興趣。
Italy	2008年總共有22台CO2 製冷系統安裝，最近又有30台CO2製冷系統的安裝，製造商為Carrier。
Belgium	至少有13台CO2 system安裝在Belgium。
Canada	CSC Group製造出使用於Supermarket Eco2 system的製冷系統，目前已安裝在市場。
Greek	Frigoglass製造CO2-R744的販賣機，call EcoCool。http://www.frigoglass.com/
USA	CliTech International Energy System Co.,Ltd發展出CO2 Heat Pump。
China	中國雖未受到溫室氣體排放限制，但目前已有廠商開始研發CO2 Heat Pump。
Japan	2008/04～2009/03Eco Cute日本當地CO2 Heat Pump出貨量達500,000台。此數字將會持續成長。Eco Cute主要製造商：Daikin, Sanden, Mitsubishi Electric, Denso, Panasonic, Fujikoki.

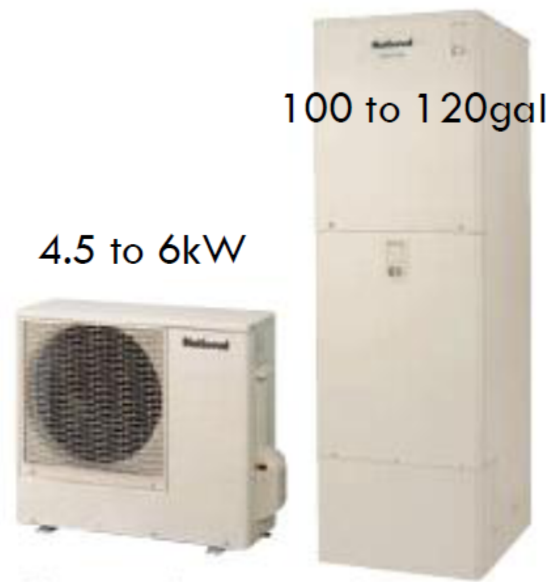
典型 CO_2 Heat Pump Water Heater

Move ambient heat into Hot water.

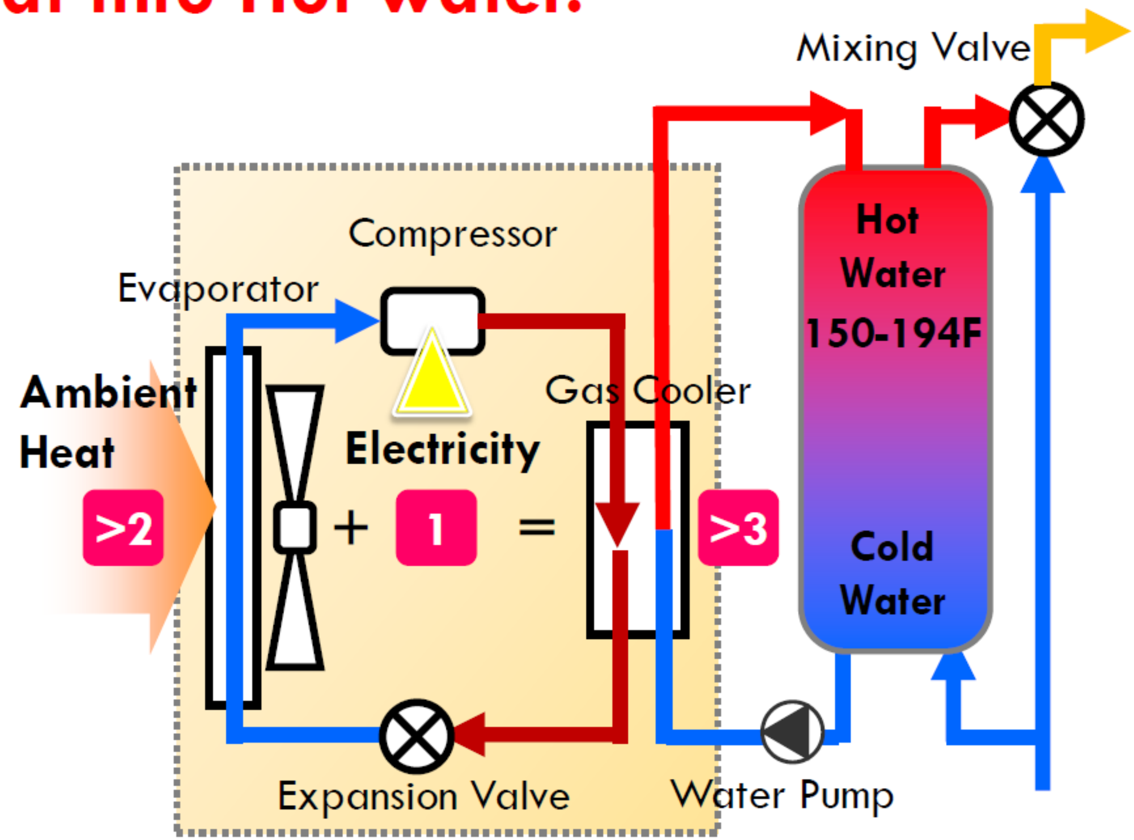
Tank unit

100 to 120gal

4.5 to 6kW

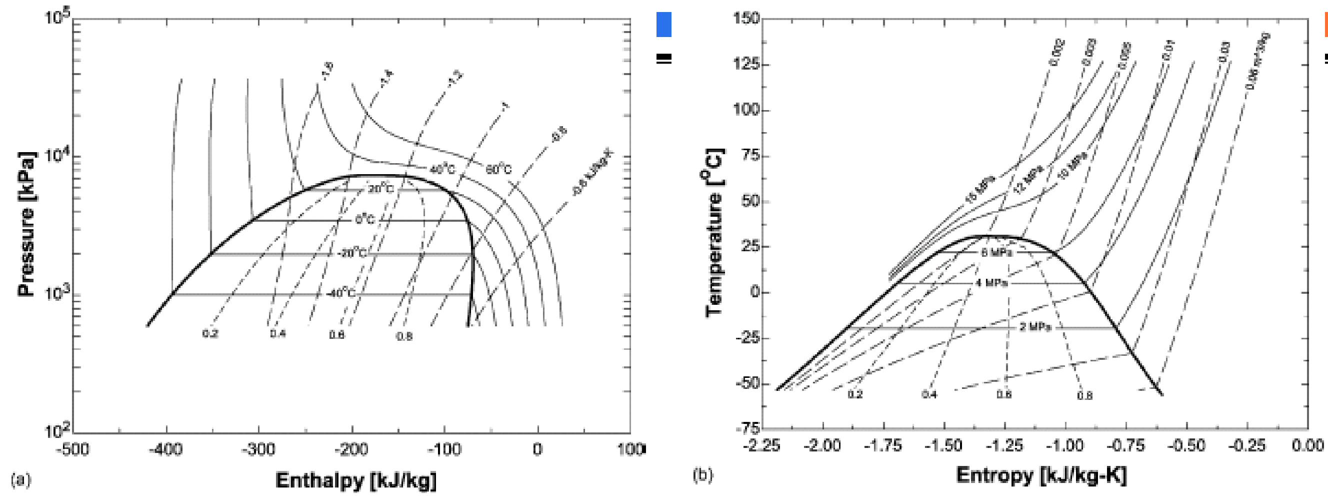


Heat pump unit

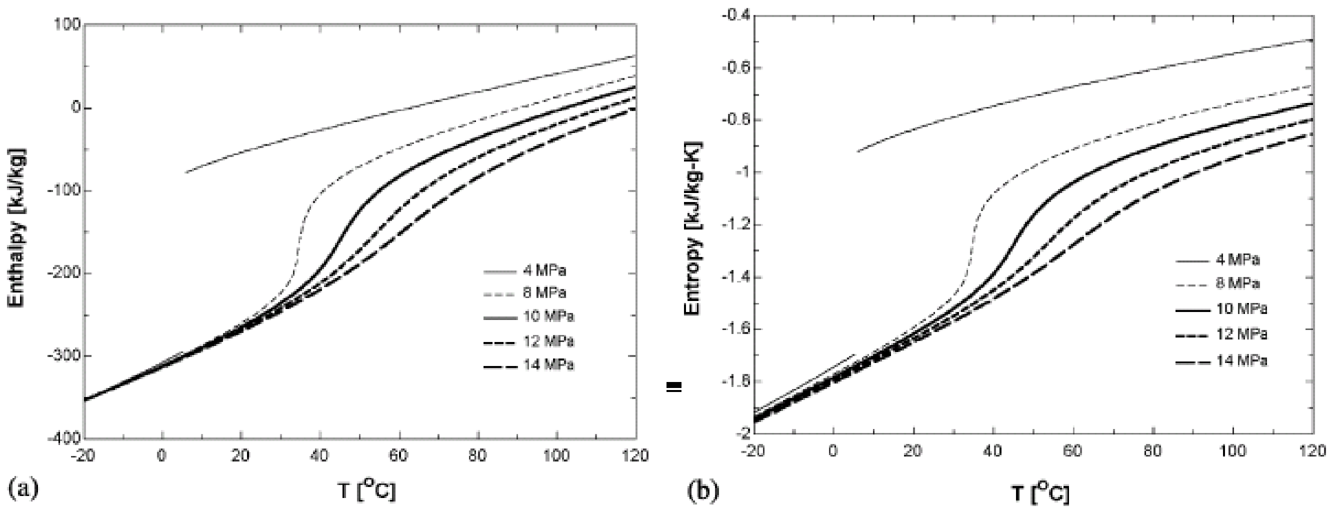


Source of the product picture: model# HE-K37AQS from Panasonic(Matsushita) online catalogue
http://national.jp/sumai/hp/online.html

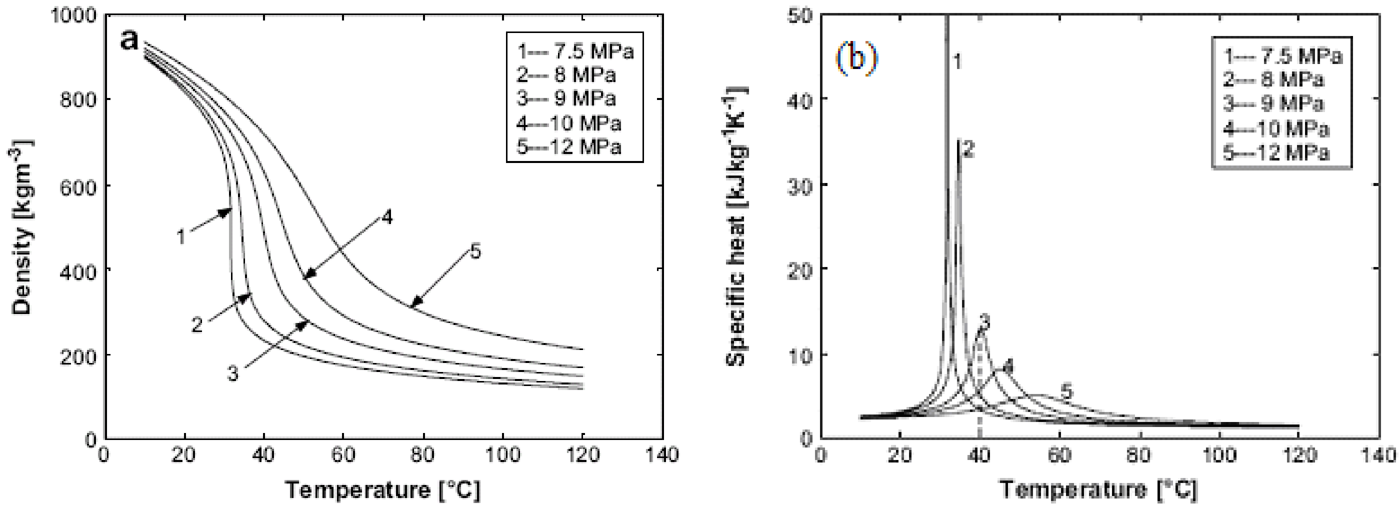
CO_2 熱力性質



CO_2 的(a)壓力/焓與(b)溫度/熵關係圖

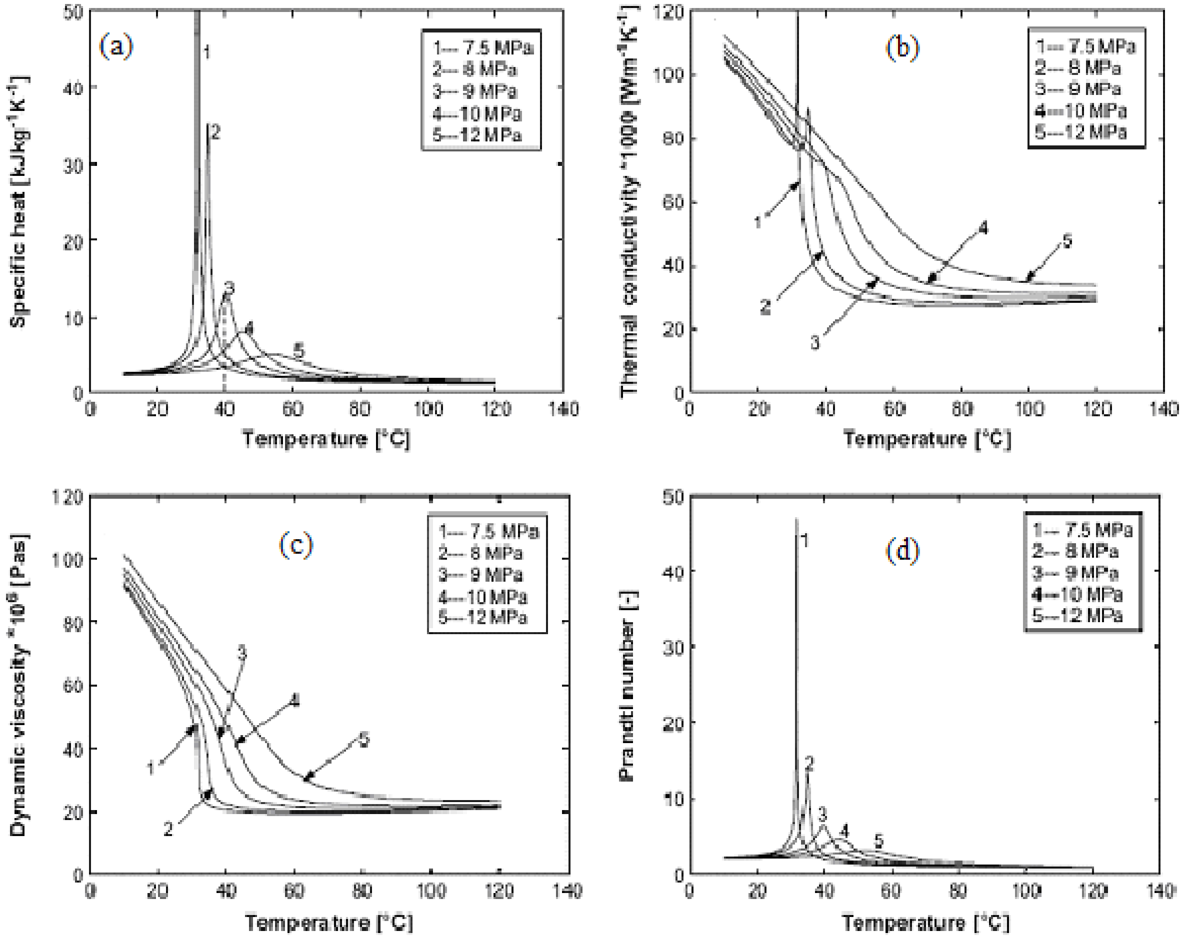


CO_2 熱力性質



CO_2 的密度與定壓比熱在不同壓力下與溫度變化關係圖

CO_2 輸送性質


(a)
1--- 7.5 MPa
2--- 8 MPa
3--- 9 MPa
4---10 MPa
5---12 MPa
Specific heat [kJkg⁻¹K⁻¹]
Temperature [°C]
(b)
Thermal conductivity *1000 [Wm⁻¹K⁻¹]
Temperature [°C]
(c)
Dynamic viscosity *10⁶ [Pas]
Temperature [°C]
(d)
Prandtl number [-]
Temperature [°C]

天然冷媒之性質與理論 COP

Applied Thermal Engineering Vol. 17, No. 1, pp. 33–42, 1997

Table 1. Characteristics of some synthetic and natural refrigerants

Refrigerant	R-12 (CFC)	R-22 (HCFC)	R-134a (HFC)	R-717 (NH_3)	R-744 (CO_2)	R-290 (propane)	R-600 (butane)	R-718 (H_2O)	R-728 (air)
Natural substance	no	no	no	yes	yes	yes	yes	yes	yes
ODP[a]	0.9	0.05	0	0	0	0	0	0	0
GWP[b]	3	0.34	0.29	0	0[d]	< 0.03	< 0.03	0	0
Toxicity TLV (ppm, volume)[c]	1000	500	1000	25	5000	1000	1000	no	no
Flammability	no	no	no	yes	no	yes	yes	no	no
Critical point temperature (°C)	115.5	96.2	100.6	133	31.1	96.8	152.1	374.2	−140
Critical point pressure (bar)	40.1	49.9	40.7	114.2	73.7	42.6	38.0	221.2	37.2
Normal boiling point (°C)	−30	−40.8	−26	−33.3	−78.4	−42.1	−0.4	100	no
Maximum refrigeration capacity at 0°C (kJ/m³)	2733	4344	2864	4360	22 600	3888	1040	1349[e]	—

[a]Ozone depletion potential—compared with R-11.
[b]Global warming potential—compared with R-11.
[c]Threshold limit value for exposure of 8 h/day, 40 h/week, without any adverse effect.
[d]Zero effective GWP, because more than sufficient quantities of it can be recovered from waste gases.
[e]At 100°C.

Table 1 – Comparative refrigerant performance

No.	Name	CoP
R-717	Ammonia	4.84
R-290	Propane	4.74
R-600	Butane	4.68
R-22	Chlorodifluoromethane	4.65
R-134a	Tetrafluoroethane	4.60
R-407C	R-32/R-125/R-134a (23/25/52)	4.51
R-410A	R-32/R-125 (50/50)	4.41
R-404A	R-125/R-143a/R-134a (44/52/4)	4.21
R-744	Carbon dioxide	2.96

Based upon a standard operating cycle of 258 K evaporating temperature, 303 K condensing temperature, 0 K subcooling and 0 K superheat.

INTERNATIONAL JOURNAL OF REFRIGERATION 31 (2008) 545–551

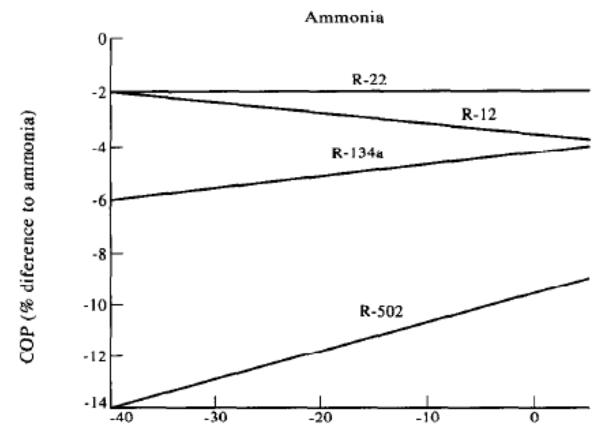


Fig. 2. Relative COP of an isentropic vapour compression cycle for different refrigerants [4].

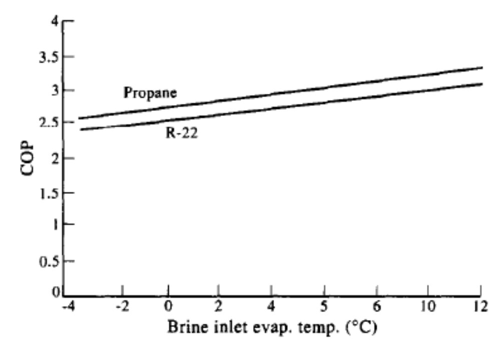


Fig. 3. COP of a heat pump with R-22 and R-290, for different brine inlet evaporator temperatures [12].

CO_2 與 R410A & R-407C 之比較

Table 1. Refrigerant characteristics

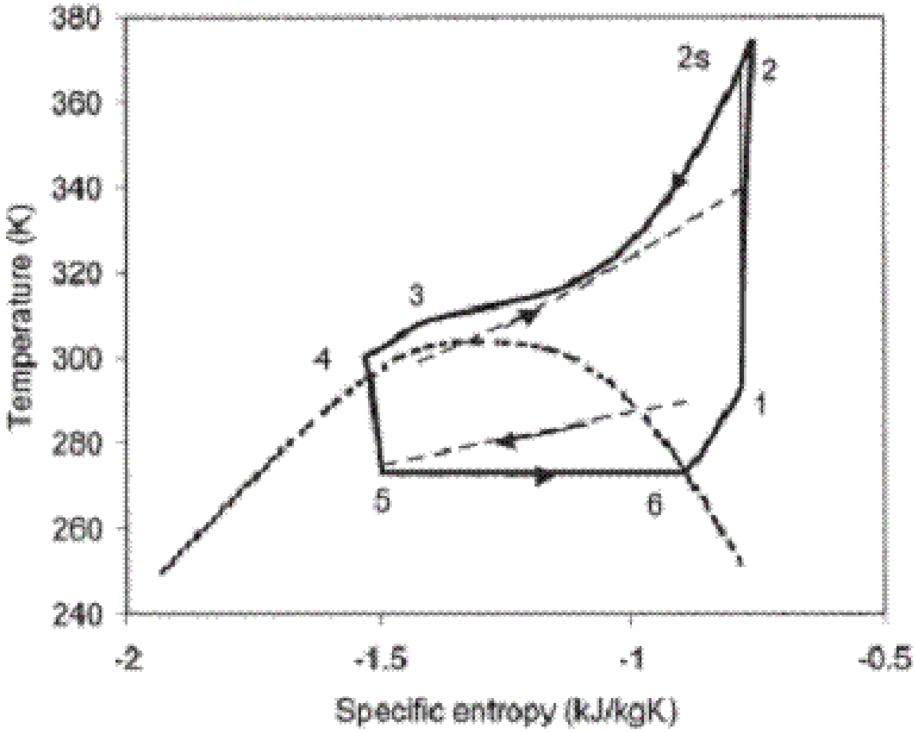
Refrigerant	HFC		Natural refrigerant
	R410A	R407C	R744(CO_2)
Practical examples of commercialization	RAC PAC Commercial Water Heater	PAC Chiller Commercial Water Heater	Residential Water Heater (Eco Cute) & Commercial Water Heater
ODP[1]	0	0	0
GWP[2]	1975	1652.5	1
Combustible	No	No	No
Toxic	Low	Low	Low
Pressure (MPa) (low/ high)	2.7/3.0	1.8/2.0	9.5/11
COP(compared to R410A) (low/high)	100	95/100	60/80

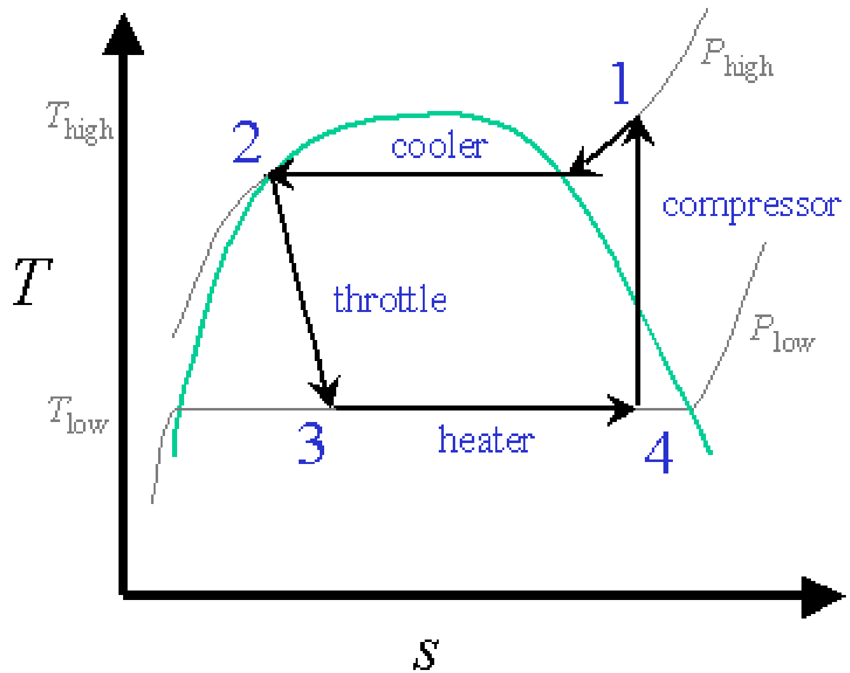
Note: RAC= residential air-conditioning; PAC= packaged air-conditioning.

[1] Ozone depletion potential

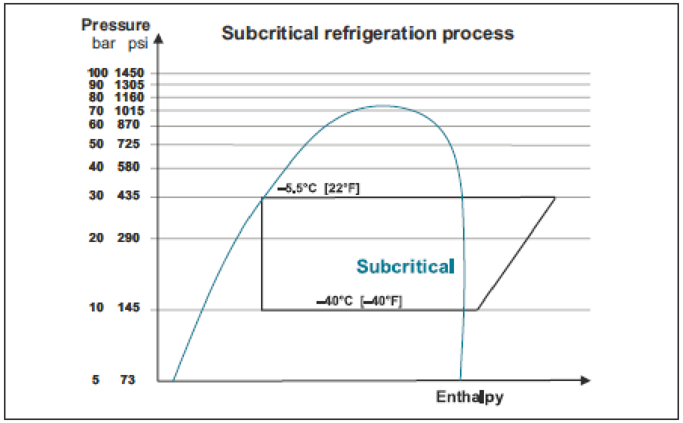
[2] Global warming potentials are based on IPCC 2001.

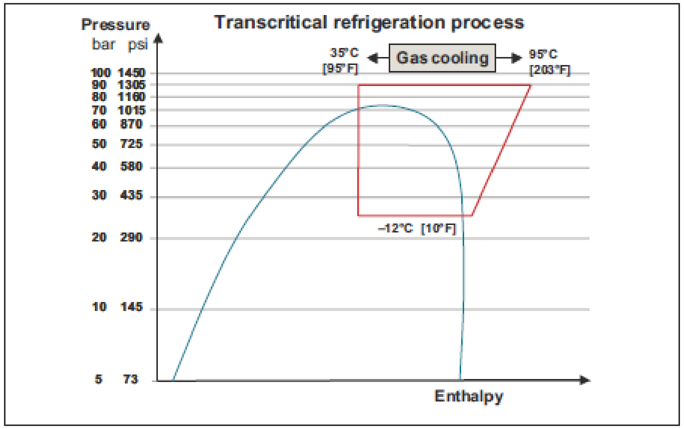
典型 CO_2 (TS diagram) 循環與傳統 R-134a 系統循環

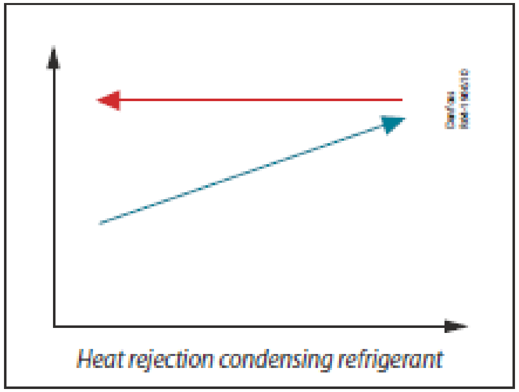


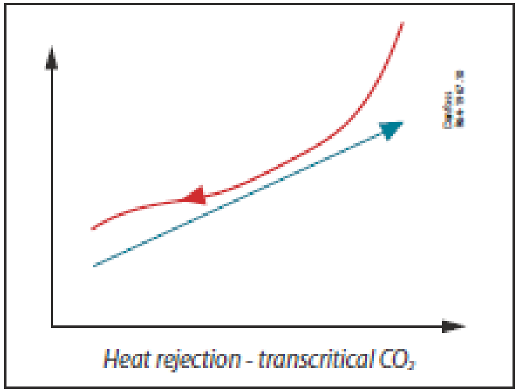


典型 CO_2 (P-h diagram) 循環與傳 統冷凍系統循環









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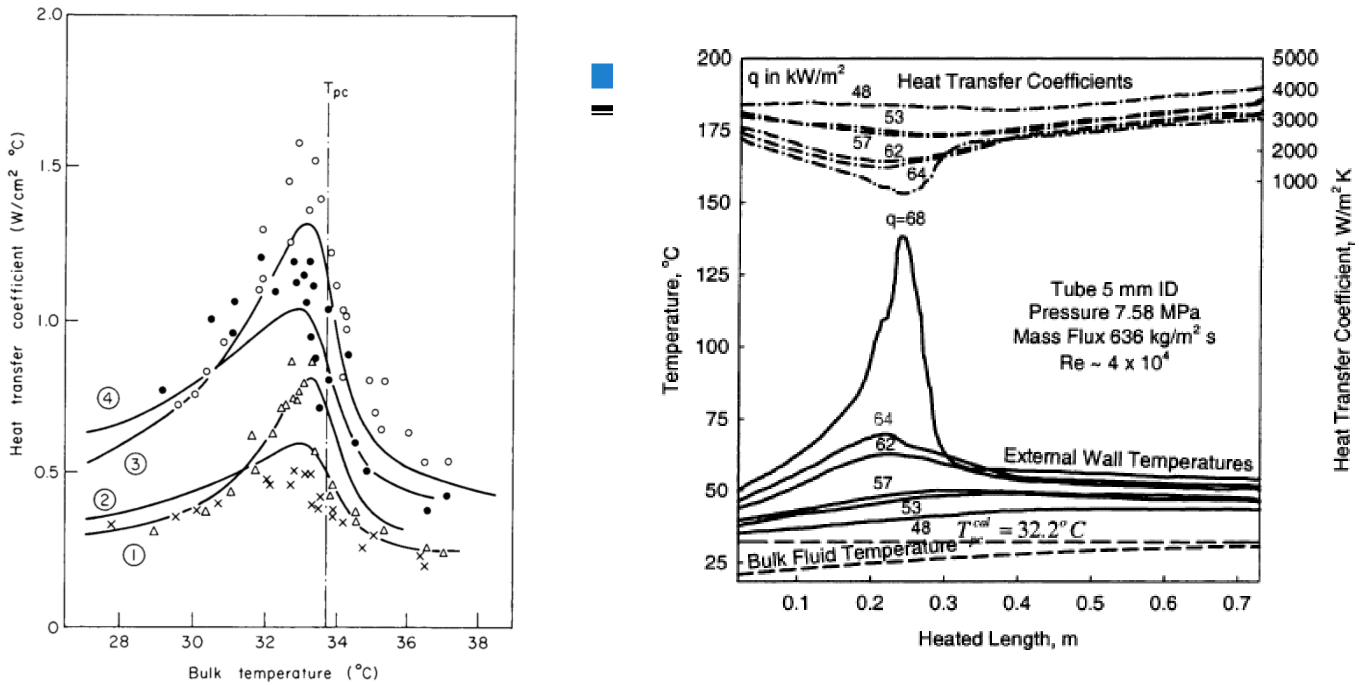
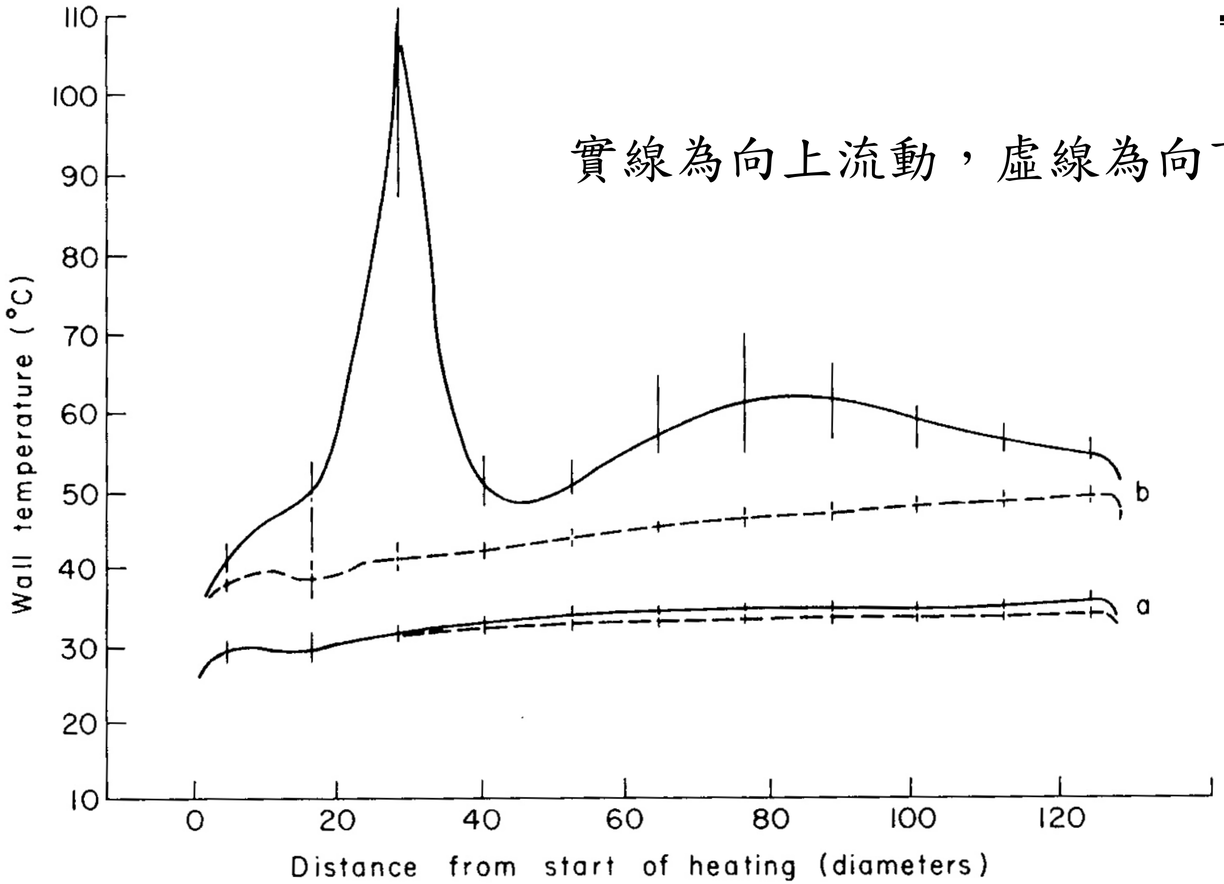
CO_2 熱傳特性



(a) (b)

(a)管徑為 1.0 cm，CO_2 向上流動的熱傳係數與熱通量、流量與中心平均溫度間的關聯， 其中空心圓為 G = 280 kg/hr, q = 3.32 W/cm^2, 實心圓為 G = 280 kg/hr, q = 5.2 W/cm^2, 三角形為 G = 140 kg/hr, q = 1.44 W/cm^2, × 為 G = 140 kg/hr, q = 2.73 W/cm^2；(b) 管徑為 5.0 mm，CO_2 向上流動的管壁溫度與熱傳係數在不同熱通量下各個位置的變化圖

CO_2 熱傳特性

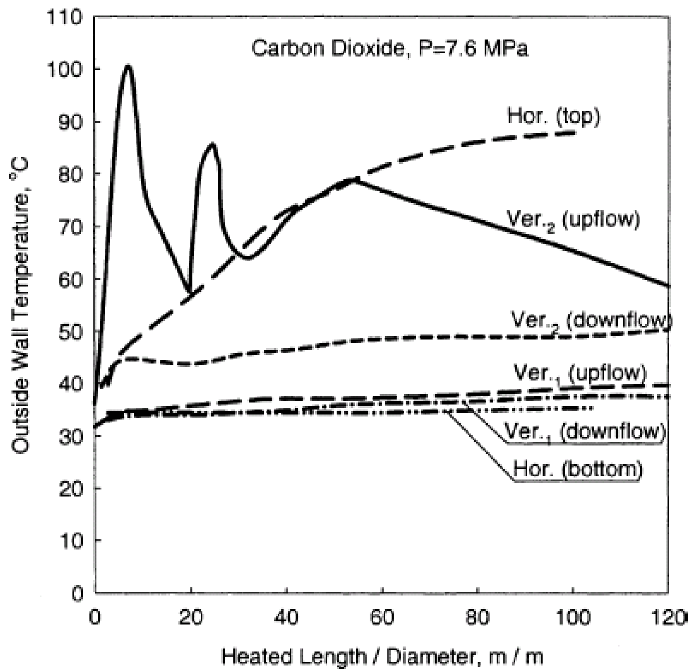


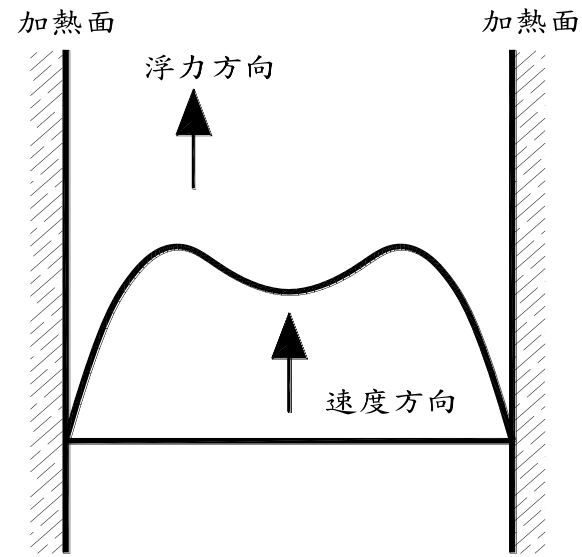
實線為向上流動，虛線為向下流動

CO_2 向上與向下流動時管壁溫度在不同熱通量隨位置的變化圖；(a) q = 3.09 W/cm^2 (b) q = 5.67 W/cm^2

CO_2 熱傳特性

Curve	Data	D, mm	T_{in}, °C	m, kg/s	q_w, kW/m^2
Horizontal	Adebiyi, Hall, 1976	22	20	0.121	27
$Vertical_2$	Weinberg, 1972	19	20	0.124	50
$Vertical_1$	Weinberg, 1972	19	20	0.124	30





CO_2 向上流動時 M 型溫度分佈示意圖

CO_2 向上、向下與垂直流動時管壁溫度在不同位置的變化圖

潤滑油對CO_2熱傳特性之影響

Mineral oil: Paraffins, naphthenics, aromatics and non-hydrocarbons – normally for CFC, HCFC, and hydrocarbon

Synthetic oil: POE, PAG, PAO, AB – normally for HFC

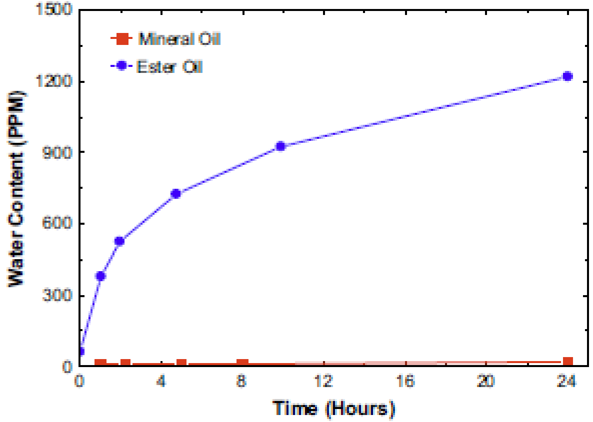


Fig. 2 – Capacity of moisture absorption of ester and mineral oils as a function of the time in open beaker. Embraco (1996).

潤滑油對CO_2熱傳特性之影響

Oil has higher surface tension than refrigerant – causing better wetting (R-12)

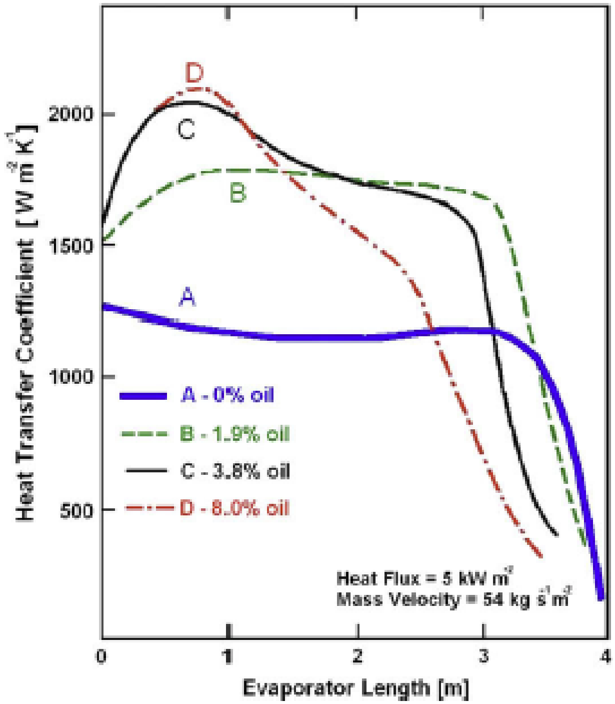


Fig. 3 – Local heat transfer coefficient in function of the evaporator length for complete evaporation at −15 °C, Worsøe-Schmidt (1960).

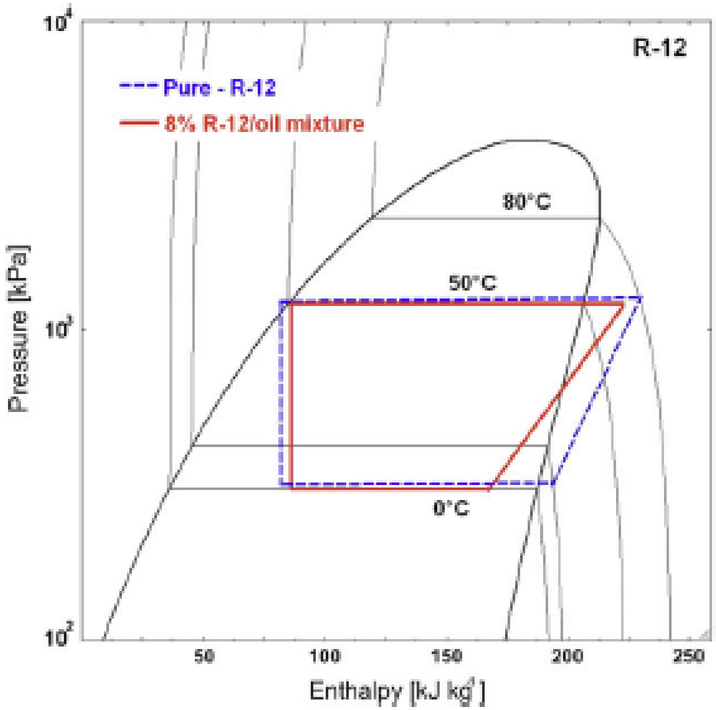


Fig. 4 – Pressure–enthalpy chart showing the influence of lubricant oil on the refrigeration cycle. Hughes et al. (1982).

潤滑油對R-134a 熱傳特性之影響

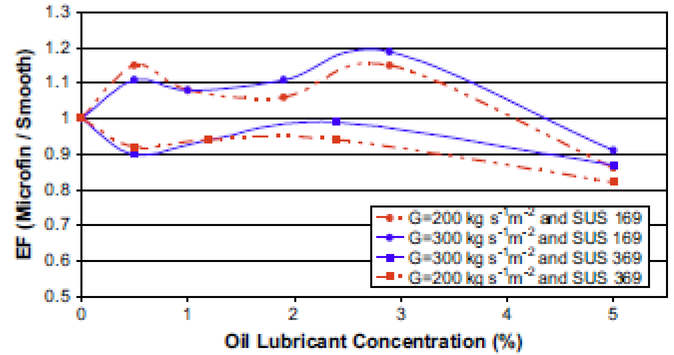


Fig. 6 – Heat transfer enhancement factor (EF) versus lubricant oil concentration, varying mass velocity and type of the oil Data from Eckels et al. (1994).

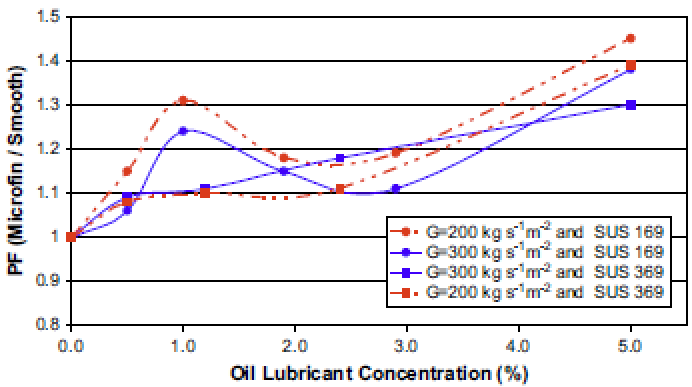
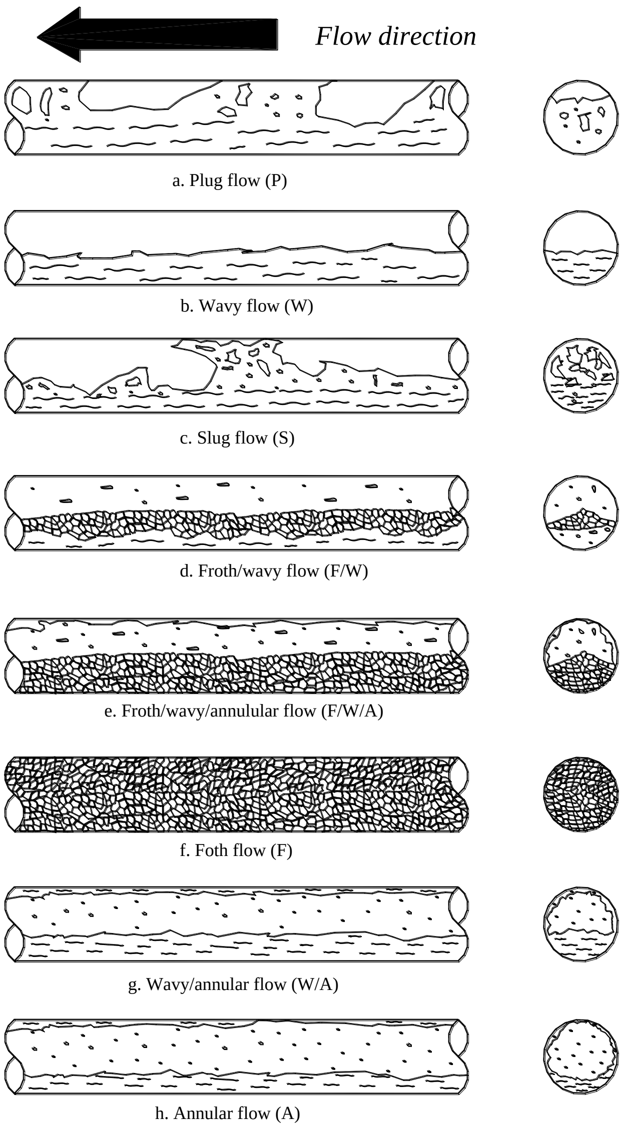
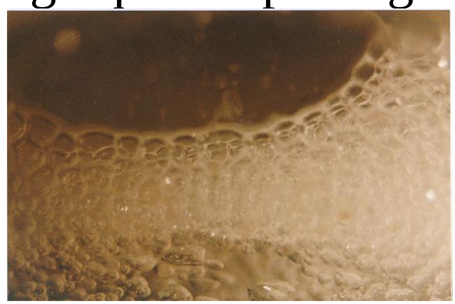


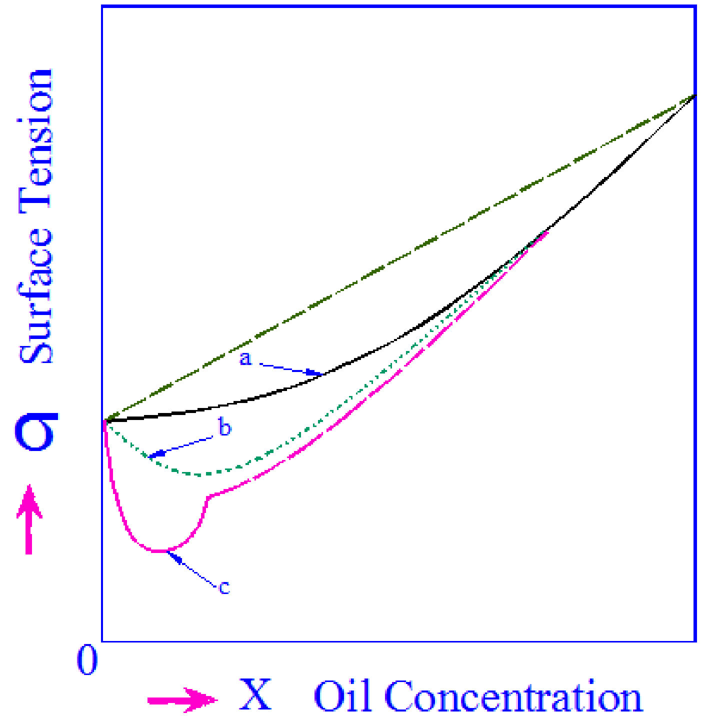
Fig. 7 – Pressure drop penalty factor (PF) versus lubricant oil concentration, varying mass velocity and type of the oil. Data from Eckels et al. (1994).



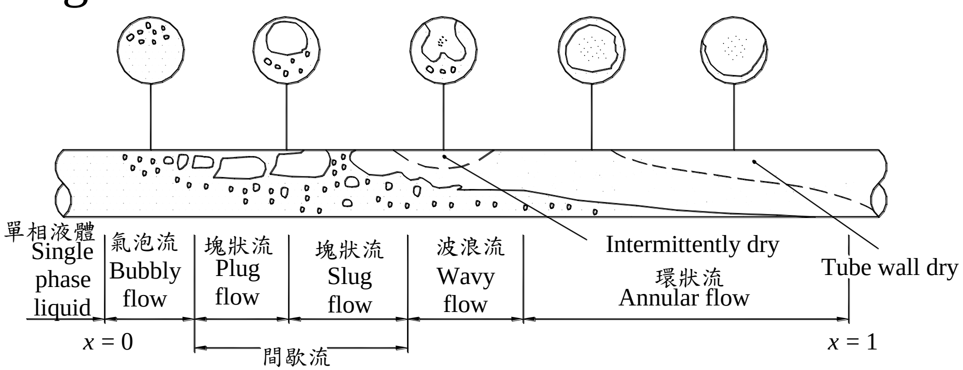
潤滑油增強熱傳性能的可能原因

possible enhancement effects caused by the foaming: (1) a thin liquid film was created between the foam and the heated surface which results in a very large heat transfer coefficient; (2) secondary nucleation caused by the bubble leaving the surface which bursting into the neighboring liquid-vapor region.





Early transition to annular flow pattern for refrigerant/oil mixture



$$W = \frac{A\sigma^3}{\left[B - C(y - x)\right]^2}$$

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International Journal of Heat and Mass Transfer 52 (2009) 870–879

潤滑油對CO_2熱傳特性之影響

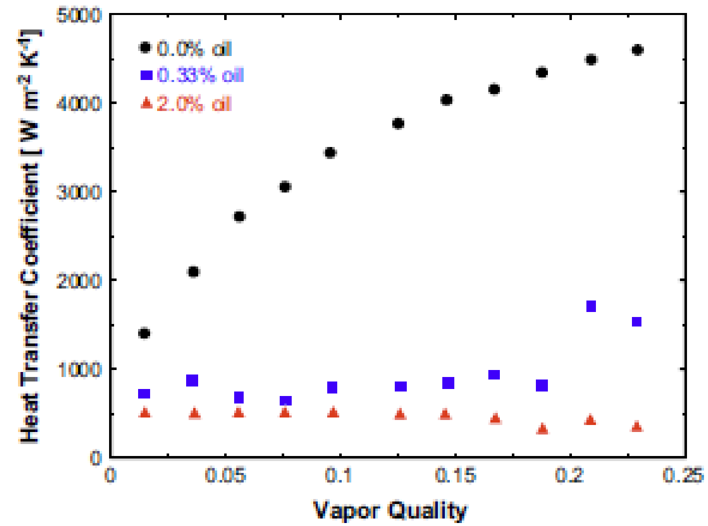


Fig. 5 – The lubricant oil effect on heat transfer coefficient in flow boiling, according to the data of Chaddock and Buzzard (1986) for the test conditions: G = 65 kg s^{-1} m^{-2}, q = 6.3 kW m^{-2} and $T_{...}$ = 9.5 °C at several oil

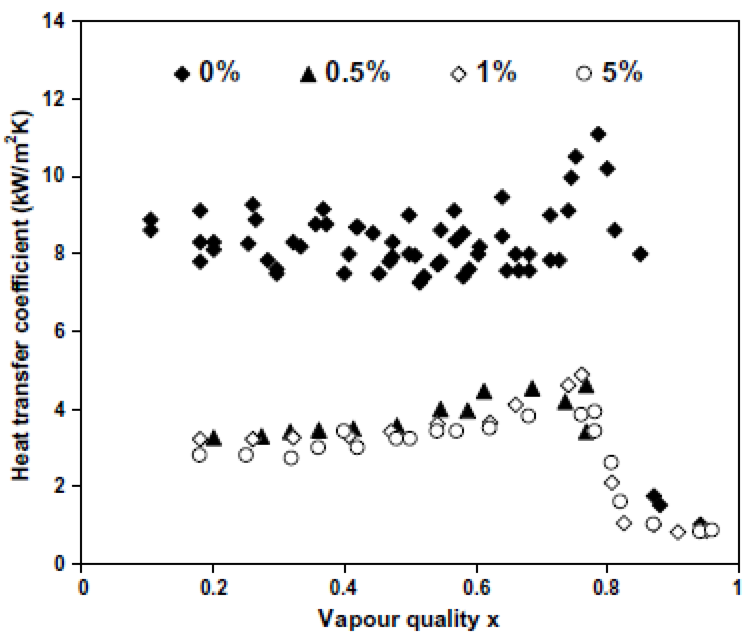


Fig. 1. Effect of oil concentration on h_{tp} of CO_2–PAG mixture (Dang et al. [32]) at G = 360 kg/m^2 s, T_{sat} = 10 °C and q = 9 kW/m^2.

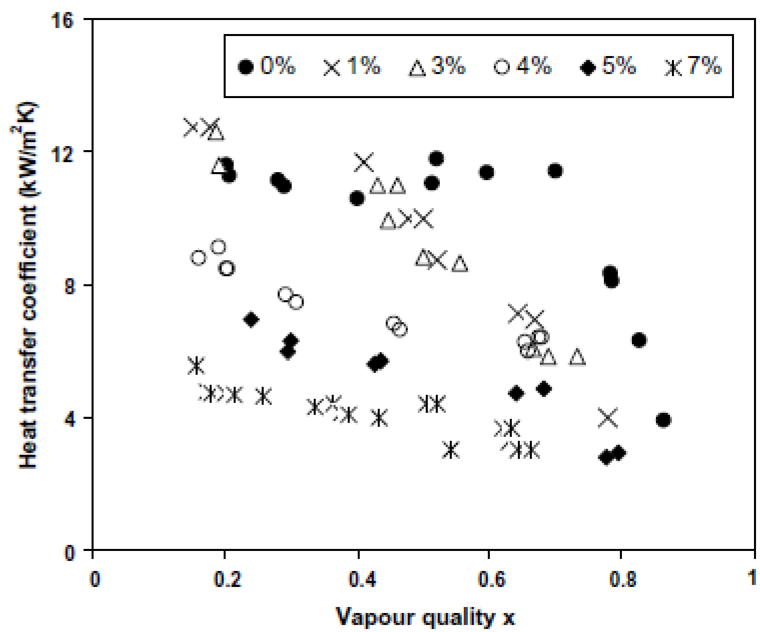


Fig. 3. Effect of oil concentration on h_{tp} (Zhao et al. [38]) at G = 300 kg/m^2 s, T_{sat} = 10 °C and q = 11 kW/m^2.

潤滑油對CO_2熱傳特性之影響 - 平滑管

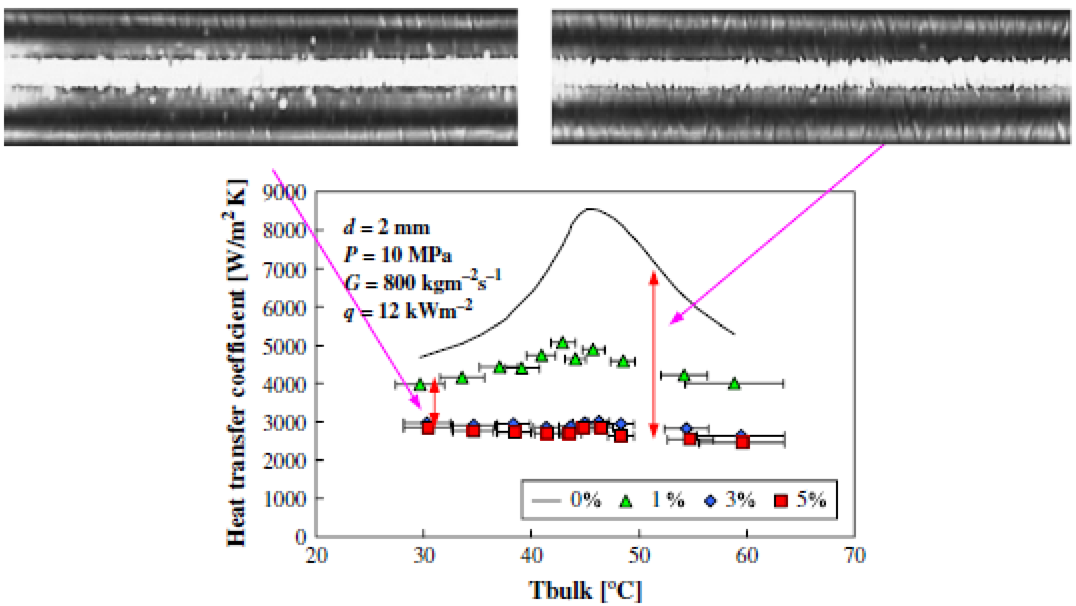


Fig. 9 – Flow pattern of CO_2–oil and heat transfer coefficient for 2 mm ID tube.

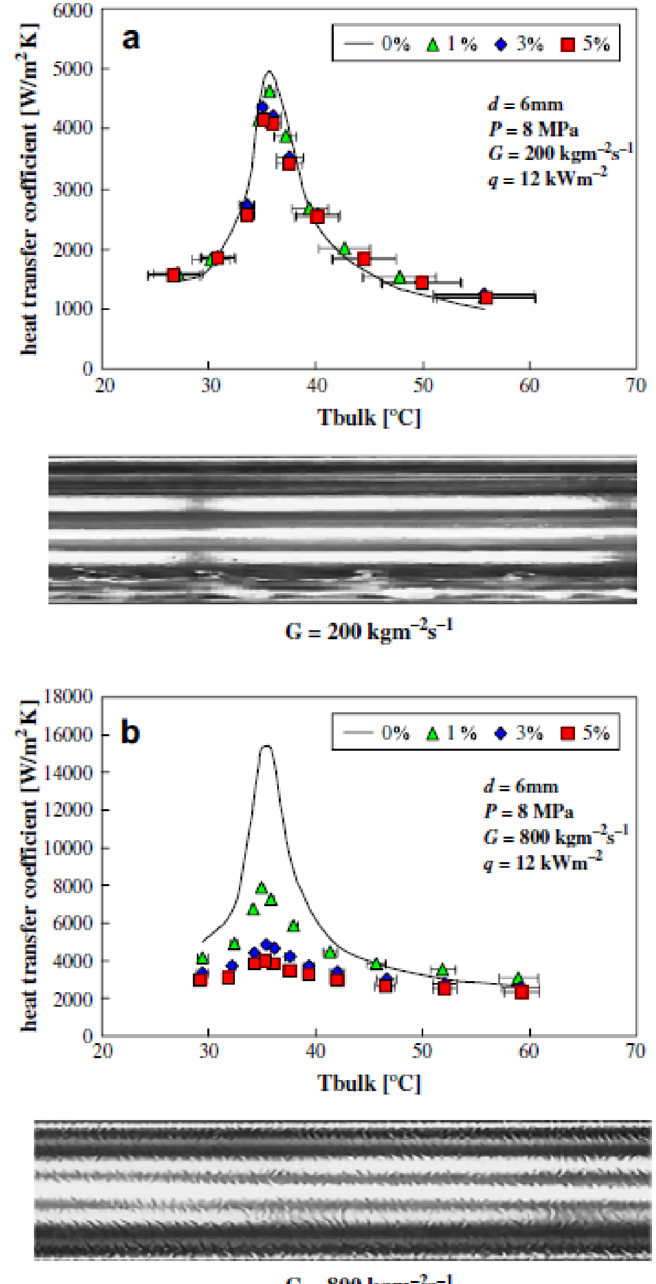


Fig. 10 – Flow pattern of CO_2–oil (40 °C) and heat transfer coefficient for 6 mm ID tube.

PAG type oil

Oil layer effect mainly @ 1~3%
Higher mass flux cause considerably degradation (droplet vs. film)

INTERNATIONAL JOURNAL OF REFRIGERATION 31 (2008) 1265–1272

潤滑油對CO_2熱傳特性之影響 - 微鰭管

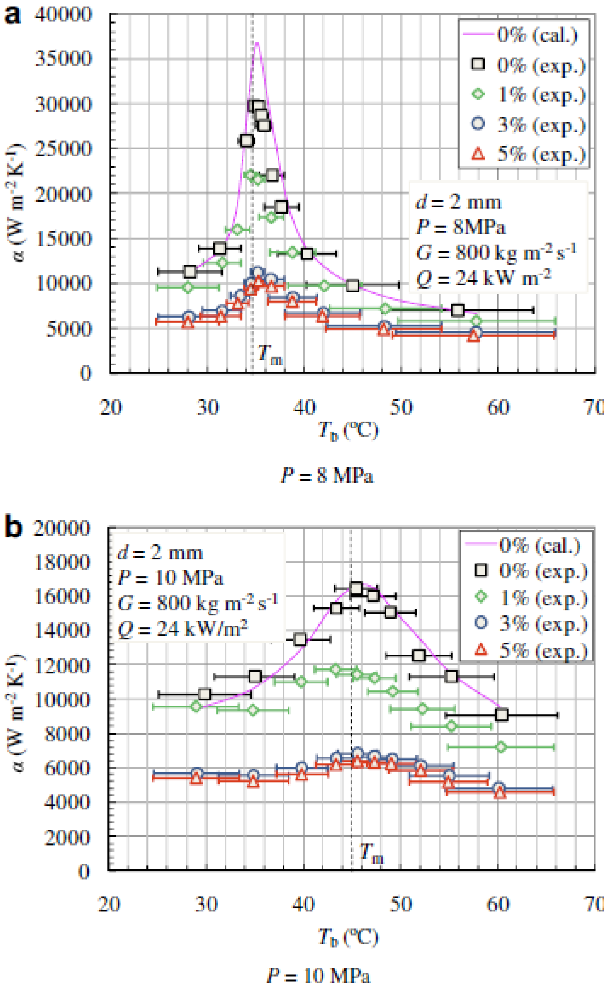


Fig. 7 – Change in α with T_b at different x values.

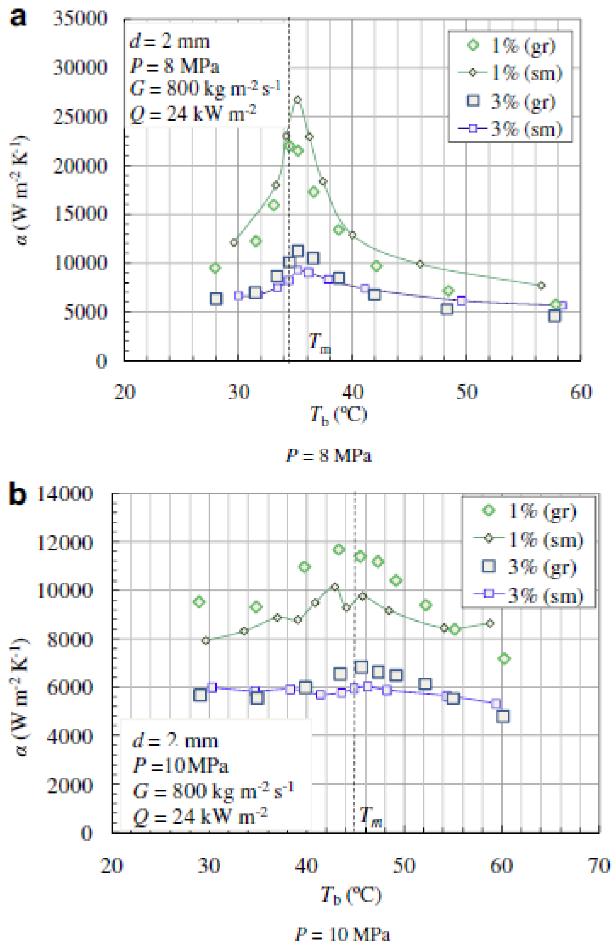


Fig. 8 – Comparison between the α values obtained for the grooved tube and smooth tube at $G = 800\ kg\ m^{-2}\ s^{-1}$.

Fig. 2 – Enlarged view of cross-sectional shape of grooved tube.

Table 1 – Specifications of internally GROOVED tube.

O.D. [mm]	2.646	I.D. [mm]	1.996
Wall thickness [mm]	0.269	Fin height [mm]	0.117
Apex angle [°]	34.8	Helix angle [°]	6.3
Number of fins	40	Area enlargement ratio	2.0

Less effect of oil for microfin helps to break up the oil film

CO_2 流動流譜 (潤滑油)

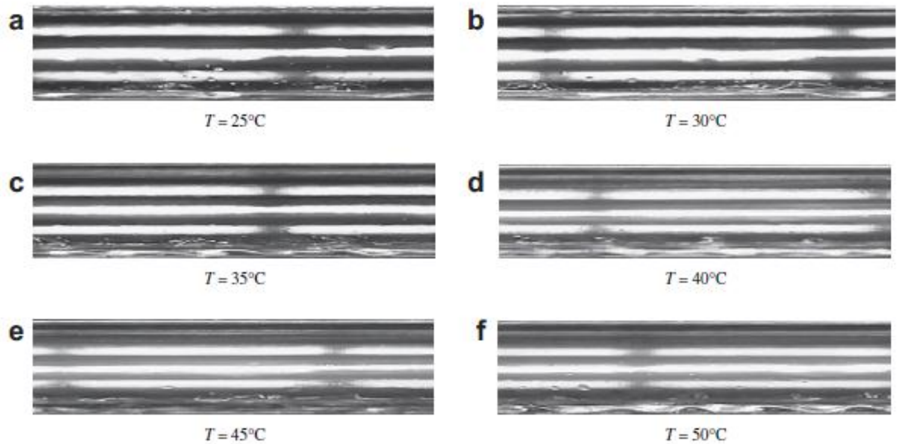


Fig. 6 – Comparison of flow pattern at different temperature for 6 mm ID tube; $d = 6$ mm, $P = 8$ MPa, $G = 200$ kg m^{-2} s^{-1}, $x = 1\%$.

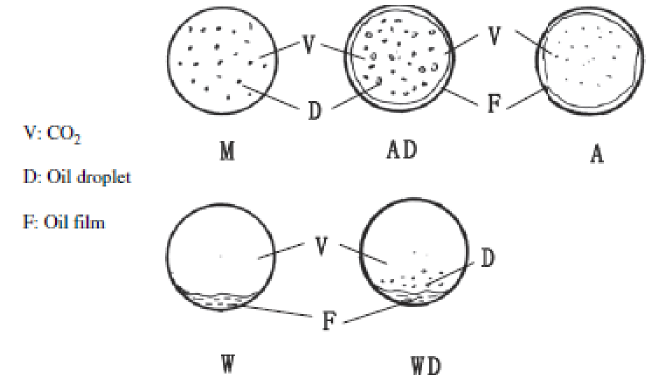


Fig. 4 – Classification of flow pattern. M: mist flow; AD: annular-dispersed flow; A: annular flow; W: wavy flow; WD: wavy-dispersed flow.

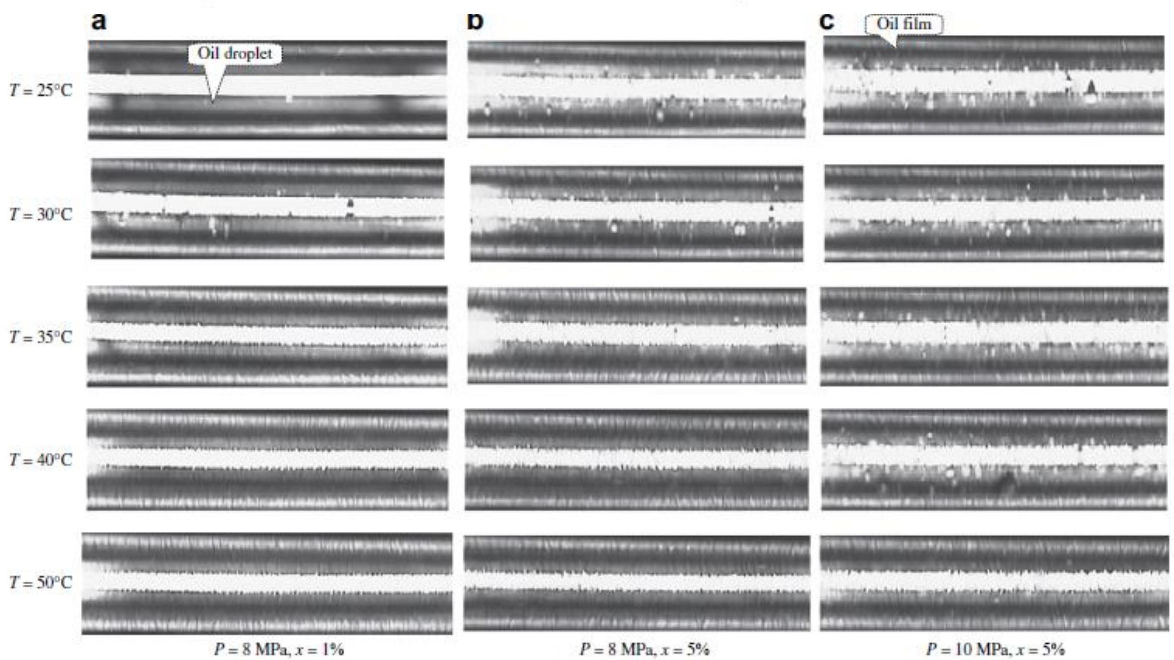
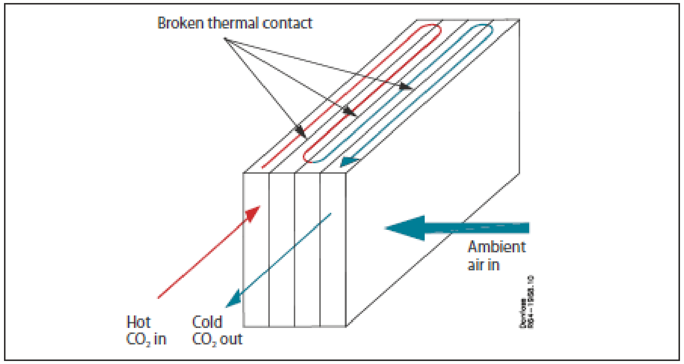


Fig. 5 – Flow visualization of CO_2 with entrained PAG oil; $d = 2$ mm, $G = 800$ kg m^{-2} s^{-1}.

系統設計應用

氣冷式系統設計

通常Gas –cooler 的壓降
不像傳統冷媒的影響之大



Food Retail CO_2 Refrigeration Systems

Designing subcritical and transcritical CO_2 systems
and selecting suitable Danfoss components

REFRIGERATION &
AIR CONDITIONING DIVISION

Application Handbook

CO_2 & R-134a 系統性能之比較

Owing to the higher average temperature of heat rejection, and the larger throttling loss, the theoretical cycle work for CO_2 increases compared to a conventional refrigerant as R-134a as indicated.

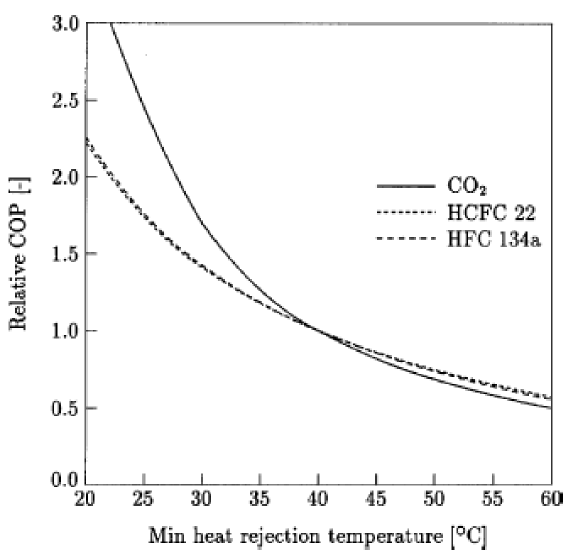


Fig. 23. Relative change in cooling COP for R-22, R-134a and CO_2 at varying refrigerant exit temperature from condenser/gas cooler (i.e. minimum heat rejection temperature). Evaporating temperature 0 °C. Reference point: 40 °C exit temperature. Based on ideal cycle calculations without subcooling or superheating.

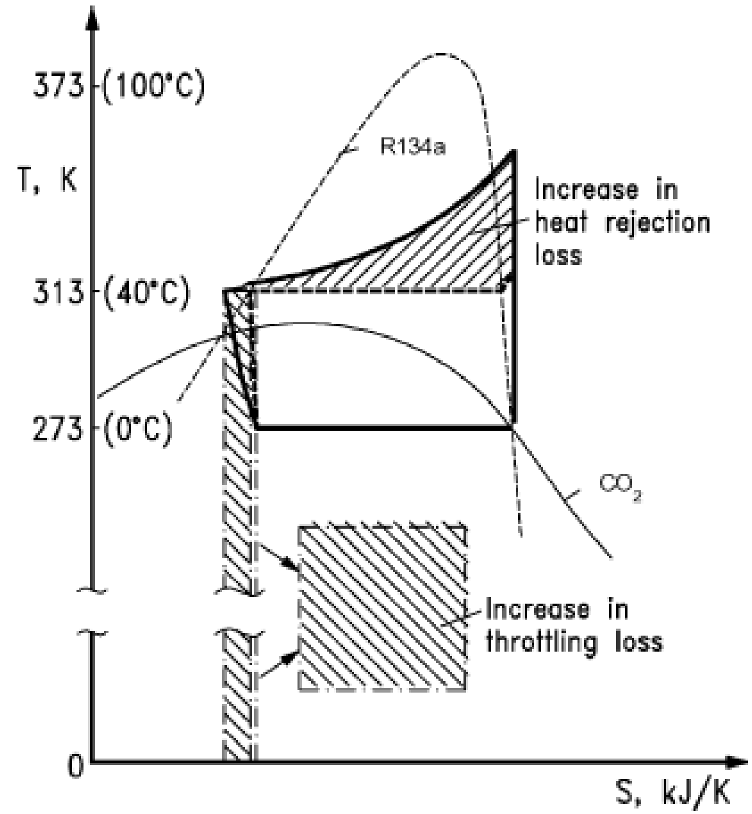


Fig. 21. Comparison of thermodynamic cycles for R-134a and CO in temperature–entropy diagrams, showing additional thermodyn-amic losses for the CO_2 cycle when assuming equal evaporating temperature and equal minimum heat rejection temperature.

Prog. in Energy & Combustion Sci, 2004, 30:119-174.

CO_2 & R-134a/R-22 系統性能之比較

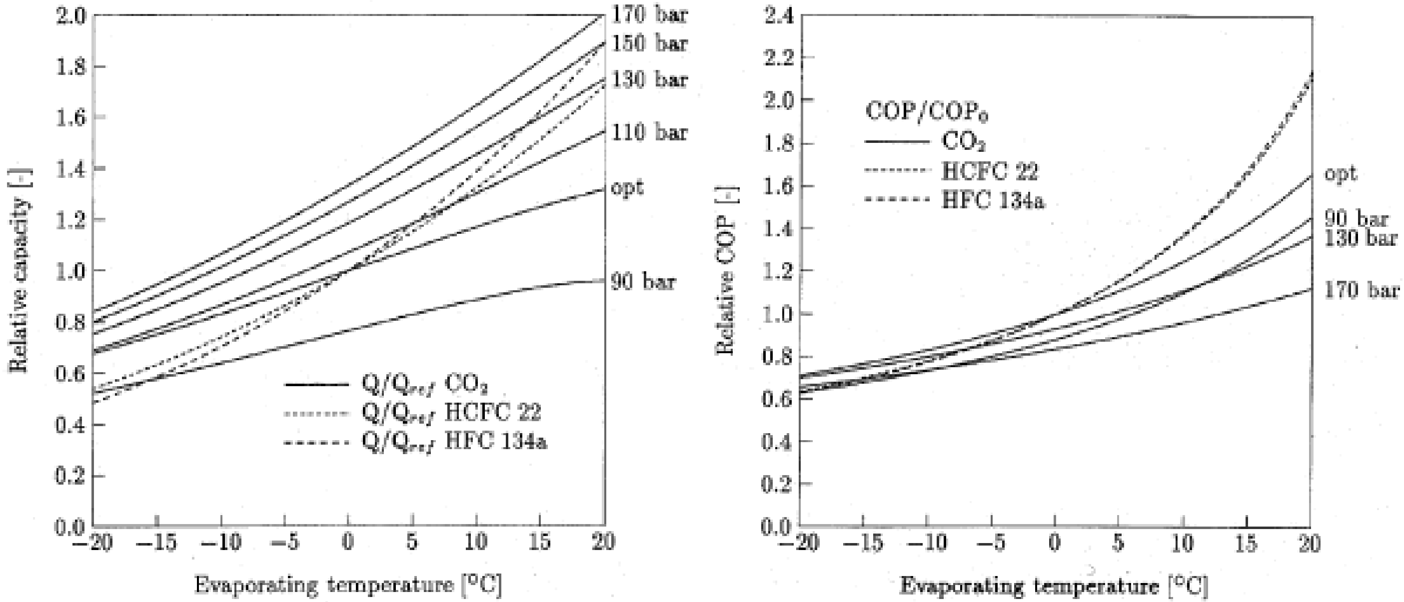


Fig. 22. Relative change in heating capacity (left) and heating COP (right) for R-22, R-134a and CO_2 at varying evaporating temperature, for a condenser/gas cooler exit temperature of 40 °C. Reference point: 0 °C evaporating temperature. Results for CO_2 are shown at COP-optimum high-side pressure, and with relative data for other high-side pressures. Based on ideal cycle calculations without subcooling or superheating.

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CO_2 系統性能與高壓側之關係

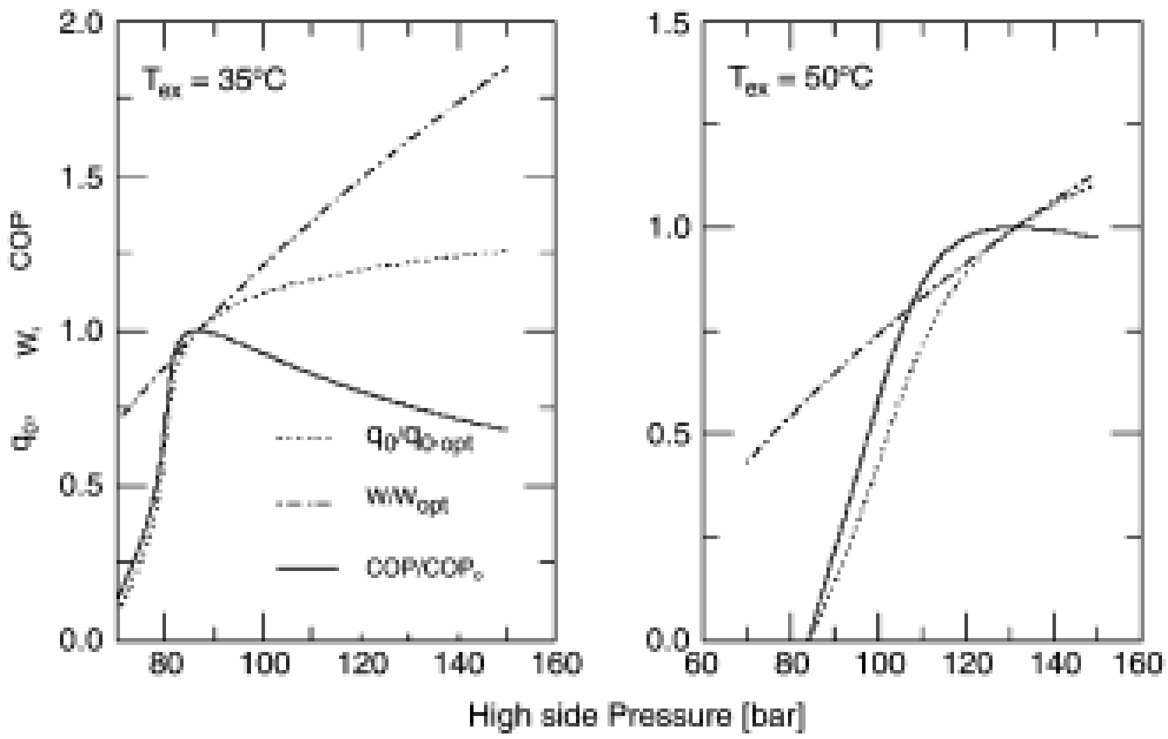


Fig. 17. Influence of varying high-side pressure on specific refrigerating capacity (q_0), specific compressor work (w) and COP in a transcritical CO_2 cycle. The results are based on isentropic compression, evaporating temperature ($T_0 = 5$ °C) and a refrigerant outlet temperature (T_{ex}) from the gas cooler of 35 °C (left) and 50 °C (right).

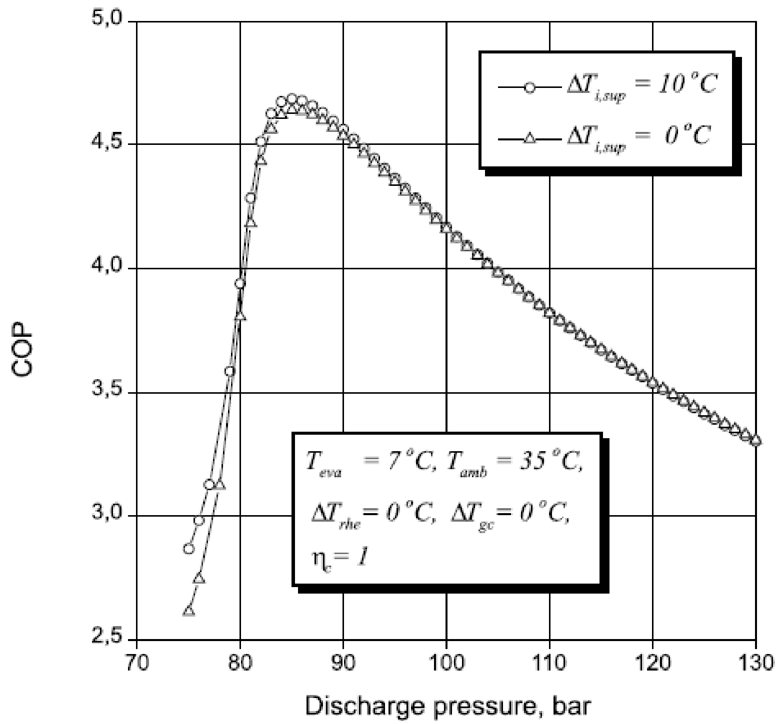
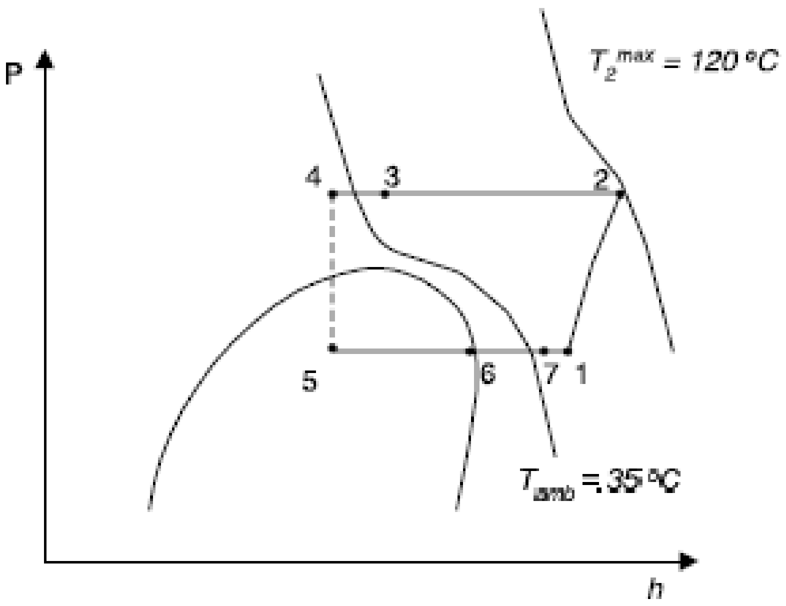


Fig. 2. Effect of internal superheating on the system COP.



提昇 CO_2 系統效率之一些做法

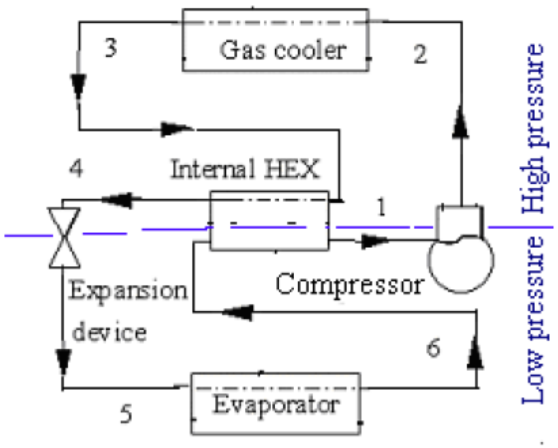


Fig. 2. Layout of transcritical CO_2 system with internal heat exchanger

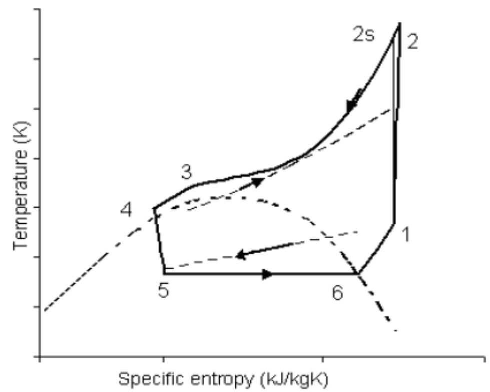


Fig. 3. T-s diagram of transcritical CO_2 system with internal heat exchanger

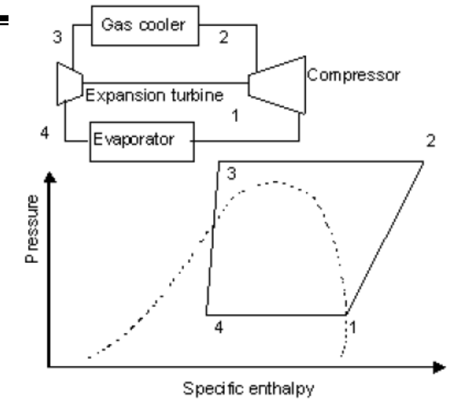


Fig. 4. Layout and p-h diagram of transcritical CO_2 system with expansion turbine

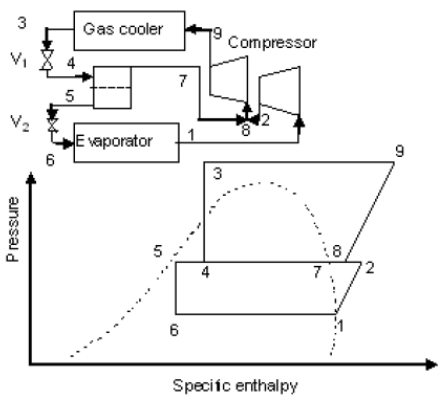


Fig. 5. Layout and p-h diagram of two-stage CO_2 system with flash gas bypass

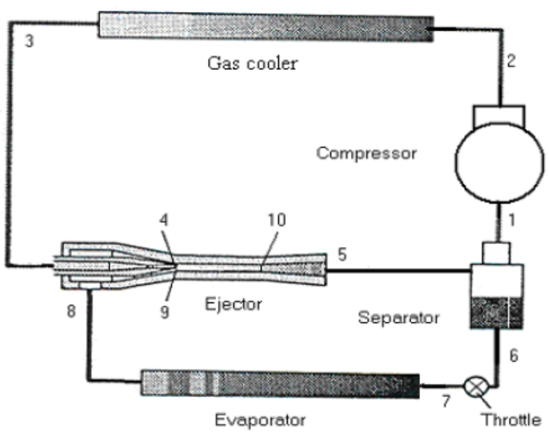


Fig. 6. Schematic diagram of ejector-expansion transcritical refrigeration cycle

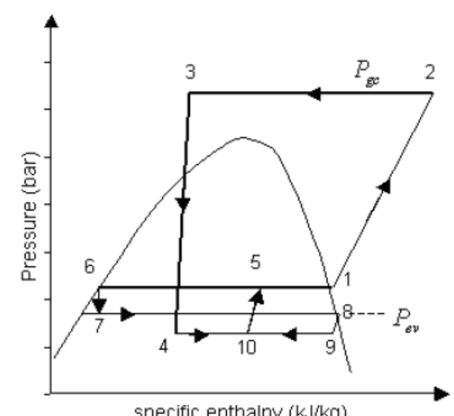


Fig. 7. p-h diagram of ejector-expansion refrigeration cycle

提昇 CO_2 系統效率之一些做法

HVAC&R, 2007, pp. 499-519

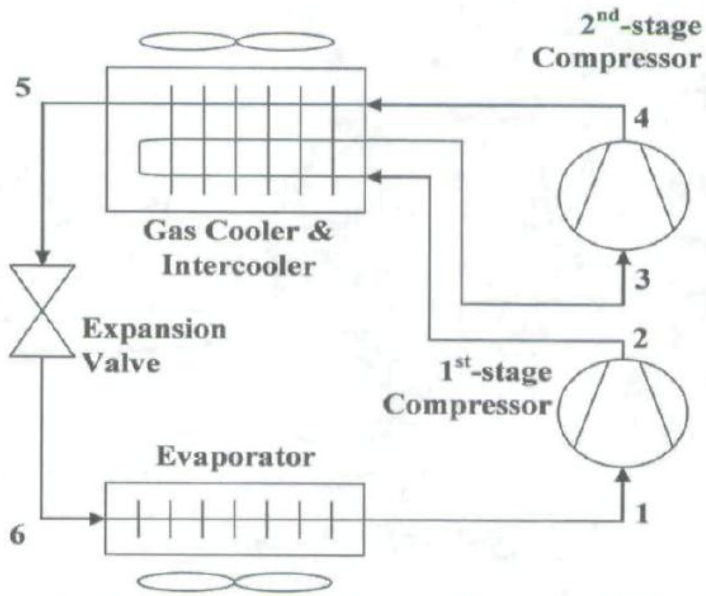


Figure 3. Schematic of two-stage transcritical CO_2 cycle with intercoolin

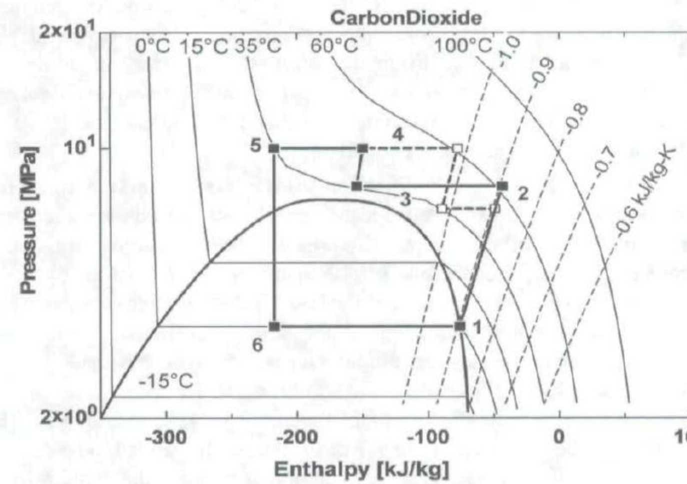


Figure 4. Two-stage transcritical CO_2 cycle with intercooling in a p-h diagram (properties from EES [Klein 2004]).

Table 1 – Comparative refrigerant performance

No.	Name	CoP
R-717	Ammonia	4.84
R-290	Propane	4.74
R-600	Butane	4.68
R-22	Chlorodifluoromethane	4.65
R-134a	Tetrafluoroethane	4.60
R-407C	R-32/R-125/R-134a (23/25/52)	4.51
R-410A	R-32/R-125 (50/50)	4.41
R-404A	R-125/R-143a/R-134a (44/52/4)	4.21
R-744	Carbon dioxide	2.96

Based upon a standard operating cycle of 258 K evaporating temperature, 303 K condensing temperature, 0 K subcooling and 0 K superheat.

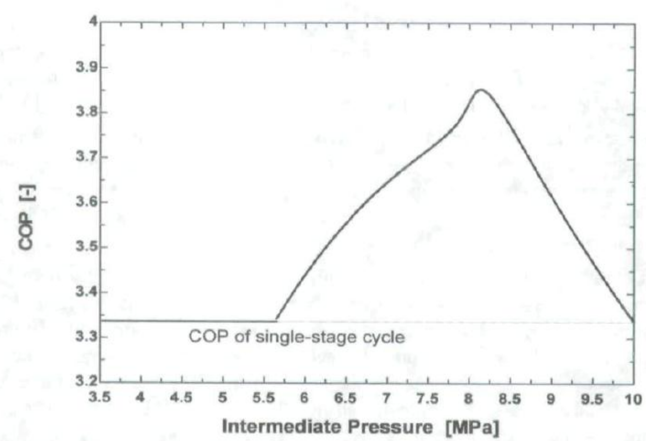


Figure 5. COP of the two-stage transcritical CO_2 cycle with intercooling versus intermediate pressure (properties from EES [Klein 2004]).

提昇 CO_2 系統效率之一些做法

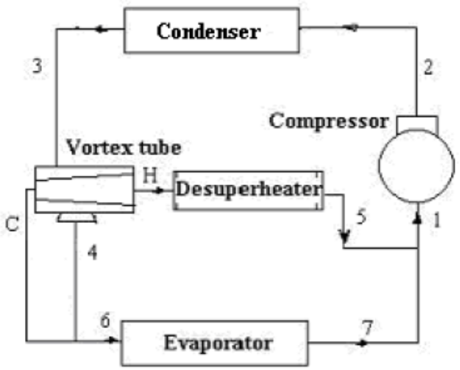


Fig. 8. Schematic diagram of vortex tube-expansion CO_2 refrigeration cycle

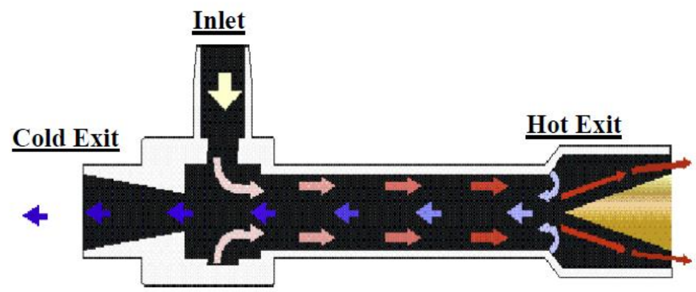


FIGURE 1: Ranque-Hilsch Tube

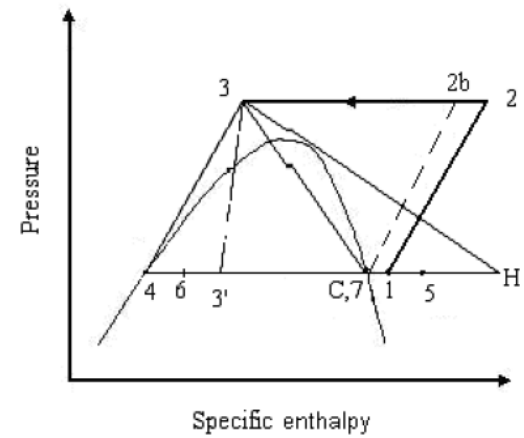


Fig. 9. p-h diagram of ejector-expansion refrigeration cycle

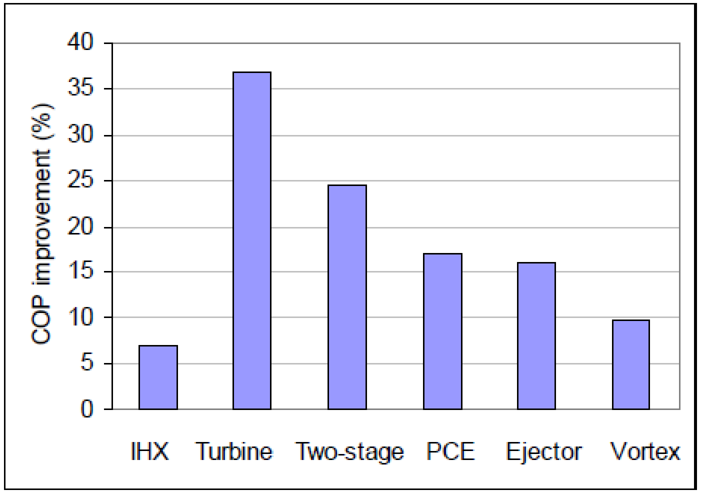


Fig. 11. Comparison in term of COP improvement

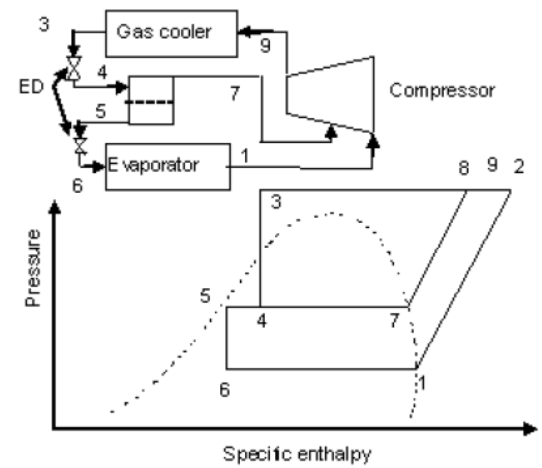
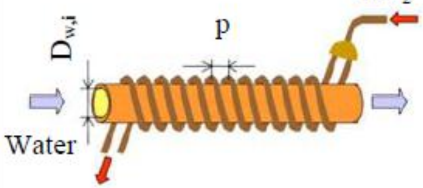
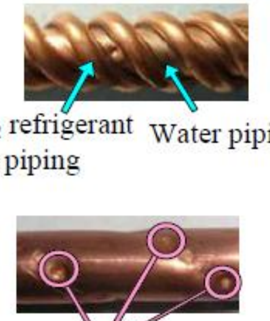


Fig. 10. Layout and p-h diagram of refrigeration cycle with parallel compression economization

Gas Cooler 熱交換器性能提升

Table 2. Comparison of water heat exchanger

	Former type	New type	
Spec	Double tube	Smooth tube (initial time)	Dimple tube (latest)
Outline of shape	 Water, Leckage detection ditch, CO_2	 CO_2, p, $D_{w,i}$, Water	 CO_2 refrigerant piping, Water piping, Dimple
Capacity ratio	1.00	0.89	0.89
Weight ratio	1.00	0.74	0.64
Performance	1.00	1.00	1.00

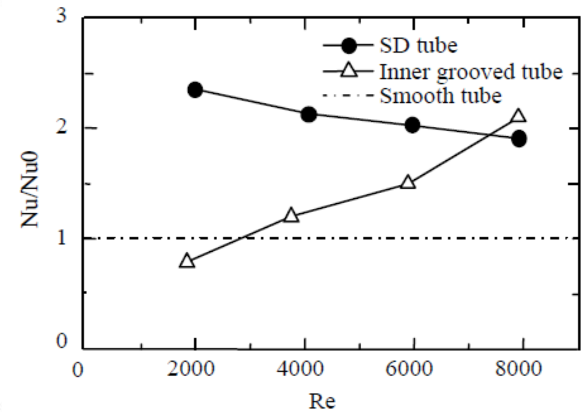


Figure 5. Evaluation of the water heat exchanger

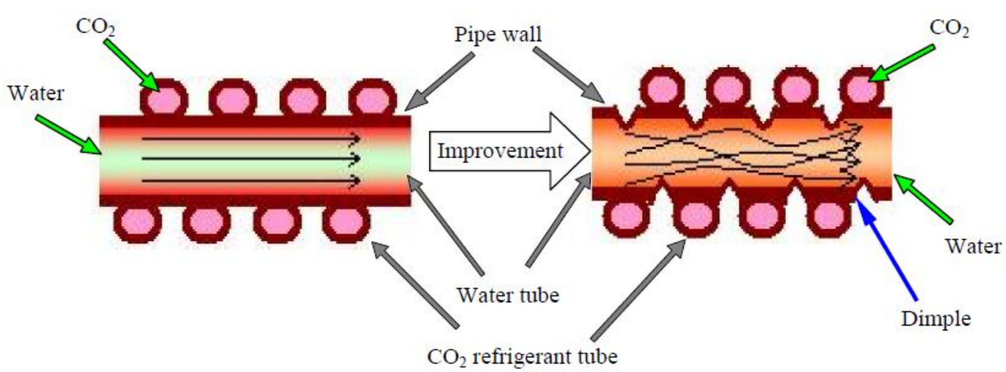
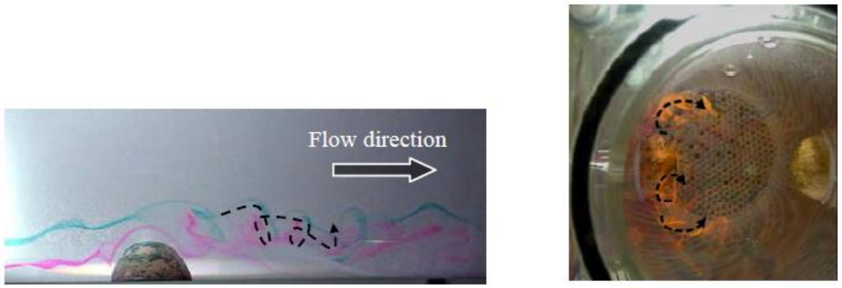


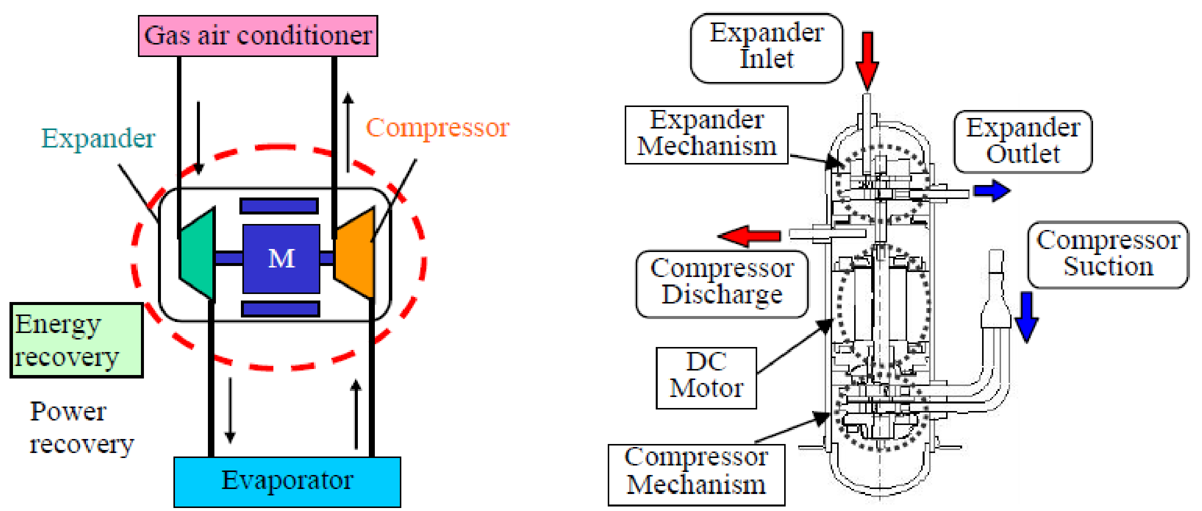
Figure 3. Improvement of a heat transfer rate of a water heat exchanger



(a)Flow pattern **(b)Vortex structure**

Figure 4. Images around the dimple.

提昇 CO_2 系統效率之一些做法 - Expander



a) Outline of improvement of efficiency

b) Schematic drawing

Figure 7. Outline of Expander-Compressor Unit

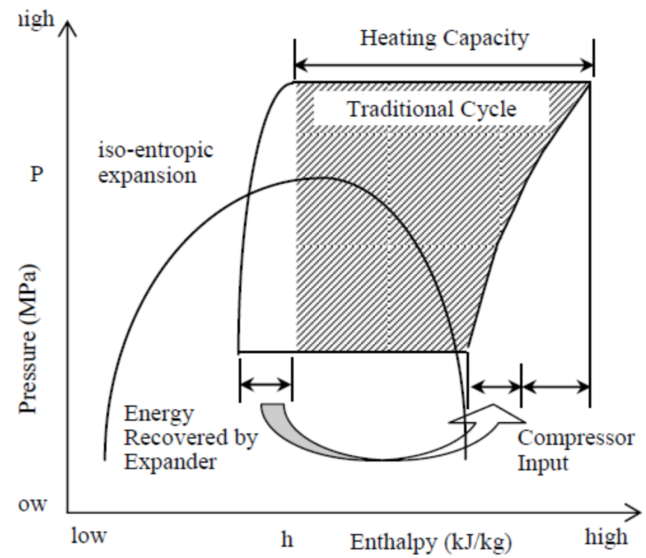


Figure 6. Principle of expander-compressor

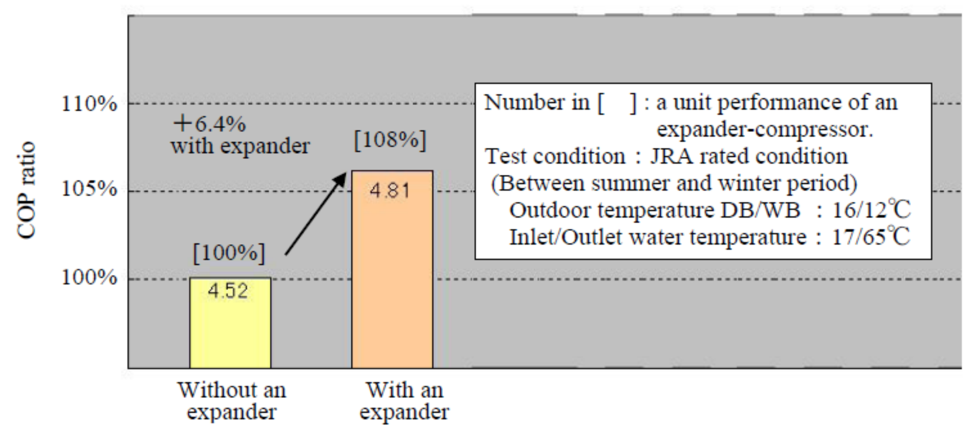


Figure 8. Evaluation of system performance

提昇 CO_2 系統效率之一些做法

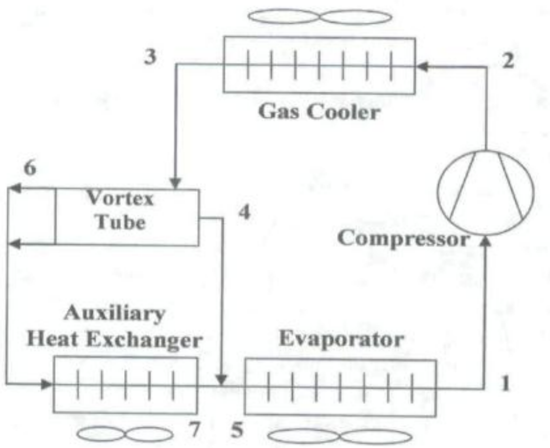


Figure 6. Schematic of transcritical CO_2 cycle with a vortex tube as the expansion device.

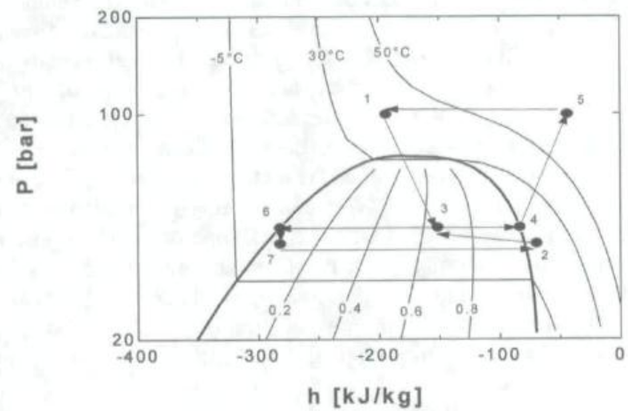


Figure 8. Ejector expansion transcritical CO_2 cycle in a p-h diagram (properties from EES [Klein 2004]).

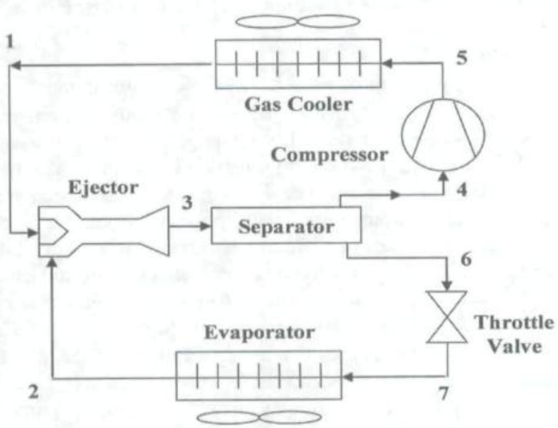


Figure 7. Schematic of transcritical CO_2 cycle with an ejector expansion device.

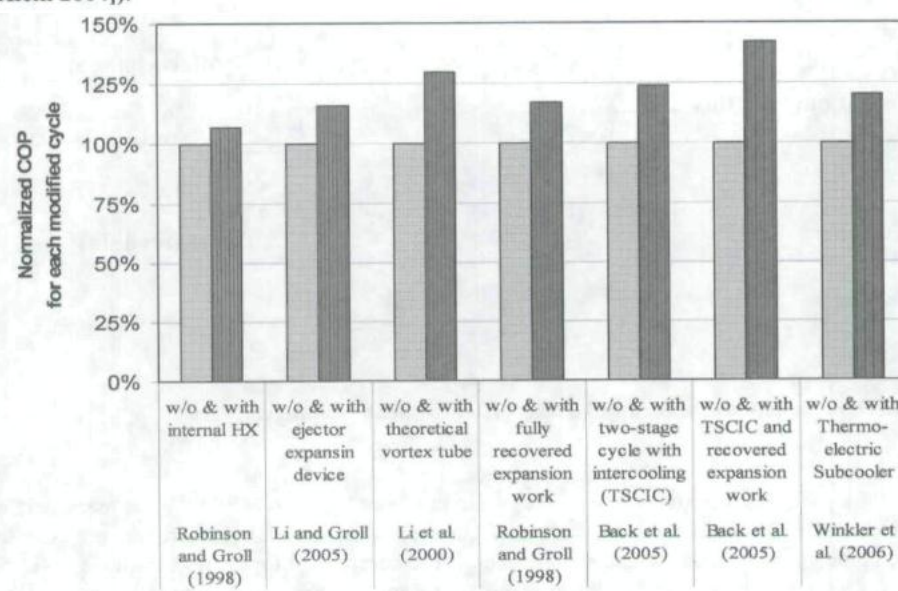


Figure 9. Relative system performances of various transcritical CO_2cycles.

提昇 CO_2 系統效率之一些做法
2-stage cycle with gas injection

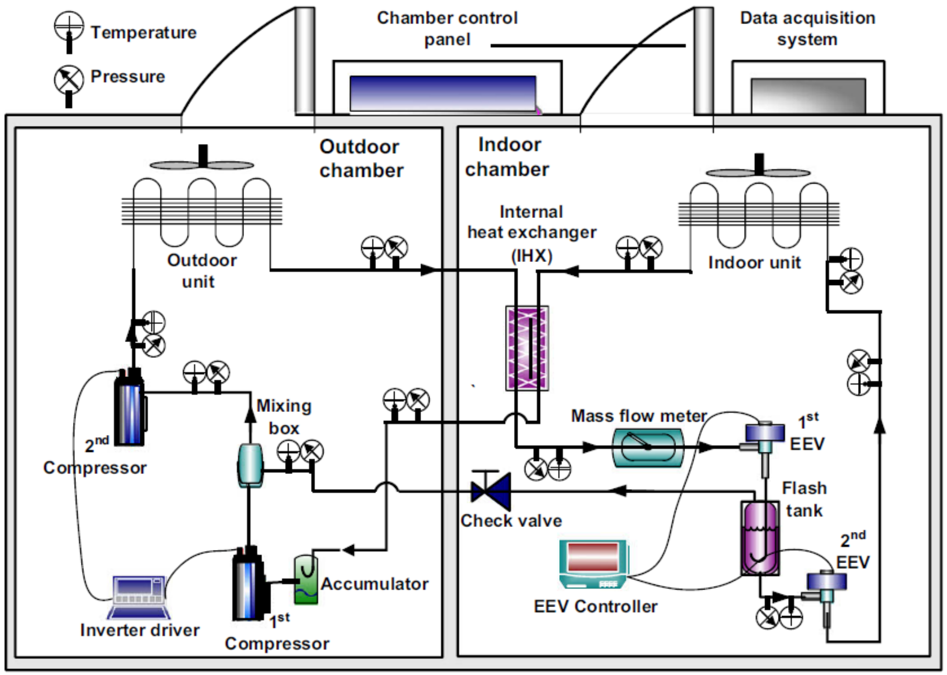


Fig. 1 – Schematic diagram of the experimental setup.

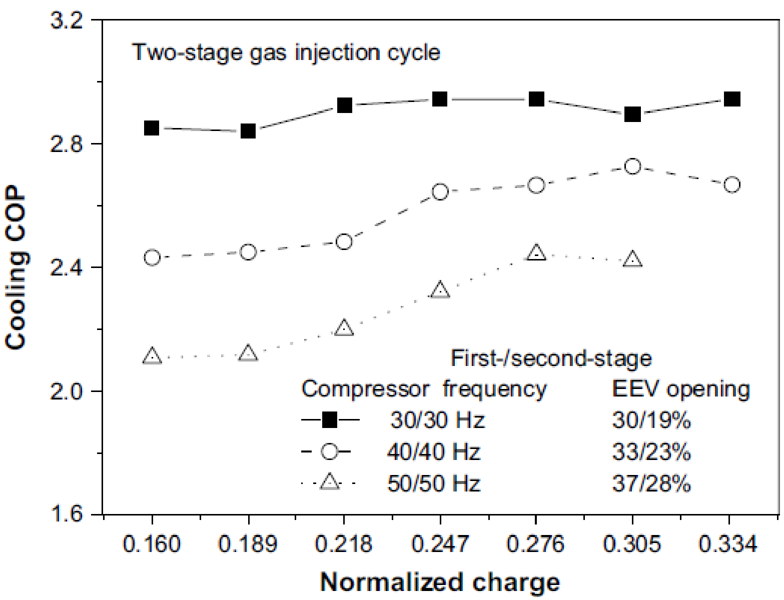


Fig. 3 – Variation of cooling COP with normalized charge.

CO_2 熱交換器 - Microchannel

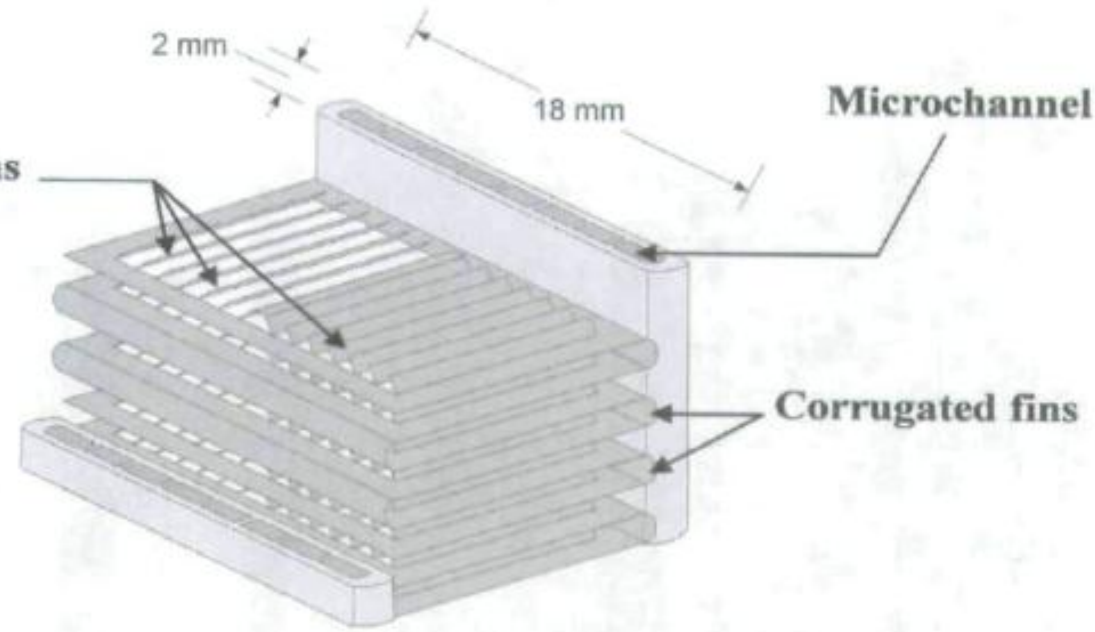


Figure 10. Schematic makeup of a microchannel heat exchanger.

氣冷式系統中間壓力控制設計

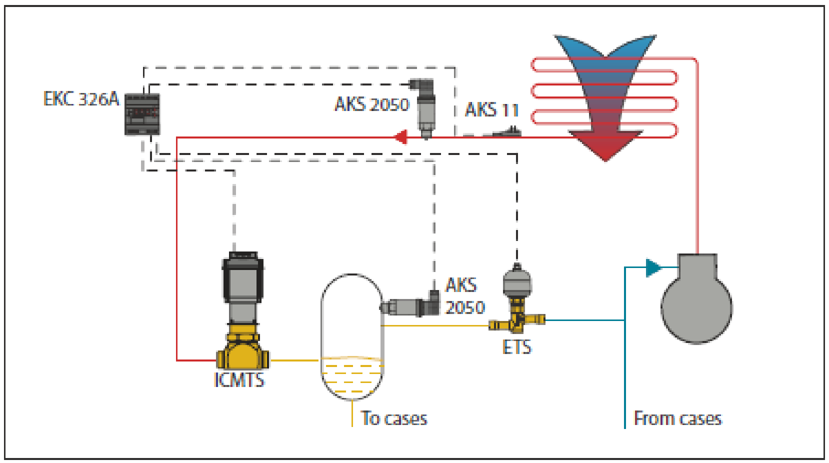


Figure 1.3.2: Intermediate pressure control by ETS stepper motor valve

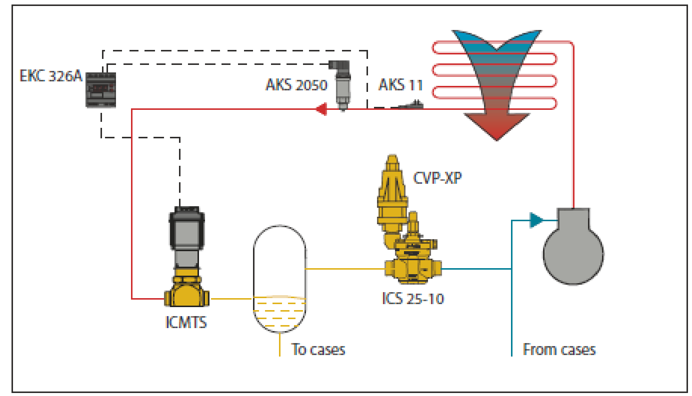


Figure 1.3.3: Intermediate pressure control by a CVP-XP pilot valve

System	Air-cooled gas cooler with a motor valve	Air-cooled gas cooler with a mechanical control valve	Air-cooled gas cooler with additional compressor
Advantages	Flexible system	Simple to use	High efficiency Reduced energy consumption
Limitations	Efficiency is lower than that of the system with parallel compression	Only one set point available	Cost and complexity

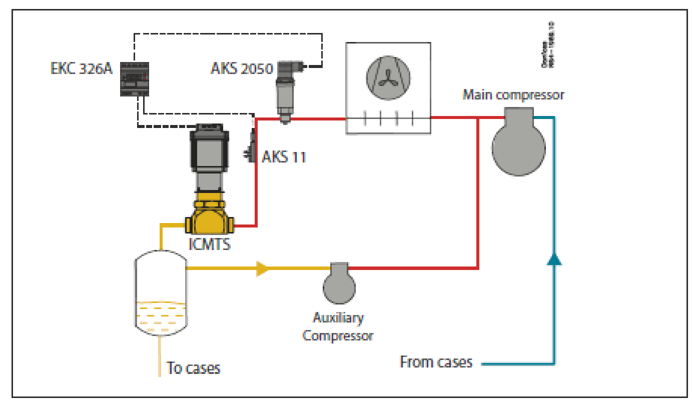


Figure 1.3.4: Intermediate pressure control with an auxiliary compressor

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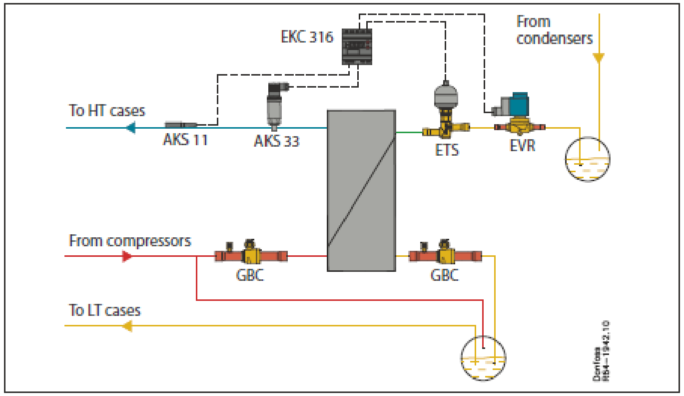
Cascade HX 系統設計

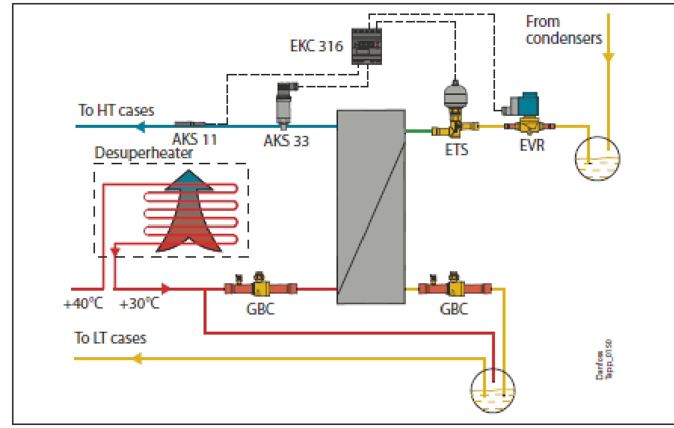
2.2 Standard cascade heat exchanger



- HP vapour refrigerant
- HP liquid refrigerant
- LP vapour refrigerant
- LP liquid refrigerant

Fig. 2.2.1: Cascade heat exchanger with direct discharge from the compressor

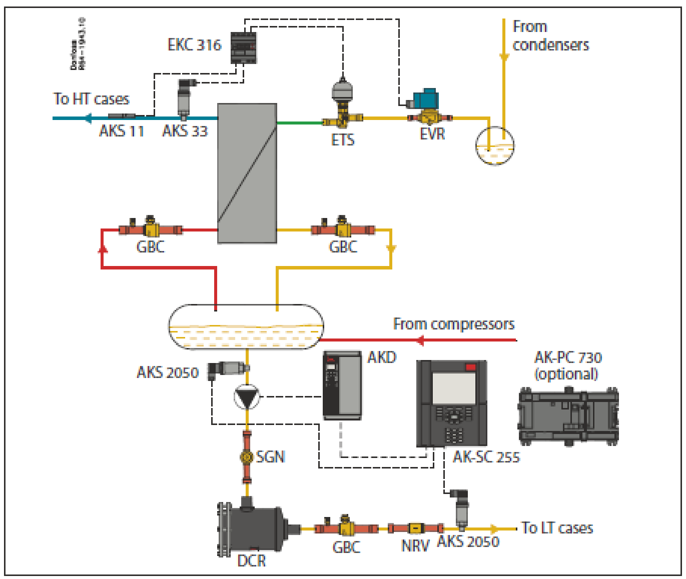
2.2 Standard cascade heat exchanger (continued)



- HP vapour refrigerant
- HP liquid refrigerant
- LP vapour refrigerant
- LP liquid refrigerant

Fig. 2.2.2: Cascade heat exchanger with desuperheater

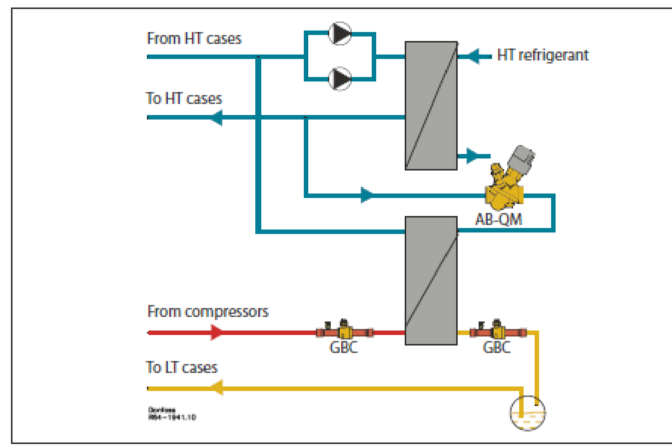
2.3 Cascade heat exchanger with an intermediate vessel



- HP vapour refrigerant
- HP liquid refrigerant
- LP vapour refrigerant
- LP liquid refrigerant

Fig. 2.3.1. Cascade heat exchanger with pumped circulation

2.4 Cascade heat exchanger with secondary cooling



- HP vapour refrigerant
- HP liquid refrigerant
- LP vapour refrigerant

Fig. 2.4.1: Brine cooled cascade heat exchanger

System	DX	DX with CO_2 vessel	Secondary cooling
Advantages	Simple piping	No need for equalisation line	Stable operation
Limitations	An equalisation line is required	Relatively complicated piping	Lower efficiency of the system

Methods for High Side Pressure Control

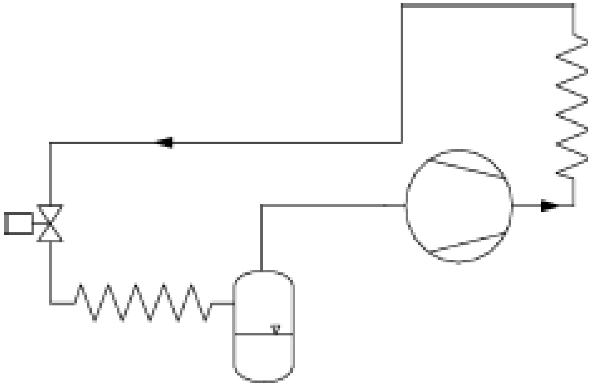


Fig. 18. System with low-pressure receiver.

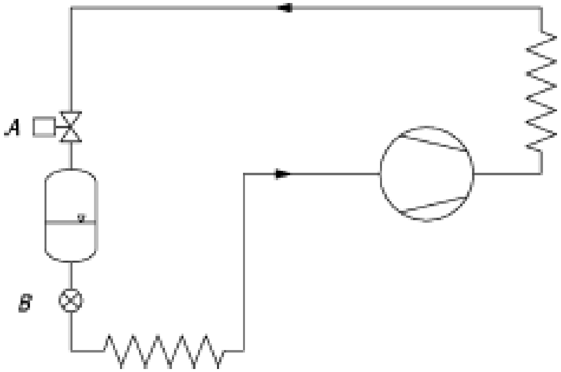


Fig. 19. System with in-line medium-pressure receiver.

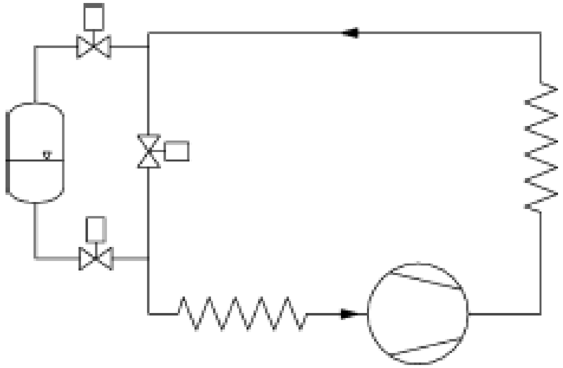


Fig. 20. System with medium-pressure receiver.

Cascade 系統設計型式

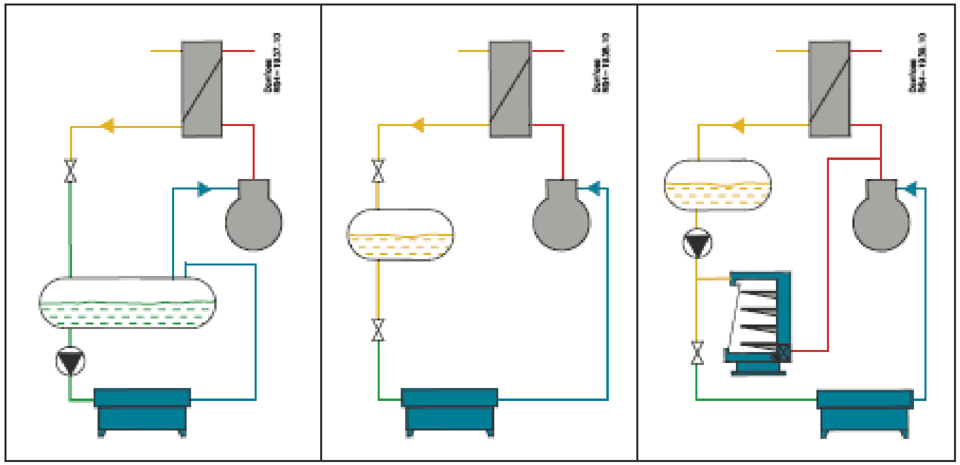


Fig. 3.1.1: Pumped system　Fig. 3.1.2: DX system　Fig. 3.1.3: Combined system

System	DX	Pump	Combined
Advantages	Simple. Does not require pumps	High efficiency; CO_2 can be pumped relatively large distances	Efficiency provides 2 temperature levels
Limitations	Energy efficiency is not optimal	Relatively complex and expensive; Energy consumption from the pump is often very high for small systems	Relatively complex; The most expensive out of the three alternatives

國立交通大學 National Chiao Tung University

簡單總結

1. CO_2系統市場以日本歐洲最為活絡，但在全球暖化的大趨勢下，勢必成為市場的發展重點；尤其在取熱暖房的應用上極具潛力.
2. CO_2 冷媒由於在臨界點上工作，其物理性質的變化與熱傳性能與傳統冷沒有極大的差異，尤其再準臨界點附近有非常大的變異。
3. 對 CO_2, 系統而言，潤滑油對熱傳矽能與流動特性有相當大的影響。
4. CO_2 系統之COP與高壓的關係和傳統冷媒不同，通常系統的COP 最高值發生在某一特定壓力下，而非隨壓力升高而下降。

Short summary, conti..

5. 通常 CO_2 系統效率比傳統冷媒低，因此為提昇系統效率，可藉由一些系統的改善設計如：

   expander, modified 2-stage system, work recovery with ejector system, and intercooling system.

6. CO_2 系統常見之熱交換器性能改善方法為：

   Microchannel Heat exchanger, Microfin tube, vortex generator (e.g. dimple).

感謝您的聆聽